EFFECTS OF ALUMINUM EVAPORATOR AND 7 MM COPPER CONDENSER COILS ON A 60,000 BTU INPUT PACKAGED REFRIGERATION SYSTEM

A Thesis by

Mehraz Khan

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The following faculty members have examined the final copy of this thesis for form and content, and recommend that it be accepted in partial fulfillment of the requirement for the degree of Master of Science, with a major in Mechanical Engineering.

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Rajeev Nair, Committee Chair

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Shuang Gu, Committee Member

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Krishna Krishnan, Committee Member

DEDICATION

To my loving and supportive Parents Iftikhar Ali Khan and Sobia Khan, who have made every step in my life easy and effortless. To my grandmother Zahida Khan and Uncle Tabraiz Khan who always dreamt of me being successful. My younger siblings, Laiba, Rija and Arib to whom I have always needed to explain the value of education and hard work. Finally, to Maheen Khan, who has provided support and encouragement while I worked on this Thesis

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ABSTRACT

The purpose of this study was to determine the effect of aluminum evaporator coils and 7 mm copper condenser coils on a 60,000 Btu refrigeration unit. The development of this unit is due to a VAVE (value analysis and value engineering) effort, and the new design is significantly cheaper than the current design. New components are smaller and use mostly aluminum compared to copper; the heat transfer is significantly lower in aluminum, but other design improvements help in maintaining consistent performance.

On the evaporator side, performance loss going from copper tubing to aluminum tubing will be made up by the lanced fins, which use the same material but have increased surface area. A blower which is intended for lower power usage increases efficiency and is designed with a new venture and scroll sides to maximize airflow. On the condenser side, the copper tube diameter is reduced from 3/8 inch to 9/32 inch and spacing between the tubes decreases from 1 inch to 0.8 inch. The performance loss due to this change is overcome by modified sub-cool loop circuits. The old design maintains certain AHRI (Air-Conditioning, Heating, and Refrigeration Institute) ratings which are met by the new cheaper model.

The unit will be certified by CSA (Canadian Standards Association) to comply with safety standard UL-1995 and UL 60335-2-40. Unit has been tested for performance per standard AHRI 210/240. The refrigeration system is tested in a psychrometric test chamber while varying indoor and outdoor conditions. Thermocouples are used to measure sub-cool and superheat of the system and pressure taps are used to measure the suction pressure, discharge pressure and liquid pressure in the system.

Given our test results, we developed a quality design which yielded us a higher capacity, better efficiency and lower cost. The 5 ton air-conditioning unit yielded us better performance than its predecessor model. Given the ease of manufacturability and performance numbers, it can be produced in large quantities and used in various parts of the world.

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CHAPTER 1

INTRODUCTION

1.1 Background

Air conditioners have been around for over a century. The first air conditioner was developed by a New York (NY) native in 1902 [1]. Willis Haviland Carrier, an engineer, started trying different things with the laws of humidity control to take care of an application issue at a printing plant in Brooklyn, NY. Acquiring from the ideas of mechanical refrigeration built up in years before, Carrier's framework, sent air through loops loaded up with cold water, cooling the air while simultaneously evacuating dampness to control room mugginess. In 1933, the Carrier Air Conditioning Company of America built up an AC system utilizing a belt-driven condensing unit and related blower, mechanical controls, and evaporator coil, and this gadget turned into the model in the developing U.S. commercial center for air-cooling [2].

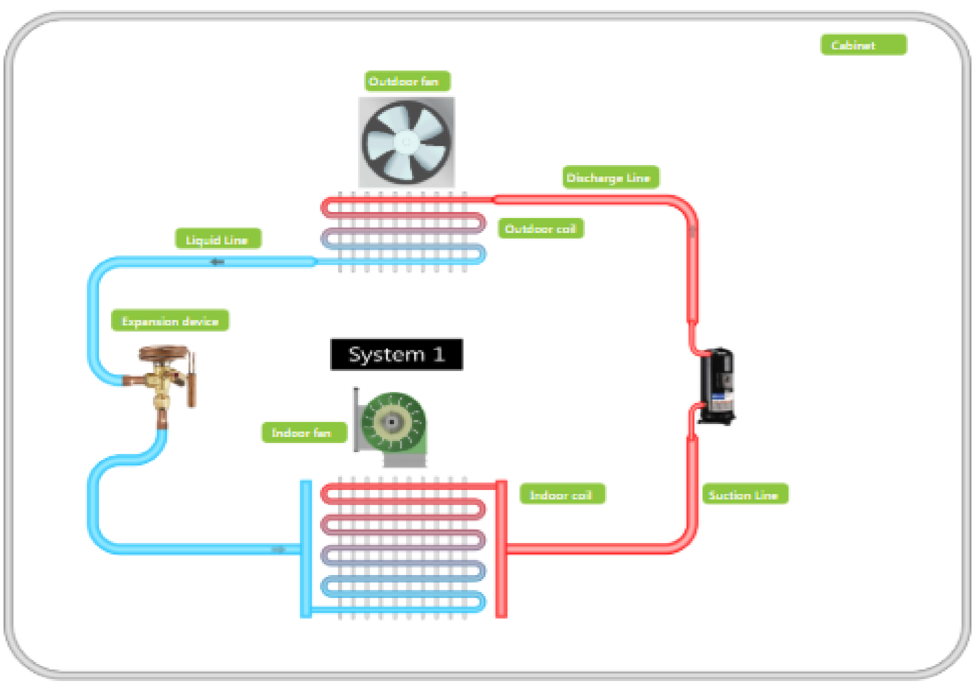


Figure 1.1 Refrigeration Cycle

The refrigeration cycle is used to determine the process of cooling down refrigerant in a system and transferring the heat to air using an evaporator coil [3]. For modern residential systems, refrigerant R410A is used [4]. The compressor is used to discharge high pressure and high temperature into the discharge line, and the vapor travels to the condenser coil where it is cooled down using a condenser fan. The fan pulls air across the coil and a third of the way inside the complex tubing, and subsequently the vapor starts to convert into liquid. A two phase flow was used at this point and by the time refrigerant leaves the coil, it is completely in liquid form. The refrigerant is converted into a low-pressure, low-temperature liquid. Further in the stream is a thermal expansion valve, which controls the flow of the refrigerant in the system. [5]

After passing through the expansion valve, the refrigerant flows in to the evaporator coil where a blower draws the air through it and provides it to the household or to the mirrors in a lab environment [6]. Through the process of heat transfer, the air passing through the coil picks up the low temperature, making the refrigerant hotter and into vapor form again. By the time, the refrigerant leaves the indoor coil, it is in vapor form yet again.

For the system in-discussion here, a package unit is used and primarily, ways are developed to effectively control the phase flow in the system. Circuits are designed that make sure the liquid leaving the condenser coil is as cold as possible. The circuiting in the indoor coil makes sure all the heat is transferred to the air stream. The compressor pressurizes the system for better flow. Sub-cool and superheat are maintained to reduce flashing any liquid into the compressor or for any vapor to leave the condenser coil.

1.2 Literature Review

The main components in the refrigeration system are as follows:

A compressor device that compresses something, adds pressure. In a refrigerant system, it compresses a refrigerant. There are multiple types of compressors; reciprocating, scroll, rotary and screw. For the system under consideration, a scroll system is used. Léon Creux first patented a scroll compressor in 1905. [7] Scroll compressors sets can separate to "ride over" liquid refrigerant. However, liquid can still damage the compressor by washing the oil out of the bearings. [8, 9]

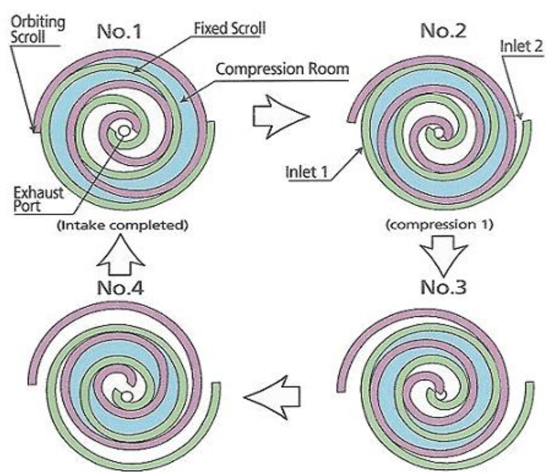


Figure1.2 Scroll Compressor [9]

Discharge pressure represents potential energy in the system and is required to move refrigerant through the system.

A condenser coil is a device that condenses refrigerant, converting R410a from vapor to liquid. It removes heat from the system and the phase change transfers maximum heat. Heat is removed by convection from refrigerant to tubes, through tubes and fins, and by convection from fins to air. There are multitude of technologies already in use and being investigated to accomplish heat transfer as efficiently and effectively as possible. Tube types used are based on design; Smooth Tube (no enhancements), Grooved Tube (axial or helical), Cross-Hatch Tube (welded seam), geometry; Smaller Diameters (5/16 inch, 9/32 inch, etc.), In 2006, Jin Jeong and Chang Kim studied thermal conductance in fin-tube coils with 7mm tubes [10] and Microchannel (Various Geometries); and Materials (copper, aluminum, etc.). [11] In 1988, Takenori Notoya,Takao Hamamoto, and Kozo Kawano found formicary corrosion in copper tubes [12]

An evaporator is a device that evaporates something to change phase from liquid to vapor. It absorbs heat from the system and the phase change transfers maximum heat. Evaporators absorb heat from air so that it can be used to cool the conditioned space. Heat is extracted from the warm supply air. Heat is absorbed by convection from the air to the fins, conduction through fins and tubes, and by convection from tubes to the refrigerant. Evaporators use similar tubes to condensers. A smooth tube is used in this system.

Most systems use copper and aluminum tubing in the same unit, which means both metals are brazed. We find galvanic corrosion on Al and Cu being in the same environment [13], to help reduce this corrosion Wallace Cook and William Buehring developed Al to Cu connector in 1940 used mostly in wires [14]

Fins are used on all coils; they maximize the surface area for heat transfer from the air. Types of fines are Simple Plate Fins, Rippled Edges, Sine Wave, Raised Lances, Louvers, and Microchannel. [15] Spine fine are special aluminum brush coils developed by Trane Corporation™ in 1998. [16] For the condenser, sinewave and Lanced fins are used for the evaporator. Oved Hanson developed the lanced fin in 1988. [17] Fins increase the contact surface area, increases the turbulence (mixing) and break up the boundary layer. [18]

The expansion device converts refrigerant from a high-pressure, high-temperature liquid to a low-pressure, low-temperature two-phase mix. [19] The two-phase mix is partially vapor, which requires more space (volume) than the liquid entering the device. As the percentage of vapor in the two-phase mix (i.e., its "quality") increases by absorbing heat, it will require more space. Types of devices used in residential air conditioning are capillary tubes, orifices, Thermostatic Expansion Valve (TXV) and Electronic Expansion Valve (EEV). [20] TXVs are used in this system. TXVs control refrigerant flow by opening or closing a needle valve in the body in order to maintain a constant evaporator superheat.

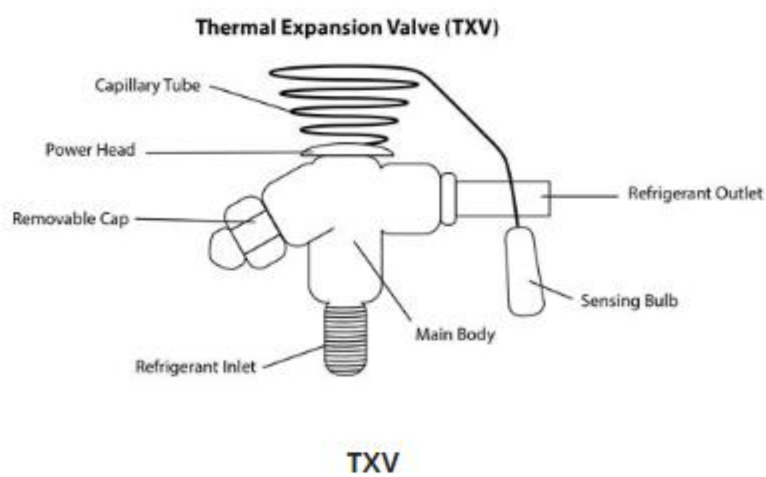


Figure1.3 Thermal Expansion Valve [25]

1.3 Objectives

The main objective of this thesis is to research and prove the use of aluminum in the HVAC (heating, ventilation and cooling) industry. The evaporator coils have traditionally used aluminum fins and copper tubes for maximum efficiency in an HVAC system. [21] The use of aluminum for tubes in an evaporator coil to be significantly cost effective and how it reduces the heat transfer has been proven. [22] But with a fin that has better surface area, the performance can be gained back. This thesis strives to prove that the reduction in condenser coil surface area effects performance, but by lowering the liquid temperature in the leaving refrigerant a better performance can be achieved. The use of a sub-cool loop in the circuiting, sends the cooled liquid back to the bottom of the condenser coil for a uniform and lower liquid temperature. The secondary objective in this research is to design copper tubing for refrigerant flow, for the least amount of charge, and use in an effective manner, given the concisely available space in the cabinet. An FEA (finite element analysis) study was conducted on the copper tubing to reflect the vibration in the system. Using the analysis, experiments are conducted using a strain gage, to ascertain the high stress points using actual physical readings. The newly developed system is also put in a psychrometric chamber and tested at multiple conditions to monitor the performance and reliability of the design. A second sample is built using production grade material and multiple tests were conducted to verify performance. The unit was also tested in a highly accelerated life cycle testing chamber to mimic over 5 years of performance in under 16 weeks, to verify the reliability of each component in the refrigeration system [23].

1.4 Methodology

We use variable steps to test our system. Simulations are ran initially using a software called IMAP™.

Then the unit is tested in the psychrometric test chamber using the following test plan.

Table 1.1 AC Test Plan

Test #	ID CFM	Test Description	ID DB/WB	OD DB/WB
	Indoor CFM		Indoor Dry Bulb/Wet Bulb / °F	Outdoor Dry Bulb/Wet Bulb / °F
1	1800	AFULL	80/67	91/71.5
2	1800	BFULL	80/67	82/65
3	1800	Cd	80/56	82/65
4	1800	AFULL	80/67	91/71.5
5	1800	BFULL	80/67	82/65
6	1800	Cd	80/56	82/65
Cooling UL Test				
7	1800	AFULL	80/67	91/71.5
8	1800	UL Input Test (230V) Cooling	80/67	95/75
9	1800	M1(240V) HI	80/67	104/80
10	1800	M1(208V) HI	80/67	104/80
Cooling AHRI Operational Test				
11	1800	Maximum Operating Conditions (Run 187V) Change Transformer Tap to 208V	80/67	115/75
12	1800	Voltage Tolerance-Hi (Run 254/Restart 218.4V) Switch back to 240V	80/67	95/75
13	1800	Voltage Tolerance-Lo (Run 180/Restart 180V) Change Transformer Tap to 208V	80/67	95/75
14	1800	K1 (Run Voltage 208/Restart Voltage 218V)	100/72	120/95
15	1800	C7-High Ambient (Run Voltage 187)	75/62	125/95
16	1800	High Superheat	85/75	95/75

Table 1.1 AC Test Plan (continued)

Service Data (no flow meter)				
17	1800	AFULL	80/67	91/71.5
18	2000	Cooling Service Data Hi		
19	1600	Cooling Service Data Lo		
Performance Confirm (no flow meter)				
20	1800	AFULL	80/67	91/71.5
21	1800	BFULL	80/67	82/65
22	1800	Cd	80/56	82/65

Table 8 from AHRI Standard 210/240-2017 suggests the following tests.

Table 1.2: AHRI Test Conditions [24]

Test Name	Air Entering Outdoor Unit [2] (°F)	Air Entering Indoor Unit [2] (°F)	Compressor Speed [3]	Indoor Airflow [4]
Cooling Mode				
A_{Full}	95.0 / 75.0 [5,6]	80.0 / 67.0	$Full_C$ [12]	$Full_C$ [12]
B_{Full}	82.0 / 65.0 [5,6]	80.0 / 67.0	$Full_C$	$Full_C$
B_{Low}	82.0 / 65.0 [5,6]	80.0 / 67.0	Low_C	Low_C
C_{Full}	82.0 / 58.0 [5,6]	80.0 / 57.0 [7]	$Full_C$	$Full_C$
C_{Low}	82.0 / 58.0 [5,6]	80.0 / 57.0 [7]	Low_C	Low_C
D_{Full}	82.0 / 58.0 [5,6]	80.0 / 57.0 [7]	$Full_C$	$Full_C$ [8]
D_{Low}	82.0 / 58.0 [5,6]	80.0 / 57.0 [7]	Low_C	Low_C [8]
E_{Int}	87.0 / 69.0 [5,6]	80.0 / 67.0	Int_C	Int_C
F_{Low}	67.0 / 53.5 [5,6]	80.0 / 67.0	Low_C	Low_C
G_{Low}	67.0 / 58.0 [5,6]	80.0 / 57.0 [7]	Low_C	Low_C
I_{Low}	67.0 / 58.0 [5,6]	80.0 / 57.0 [7]	Low_C	Low_C [8]
Cooling Mode Operation Tests				
Voltage Tolerance	95.0 / 75.0 [5]	80.0 / 67.0	$Full_C$	$Full_C$
Low Temperature	67.0 / 57.0	67.0 / 57.0	$Full_C$	$Full_C$
Insulation Efficiency	80.0 / 75.0	80.0 / 75.0	$Full_C$	$Full_C$
Condensate Disposal	80.0 / 75.0	80.0 / 75.0	$Full_C$	$Full_C$
Maximum Operation	115.0 / --	80.0 / 67.0	$Full_C$	$Full_C$
Extra High Maximum Operation (Optional)	125.6 / --	80.0 / 67.0	$Full_C$	$Full_C$

The 3 main rating tests A_{full}, B_{full} and D_{full} required steady state capacity, EER and heat balance; Capacity Calculations given by AHRI:

AHRI STANDARD 210/240-2017 [24]

11.1.1.1 *Total Cooling Capacity (Indoor Air Enthalpy Method).* The net total capacity for all steady state cooling tests shall be calculated using Equation 1.2 for Blower Coil Systems or using Equation 1.3 for Coil-only Systems.

$\dot{q}x$=60 · $\dot{Q}(h\,a1-h\,a2)v'n(1+Wn)$ 1.1
$\dot{q}tci$,=$\dot{q}x$ + $\dot{q}duct,ci$ 1.2
$\dot{q}tci$,=$\dot{q}x$+ $\dot{q}duct,ci-\dot{q}sadj,x$ 1.3

Where Equation 1.4 shall be used when the Indoor Unit is in the indoor psychrometric chamber, Equation 1.5 shall be used when the indoor section is completely in the outdoor chamber. Equation 1.6 is shown for reference. Duct loss, $\dot{q}duct$,, shall be set to 0 for steady state tests C and G.

$\dot{q}duct$,=$UAID,si(ta1-ta2)$ 1.4
$\dot{q}duct$,=$UAID,ro(ta0-ta1)+UAID,so(ta0-ta2)+UAID,si(ta1-ta2)$ 1.5
$v'(1+Wn)=vn$ 1.6

11.1.1.3 *Total Cooling Capacity (Refrigerant Enthalpy Method).* The net total capacity for all steady state cooling tests shall be calculated as follows..

$\dot{q}ref$,=$x\dot{m}ref,x(h\,r2-h\,r1)-\dot{q}sadj,x$ 1.7

11.1.1.4 Indoor motor heat capacity adjustment, $\dot{q}sadj$.

For all Blower Coil Systems:
$\dot{q}sadj$,=3.412·$Pfan,x$ 1.8

11.1.1.5 *Heat Balance.* If using the outdoor enthalpy as an alternate method, use Equation 11.12, or if using refrigerant enthalpy as an alternate method, use Equation 1.10.

$HBx=\dot{q}tci,-\dot{q}tco,x\dot{q}tci,x$ 1.9
$HBx=\dot{q}tci,-\dot{q}ref,x\dot{q}tci,x$ 1.10

11.1.2 *Cooling Steady State Power.* The steady state power, $Ptot$,, shall be as measured during test, adjusted as follows, using Equation 1.11 for Blower Coil Systems.

$Ptot$,=Pm,x+ $Padj$ 1.11

Where:

P_{sadj},=365 1000 $\cdot \dot{Q}_s$ 1.12

Where 365 watts is a default power consumption per 1000 scfm, and P_{adj} only applies for Constant-volume AMS per Section 6.1.5.1.3[20] (*Padj* is 0 for all other Blower Coil Systems).

11.1.3 *Cooling Steady State Efficiency, EER.* The steady state efficiency shall be calculated as follows.

$EER_x = \dot{q}_{tci,ot,x}$ 1.13

11.1.4 *Cooling Cyclic Net Total Capacity*. The net total capacity for all cyclic cooling tests (tests D and I) shall be calculated as follows. $\dot{Q}_{mi}$, c_{pa}, v'_n, $P_{fan,}$, and W_n shall be the average values recorded during the corresponding dry coil steady state tests (tests C and G).

q'_{cyc},=60 $\cdot$ $\dot{Q}_{mi} c_{pa2} \Gamma v'_n (1+W_n) - q_{cadj,x}$ 1.14

$\Gamma = FCD* \int [t_{a1}(\theta) - t_{a2}(\theta)] d\theta \theta 2 \theta 1$ 1.15

Where $FCD*$ is calculated per Section 5.2.3.3 using values measured during C & D tests.

For Blower Coil Systems with Constant-volume AMS or Constant-torque AMS which has the blower disabled for Cyclic Test:

q_{cadj},=3.412$\cdot P_{fan,x} \cdot [\theta 2 - \theta 1]$ 1.16

For all other Blower Coil Systems:

q_{cadj},=0 1.17

For all other Non-ducted Systems:

q_{cadj},=3.412$\cdot E_{fan,x}$ 1.18

For Non-ducted Systems, subtract the electrical energy used by the indoor fan, E_{fan}, during the 3 minutes after compressor cutoff from the Non-ducted System's integrated cooling capacity, q'_{cyc},.

See Appendix definitions for nomenclature.

Using the calculations above, we run an A test , B test and Cd test to calculate heat balance (no more than 6%), EER (over 11.2) and capacity, no lower than 57000 btu (95% of 60000btu). The A and B test are steady state tests run in a psychrometric test call, the test conditions in Table 1.1 are to be met. The cell is conditioned for an approximately 2 hours given the stability of the cell and unit, the test is run and readings are taken, given in appendix 4A and 4C. The cyclic test is run where the cell is conditioned as per Table 1.1 and the 3 cycles are run yielding 3 different Cd values that are then used to calculate Seasonal Energy Efficiency Ratio)

For CFD

Based on the polynomial equation viscous resistance co-efficient and inertial resistance coefficient were calculated

$$\Delta p = avi + bvi2 \qquad 1.19$$

$$a = -\frac{\mu}{\alpha} . \Delta n \qquad 1.20$$

$$b = -\frac{1}{2} . C2 . \rho . \Delta n \qquad 1.21$$

where,

C_2 = Inertial resistance factor

v_i = Velocity, m/s

ρ = Density of the flowing medium (kg/m^3)

μ = Dynamic viscosity of the flowing medium ($N\text{-}s/m^2$)

Δn = Thickness (m) (0.0879856 m)

$y = 10.8x^2 + 16.536x$ (equation obtained from best fit curve)

Comparing Eq. 1 with Eq.2 we obtain the following values

a=16.536 and b= 10.8

Equating the values viscous resistance and inertial resistance co-efficient are obtained as follows:

Viscous Resistance Co-efficient $\frac{1}{\alpha}$ = 1.0177E+07

Inertial Resistance Co-efficient (C_2)= 208.343

CHAPTER-2

THEORETICAL FRAME WORK

2.1 Condenser Outdoor Coil Design

The outdoor coil was developed using trial and error.

An early coil idea contained multiple stacked circuit, the liquid temperature coming out of the coil going towards the Thermal Expansion Valve was at a high 104^0 Fahrenheit, as measured by simulation. After a number of redesigns, the final circuit was converted into a sub-cool loop circuit. The unit uses, 8 circuits; 6 stacked and 2 sub-cool loops. It is divided into two parts, the left side and right side. Total height of each side is 19.2 inches and total length of the coil is 77.77 inches.

Figure 2.1 Right Side Coil Circuit

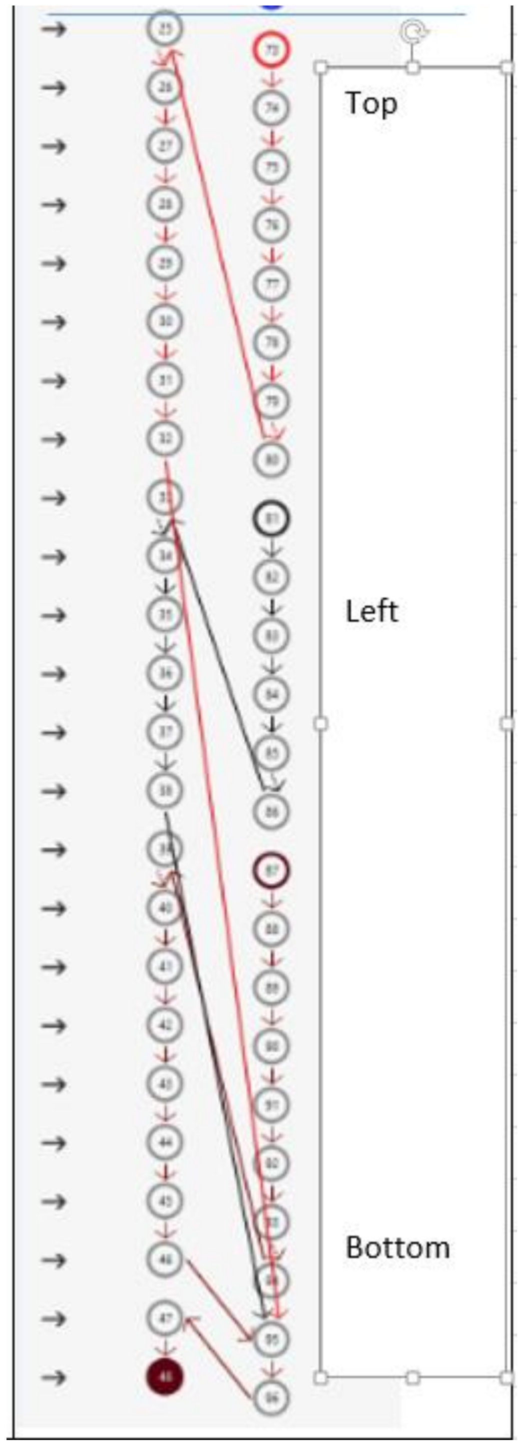


Figure 2.2 Left Side Coil Circuit

2.2 Evaporator Indoor Coil Design

The biggest change in the refrigeration system is the indoor coil. The CFD (computational fluid dynamics) analysis was initially run on the indoor unit to determine the airflow of the system. The air is taken into the system from a return duct and is drawn through the coil. Multiple coil circuits were run and best balanced circuits were picked.

The final design includes a 10 circuit coil stacked on top of each other. The temperature across the tubes were measured and the 'entering' and 'leaving' circuits are thermo-coupled. The purpose of this activity was to make sure that there is no liquid leaving the indoor coil.

Table 2.1 Indoor Coil Circuit Temperatures

Description of Data Collected:	Average	Max	Min	Range
ID Circuit Temps				
Inlet Circuit #1 (°F)	52.30	52.48	52.07	0.41
Inlet Circuit #2 (°F)	53.10	53.27	52.86	0.41
Inlet Circuit #3 (°F)	52.03	52.20	51.78	0.42
Inlet Circuit #4 (°F)	52.49	52.69	52.26	0.43
Inlet Circuit #5 (°F)	53.49	53.80	53.20	0.60
Inlet Circuit #6 (°F)	52.89	53.16	52.61	0.56
Inlet Circuit #7 (°F)	53.32	53.59	53.00	0.59
Inlet Circuit #8 (°F)	52.10	52.33	51.84	0.49
Inlet Circuit #9 (°F)	53.27	53.51	53.03	0.48
Inlet Circuit #10 (°F)	52.51	52.76	52.27	0.48
Outlet Circuit #1 (°F)	73.50	73.96	73.01	0.95
Outlet Circuit #2 (°F)	53.34	53.68	53.02	0.66
Outlet Circuit #3 (°F)	54.73	55.11	54.34	0.77
Outlet Circuit #4 (°F)	71.19	71.63	70.79	0.84
Outlet Circuit #5 (°F)	69.95	70.38	69.58	0.80
Outlet Circuit #6 (°F)	74.64	75.04	73.97	1.07
Outlet Circuit #7 (°F)	75.73	76.01	75.41	0.61
Outlet Circuit #8 (°F)	72.79	73.10	72.47	0.62
Outlet Circuit #9 (°F)	73.05	73.35	72.76	0.59
Outlet Circuit #10 (°F)	72.77	73.09	72.43	0.66

The final circuit coil diagram as seen in figure 2.3, shows the air flowing from the coldest side of the coil to the hottest side, it makes sure there is no heat transfer between the coil tubes.

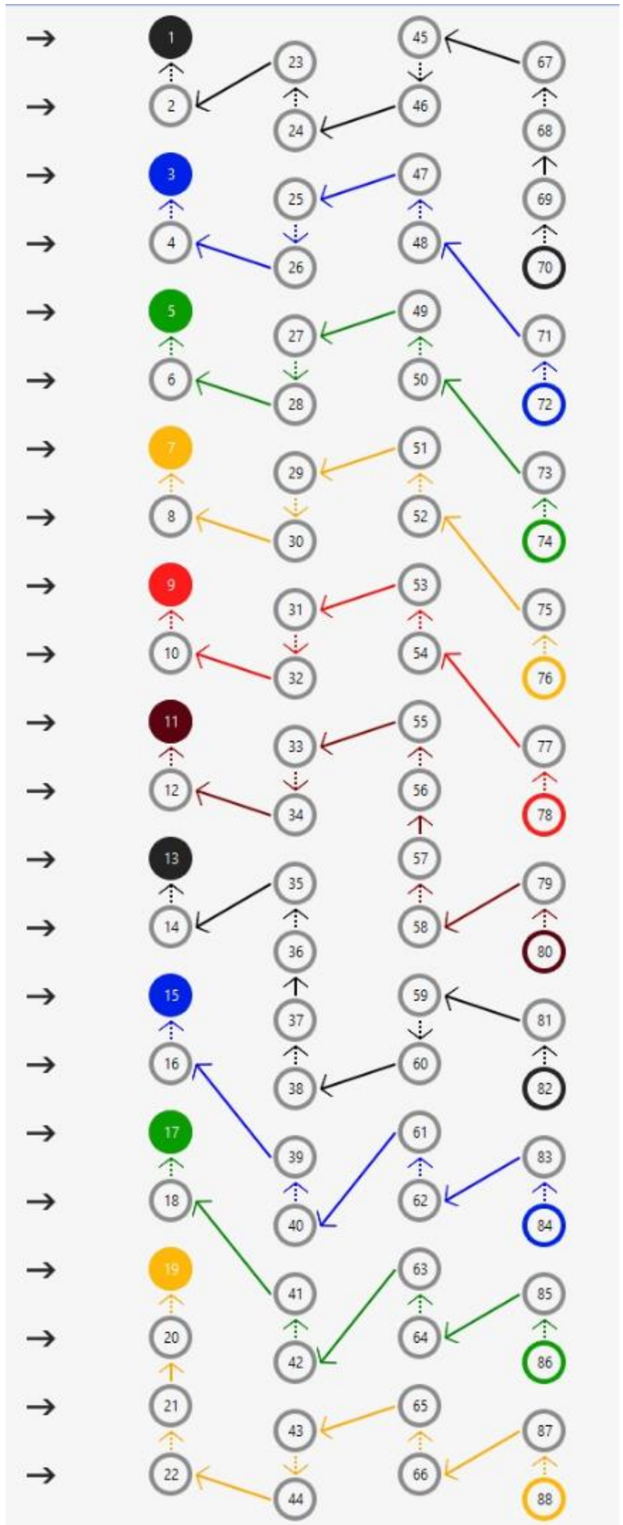


Figure 2.3 Evaporator Indoor Coil Circuit

2.3 Compressor

The compressor used was bought from an external supplier LG®.

Table 2.2: LG Compressor Specifications [26]

1	Compressor Model Name	ABG051KAC
2	Compressor Type	Hermetic Motor Compressor
3	Compression Type	Scroll
4	Displacement	2.94 in^3/rev (48.2 cm^3 / rev)
5	Refrigerant	R410A
6	Safety approval	UL / CSA
7	Oil / Oil Charging Amount	POE(RB32G) / 1,280 ± 20 cc (43.3 Oz)
8	Nitrogen Gas Holding Pressure	5.7 ± 2.85 psi (0.4 ± 0.2 kg/cm^2G)
9	Painting	Black Color Paint
10	Net Weight (Including Oil, Reference)	71.4 ±1.1 lb (32.4 ± 0.5 kg)
11	Suction Tube I.D.	Φ 0.878 ~ 0.886 inch (Φ22.4±0.1mm)
12	Discharge Tube I.D.	Φ 0.500 ~ 0.509 inch (Φ12.8±0.2mm)

1.2 Motor

1	Motor Type / Starting Type	Single Phase Induction Motor	
2	Pole / Rated Output	2 Pole / 4,000 watts	
3	Power Source	1 Ph - 208/ 230 volt - 60 Hz	
4	Rated Revolution	3,500 rpm	
5	Insulation Class	B CLASS	
6	Winding Resistance (at 77℉)	Main	0.51 ±7% ohm
		Sub	0.84 ±7% ohm

Table 2.3: LG Compressor Performance [26]

1.4 Performance

	ARI	DOE B
Voltage	230V	230V
Cooling Capacity (95%↑)	51,000 Btu/h	69,500 Btu/h
	12,852 kcal/h	17,514 kcal/h
Power Input (105%↓)	4,722W	3,188W
E.E.R (95%↑)	10.8 Btu/W·h	21.8 Btu/W·h
Running Current(reference)	21.0 A	14.3 A
Locked Rotor Ampere	128 A	
MCC	34.7 A	
Noise	Max 75dB (A)	
Vibration	Max 50μm	

2.4 Blower

The blower in the system is responsible for the air flowing in the system. In a packaged unit, given constrains in the design, it is very difficult to maintain laminar flow. The airflow is measured in CFM, cubic feet per minute (cu. ft. /min.). For this unit, it has been tried to maintain an 1820 cfm on a rating A test. It corresponds to 364 cfm per ton on this 5 ton AC (Air Conditioner)

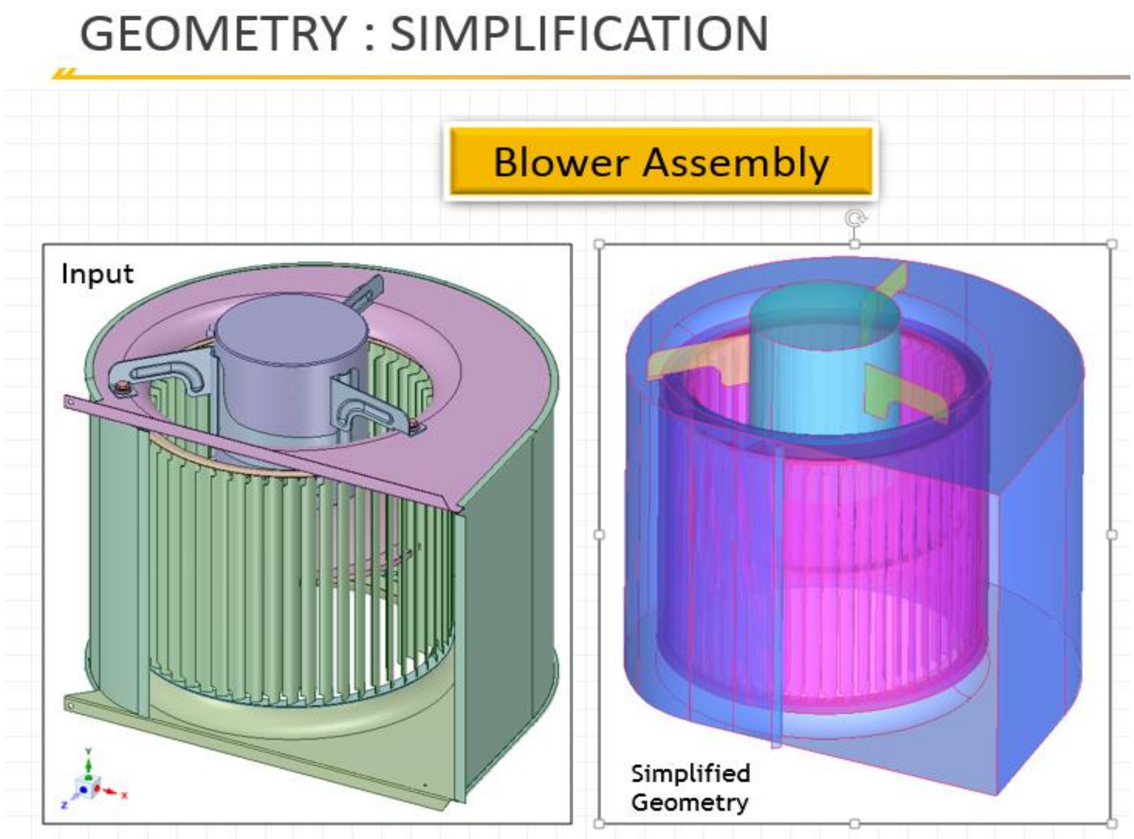


Figure 2.4 Blower Assembly Model

The blower assembly consists of three main parts, the blower housing, standard constant torque motor and blower wheel. A 1 horsepower blower was used to circulate air in the system and the changes in scroll sides and the venturi of the blower helps reduce watts in the system [20].

A comparison of the old blower design vs the new one blower listed above was performed.

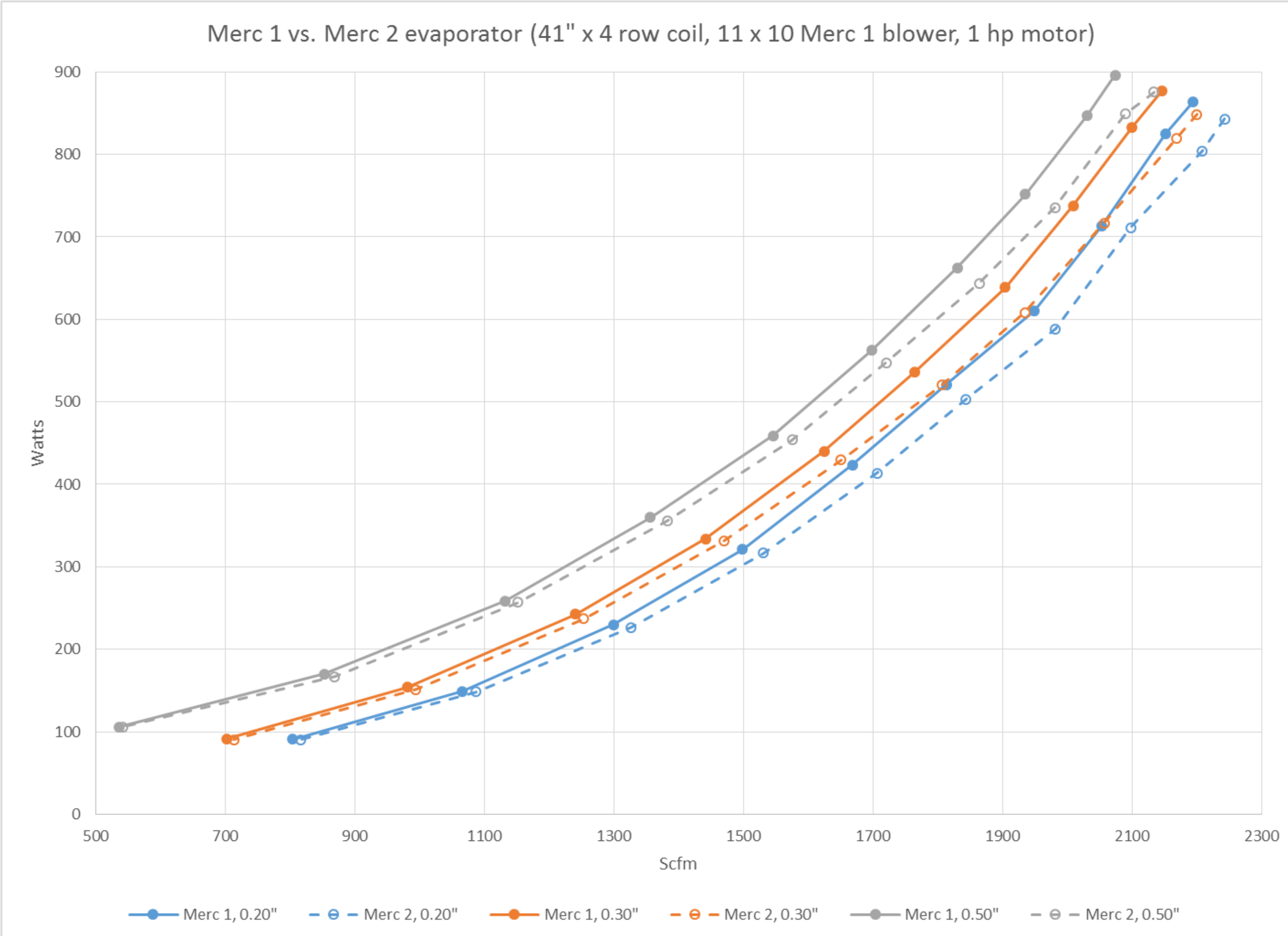


Figure 2.5 Old vs New Blower Comparison

For ease of differentiation, the old blower was named Merc 1 and the new blower, Merc 2. The graph shows a watts vs scfm (standard cubic feet per minute) curve. The watts in a system are non-linearly proportional. Both the blowers were operated under the same given conditions on multiple static points. The blue solid line and blue dotted line is a comparison. At 0.20 static for both the blowers, a big difference is not seen between the two blowers at lower scfm, but the watts are much lower with the Merc 2 blower at higher scfms. At higher static, a larger efficiency gain is seen when using a Merc 2 blower.

CHAPTER 3

UNIT CONSTRUCTION

3.1 3D Model

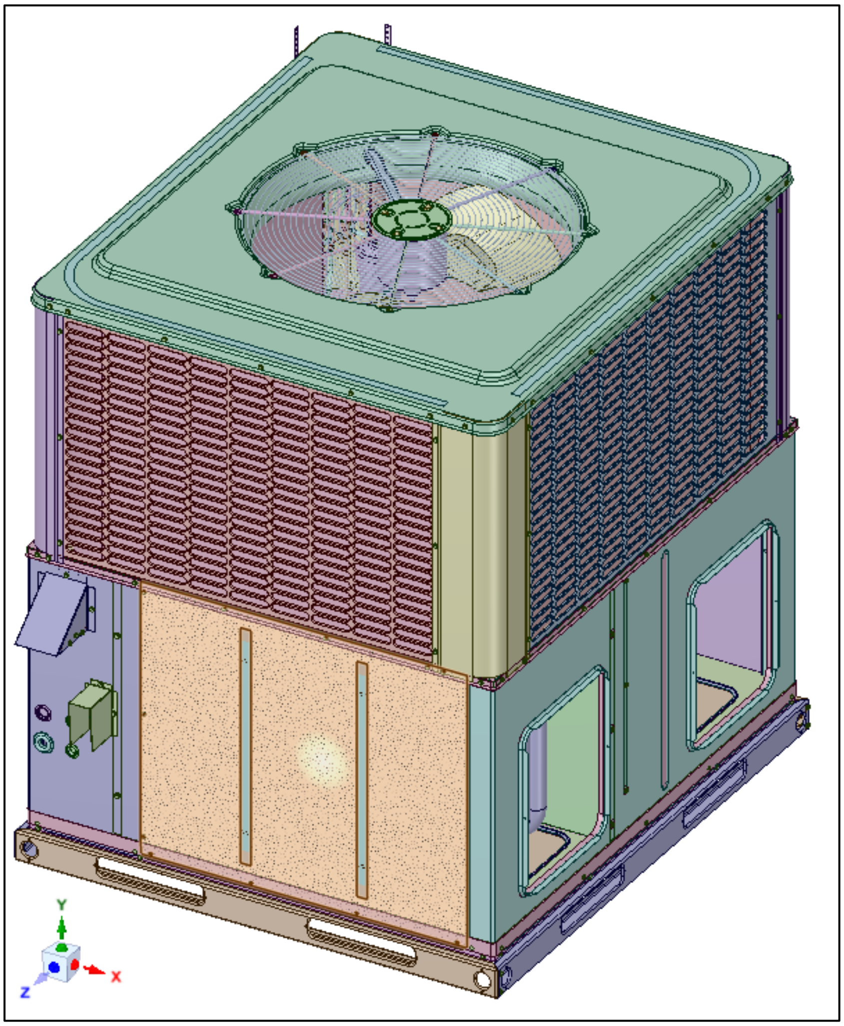


Figure 3.1 Final Assembly Isometric View

In the figure 3.1, a 5 ton, 60, 000 btu AC unit is displayed. The unit is divided into multiple assemblies. The bottom most piece is the base ban assembly that contains the base rails, insulation and the base pan, which holds the weight of the total system. The 1st level contains the indoor side, the duct panel that can be seen the picture and the blower compartment and compressor compartment (not shown). The top level contains the condenser coils and condenser fan.

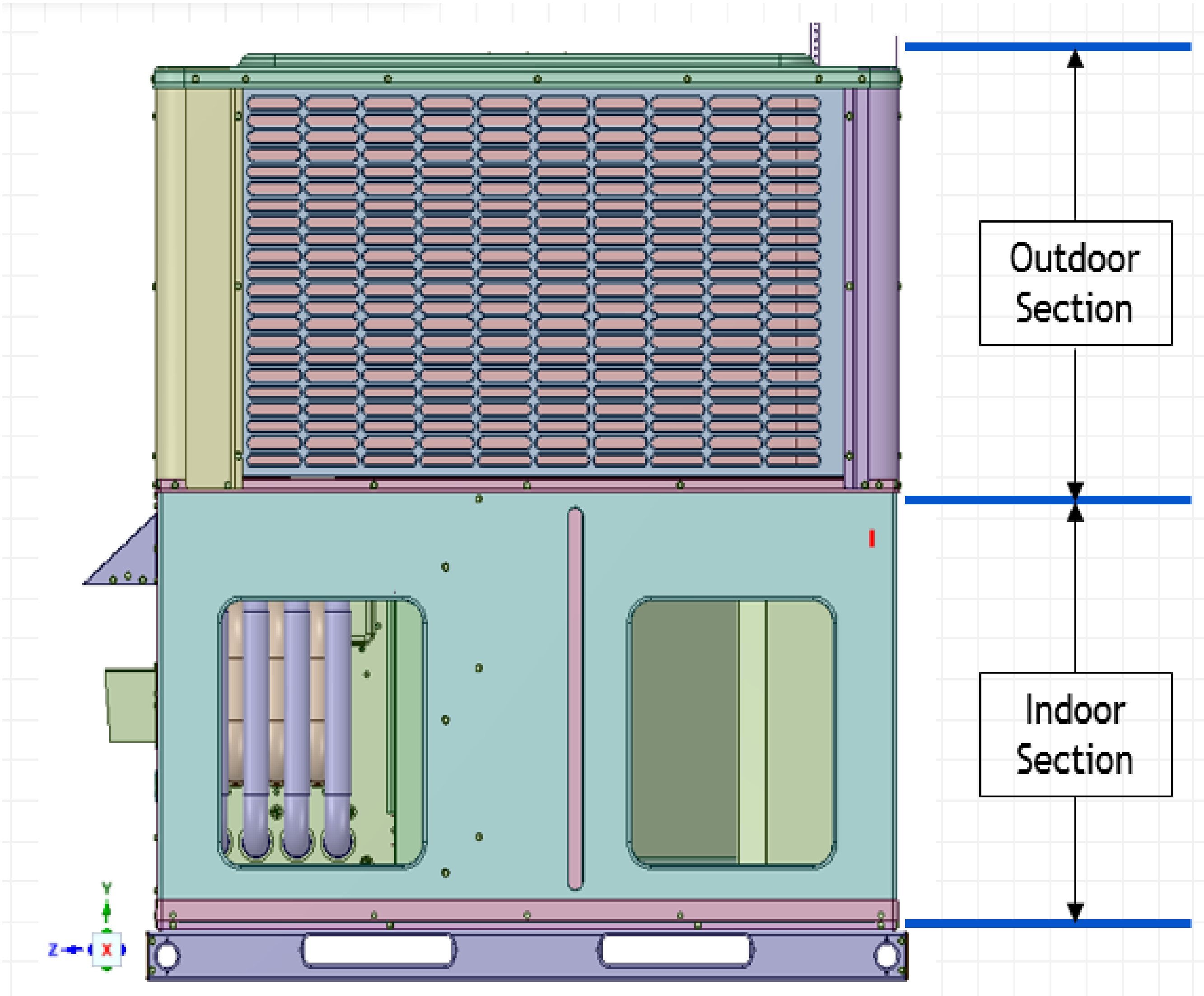


Figure 3.2 Final Assembly Back View

In the figure 3.2, the unit can be seen from the back view. The two duct openings are connected to the return air duct and supply air duct. Unconditioned air enters from the right panel, goes through the refrigeration system and cold air is dispensed from the left duct and is distributed to every of the space that is conditioned. The total length of the unit is 51.25 inches and width is

45.75 inches. Given the shortening of the condenser coils, the total height of the unit is now 50.2 inches.

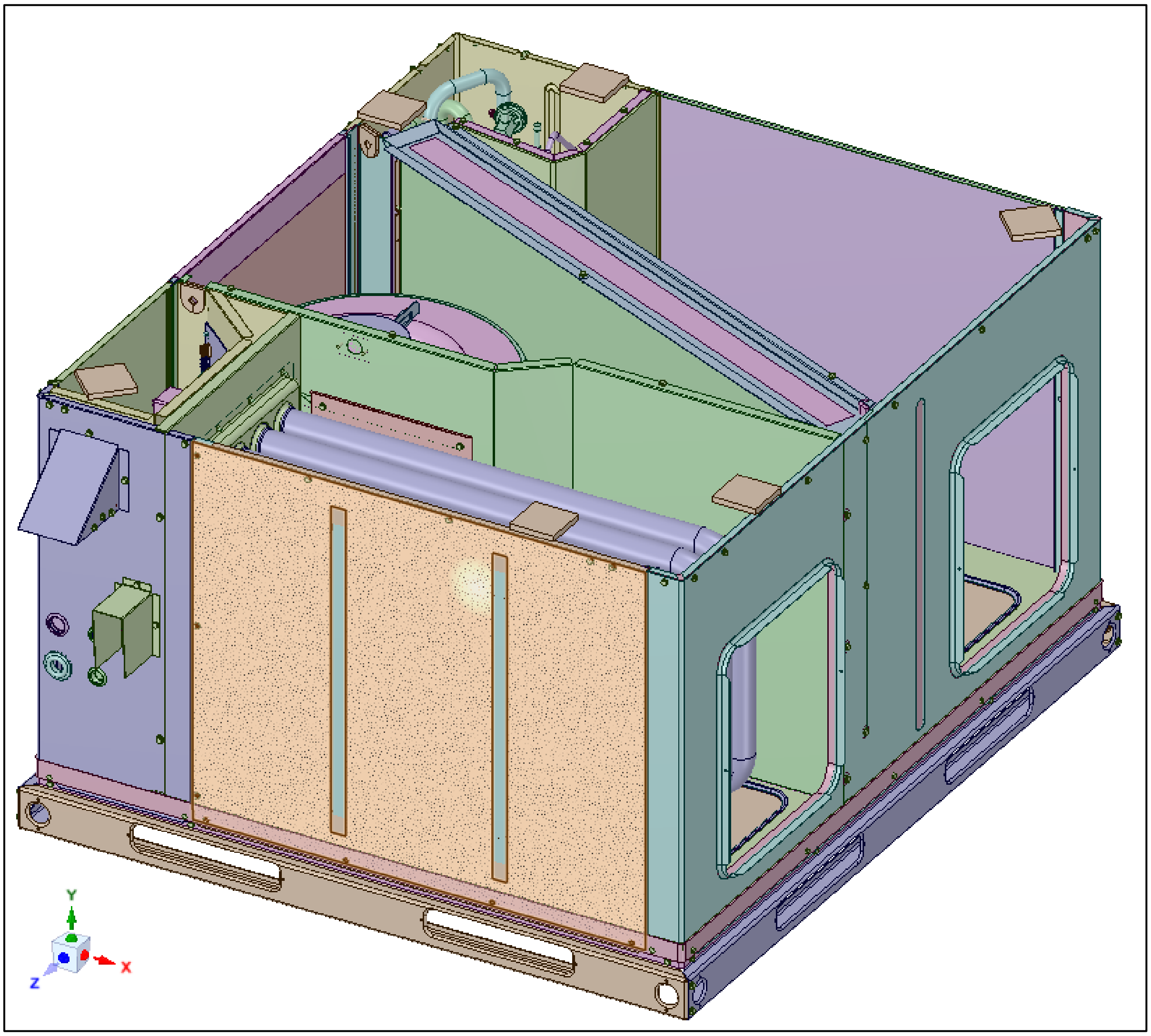


Figure 3.3 Base Unit Assembly

The base unit assembly which is 31 inches tall (figure 3.3) contains the indoor side of the packaged unit. The refrigeration system cools the tubes in the indoor coil and air is drawn through the return air duct, travels through the blower compartment and is dispensed out on the supply duct side. In the proposed model, a furnace can be seen, which may or may not be available. The effect of furnace is negligible and indifferent to refrigeration changes.

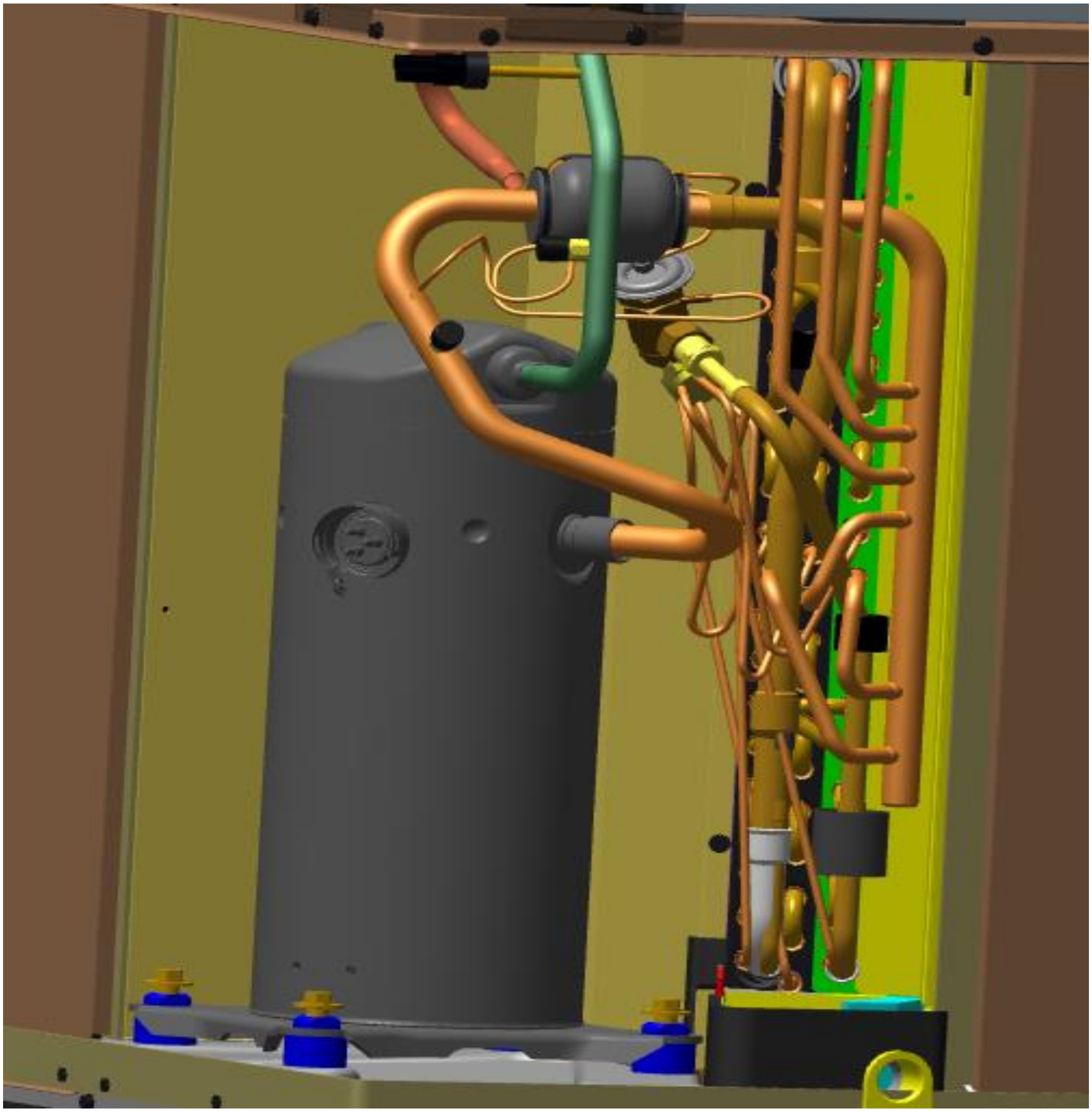


Figure 3.4 Old Design Compressor Compartment

In figure 3.4, an older design compressor compartment can be seen. It contains the copper indoor coil with copper headers, with a suction line connected to the compressor. A copper distributor connected to the thermal expansion valve can also be seen displayed in the figure.

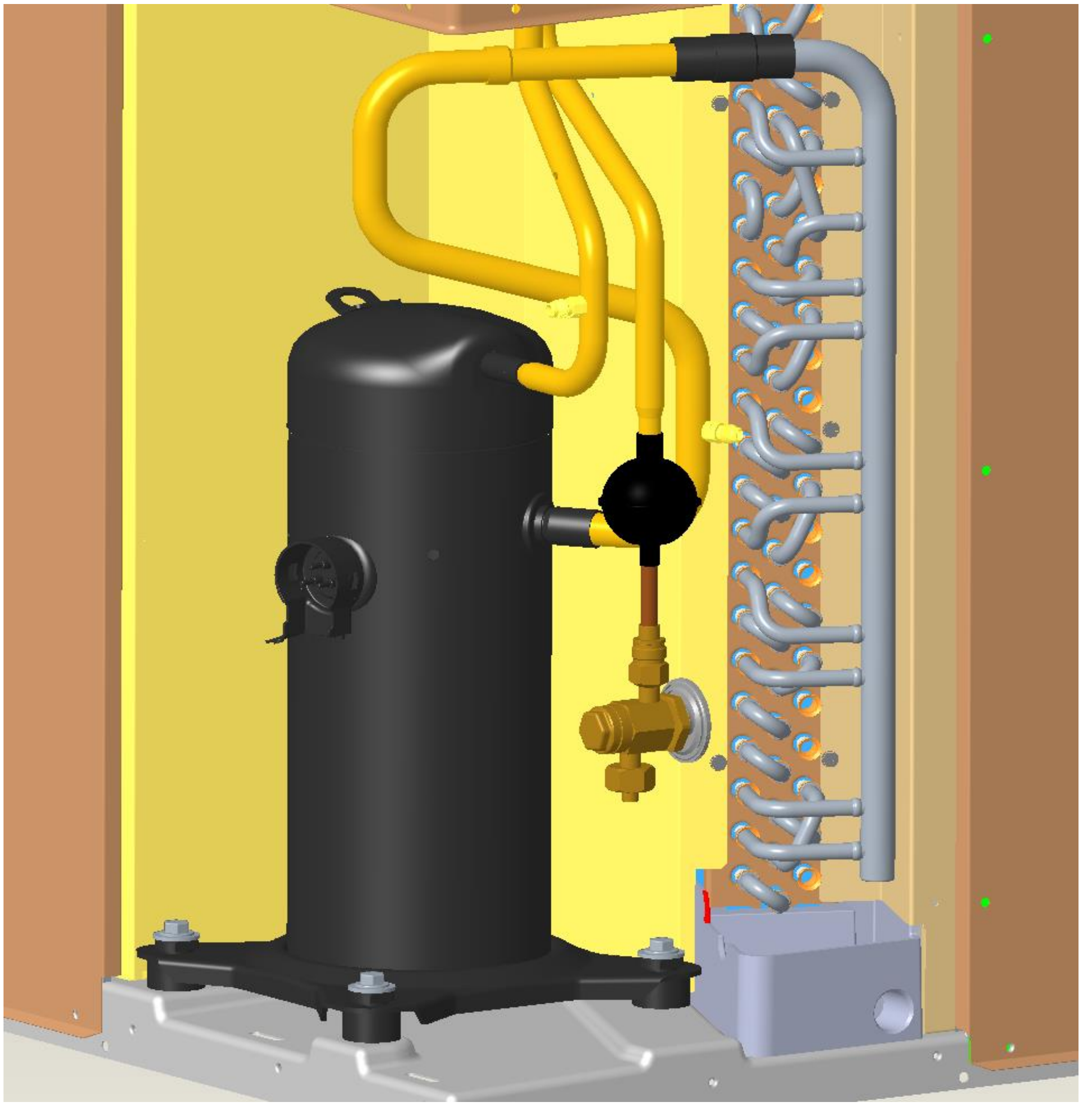


Figure 3.5 New Design Compressor Compartment

In the figure 3.5, a compressor compartment can be seen from the new design with copper coils. The evaporation coil now contains aluminum tubes instead of the copper tubes. The coil header is also aluminum, it uses a heat shrink to converge into copper which can then be brazed to the suction tube going to the compressor. The thermal expansion valve location has moved to a more serviceable location and the copper tubing is now simplified. The distributor can't be viewed in this model as it hides the coil circuiting.

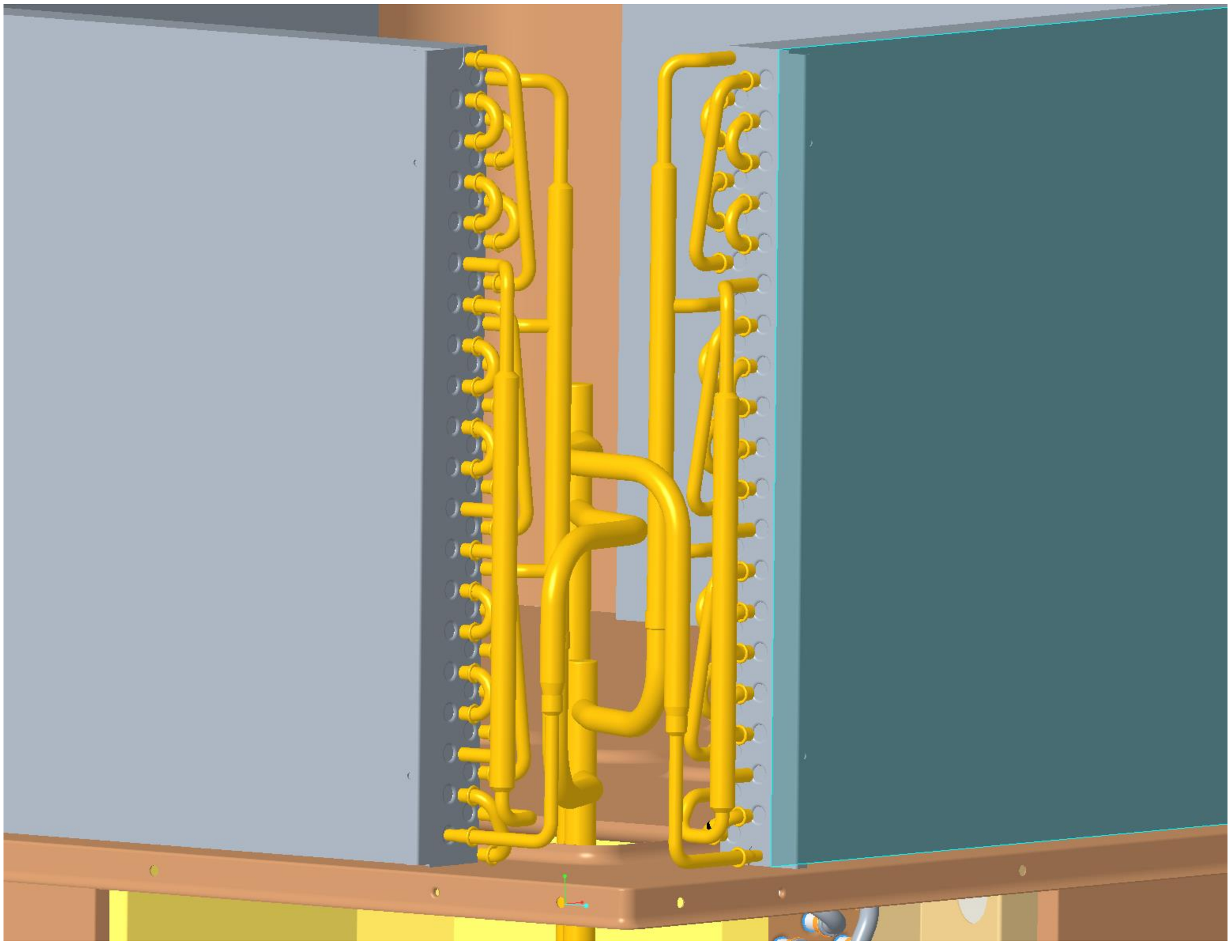


Figure 3.6 New Design Condenser Coil

The figure 3.6 shows the condenser coil circuiting. The inside of the coil is connected to the discharge line which brings in hot vapor into the coil, the vapor travels through the coil and is cooled and liquefied as it exits the coil on the outside row and travels to the expansion valve.

3.2 Drawings

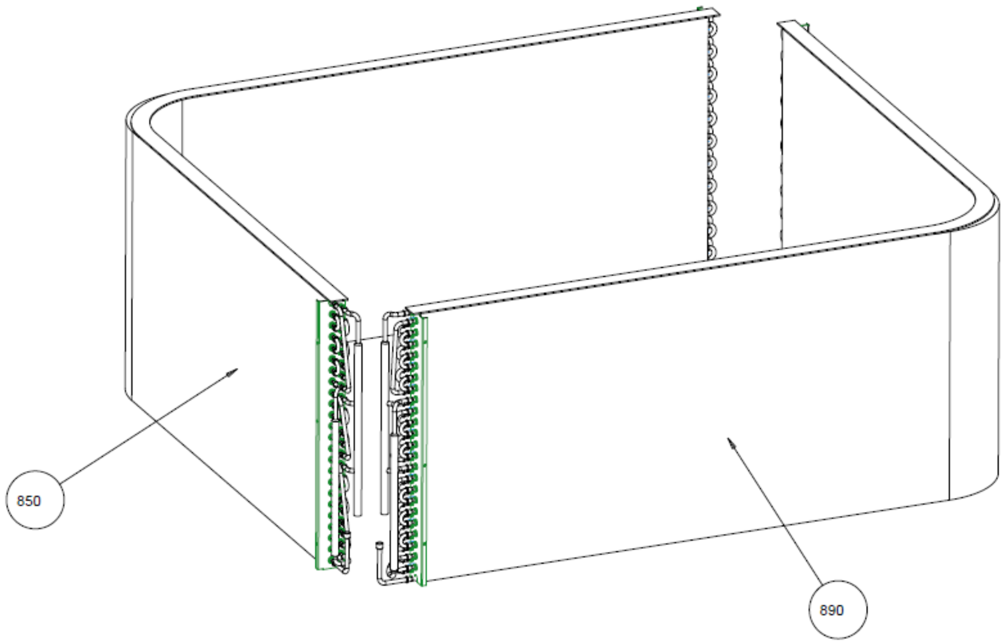


Figure 3.7 Outdoor Coil Drawing

Figure 3.7 shows the outdoor coil divided into two pieces. Since space is constrained in a package unit, two row coils are used on the condenser side. The coil slabs are 79 inches long and 19.2 inches tall with 24 tubes and 12 hairpins.

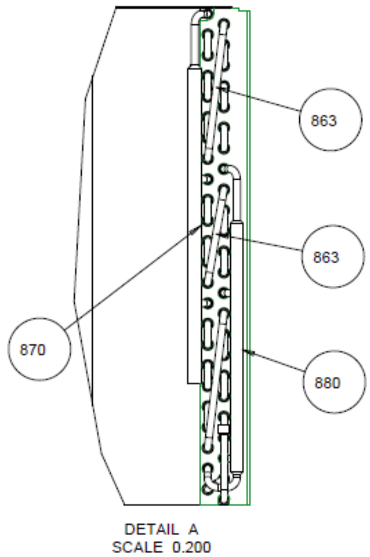


Figure 3.8 Outdoor Coil Circuiting

The schematic in figure 3.8 depicts the right coil slab circuit. The refrigerant enters the coil from the inside discharge line and exits towards the liquid line on the right. The liquid is again sent back to the coil to cool further, it helps us get the liquid temperature to under 100 degrees Fahrenheit.

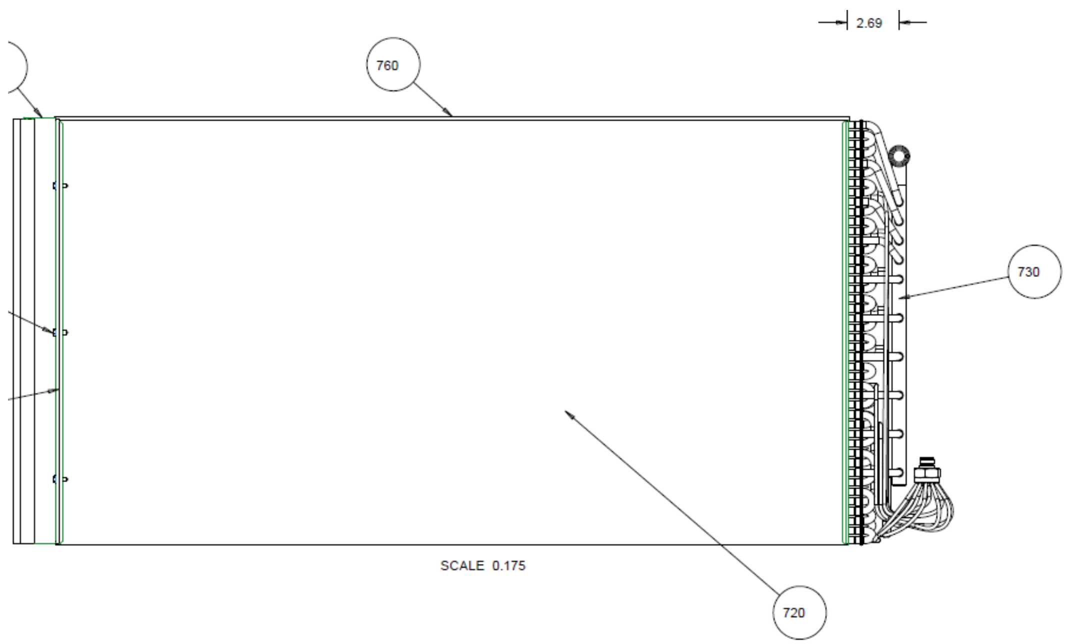


Figure 3.9 Indoor Coil Drawing

The evaporator coil is 41 inches long and contains 4 rows. The left side view is displayed in figure 3.9.

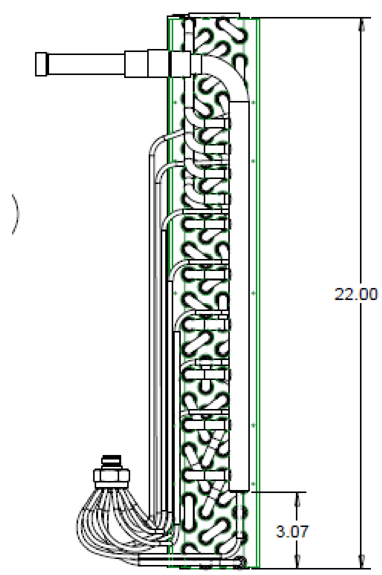


Figure 3.10 Indoor Coil Circuiting

The front view of the coil (figure 3.10) shows us the indoor coil circuiting, we can see 10 circuit header and distributor, the liquid is dispensed into the coil through the distributor, the heat is transferred to the airstream through the fins and the header takes the hot vapor to the compressor.

CHAPTER-4

TEST RESULTS

The test results comprise of 3 different parts, psychrometric test results, CFD, and vibration analysis on copper tubing.

The CFD analysis was the first step in the testing process. The airflow was measured in the system and the indoor circuits were designed accordingly. Later, the coil circuitry is modified given the temperatures acquired in a test.

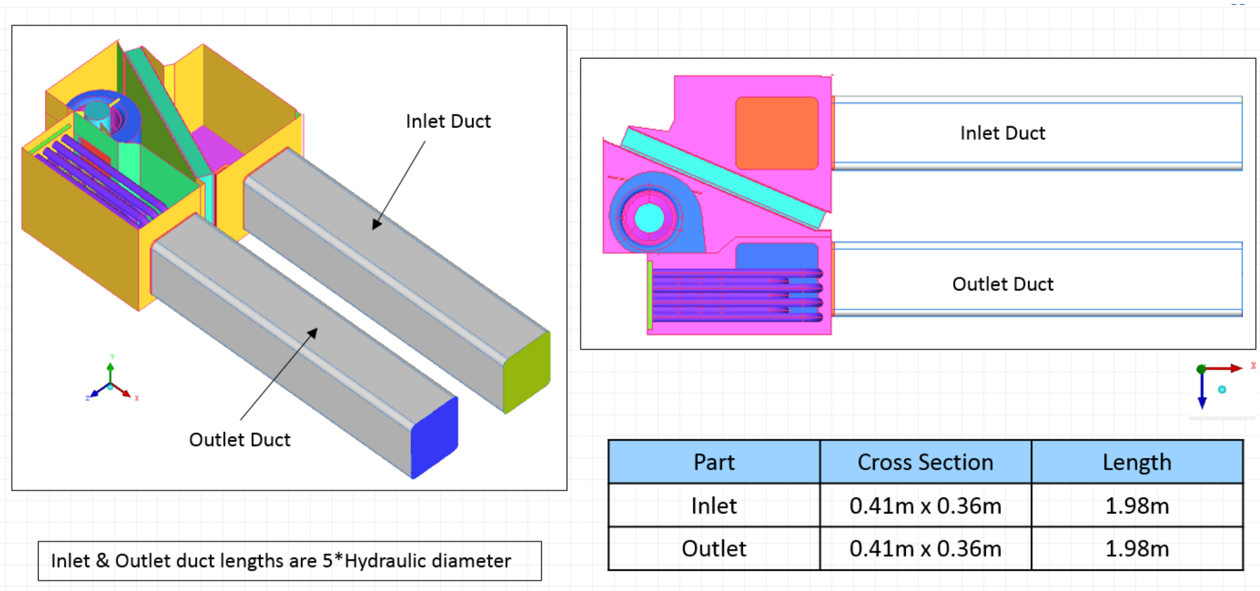
The psychrometric test results provide performance numbers. There are two conditional tests; A_{full}, and B_{full} rating tests. One cyclic test which is required to calculate the seasonal energy efficiency ratio (SEER) [15].

Copper tubing is designed keeping space constrains in mind. The accessibility to components is essential. These units are serviced regularly as easy to work units are preferred by the servicemen. Simulations were performed in Ansys™ and then the tubing was measured using a strain gage to measure the actual values.

4.1 CFD

'Flow-only' simulations are carried out for different indoor & outdoor units under given operating scenario. The scope of present analysis is to perform steady-state, flow analysis of PCE4B60, 5 ton AC package unit with side inlet and side exit configuration to achieve the flow rate of 1800 ft^3/min.

The domain considered in this particular simulation is indoor section of PHE6 with blower speed of 1150 rpm to evaluate the air flow rate, velocity distribution over heat exchanger, pressure drop across system and also to study overall flow system characteristics (figure 4.1).



Part	Cross Section	Length
Inlet	0.41m x 0.36m	1.98m
Outlet	0.41m x 0.36m	1.98m

Figure 4.1 Unit Geometry

- Outlet duct exit is at 0.69.67 pascal static pressure
- Flow is considered as an incompressible, steady state and turbulent
- All walls are considered as no slip walls and heat transfer across the cabinet is not simulated (adiabatic walls)
- Flow inside the coil is laminar

Material/Fluid	**Density (kg/m³)**	**Dynamic viscosity (Ns/m²)**
Air @at 26.6°C	1.17832	18.467×10^{-6}

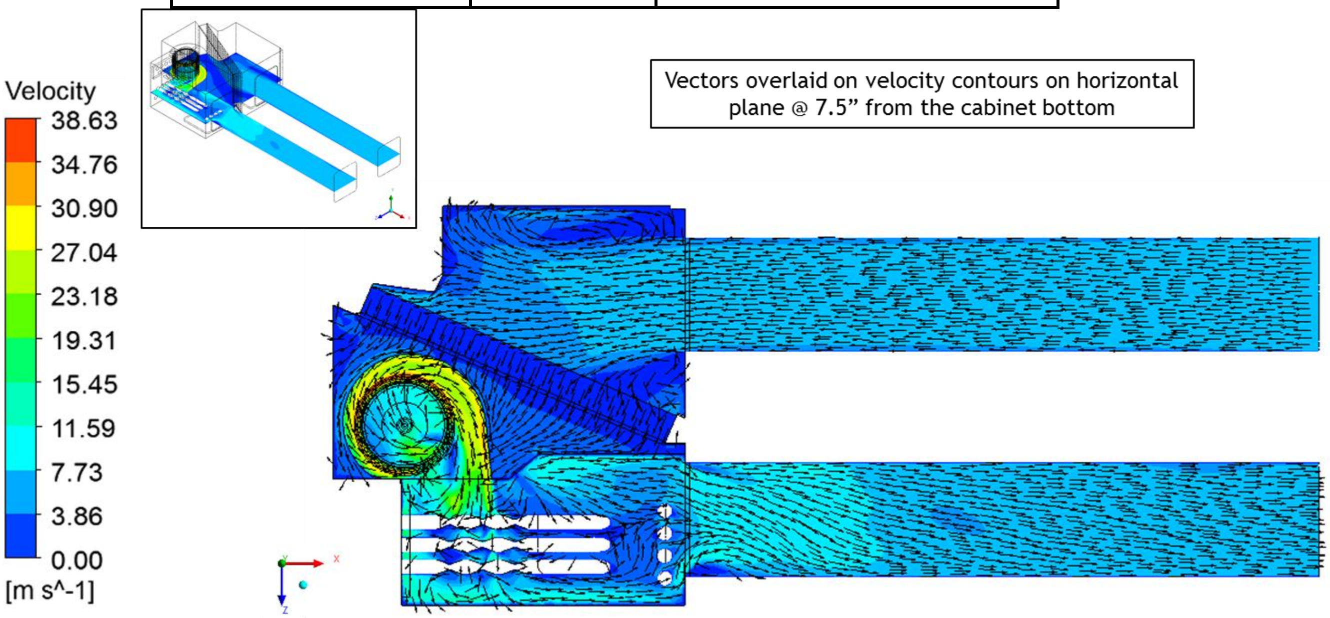


Figure 4.2 Unit Geometry 2

In the CFD model in figure 4.2, the blower can be seen pulling air at approximately 5m/s, and it loses its speed as it gets into the evaporator coil. The fins transfer heat and the 1 horsepower blower motor pushes air at approximately 29 m/s to the outlet duct. Given the geometry of the system, restrictions are seen, like the heat exchanger tubes and the small size of the compartment and 'leaving air' is close to 10 m/s which in most cases is sufficient to supply cold air to a space.

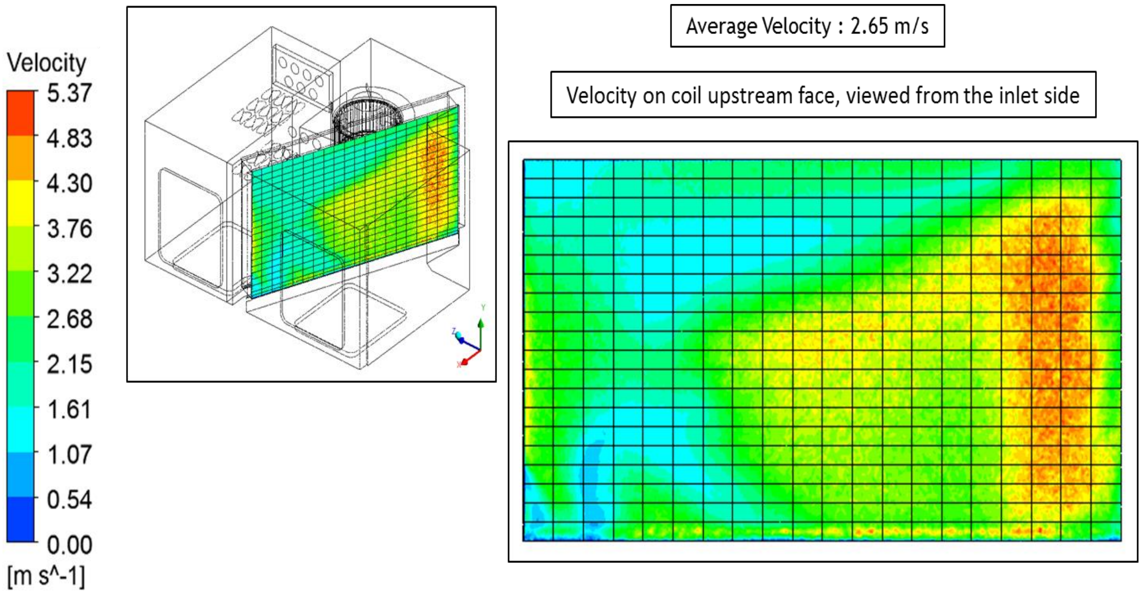


Figure 4.3 Velocity Contours: Coil Upstream

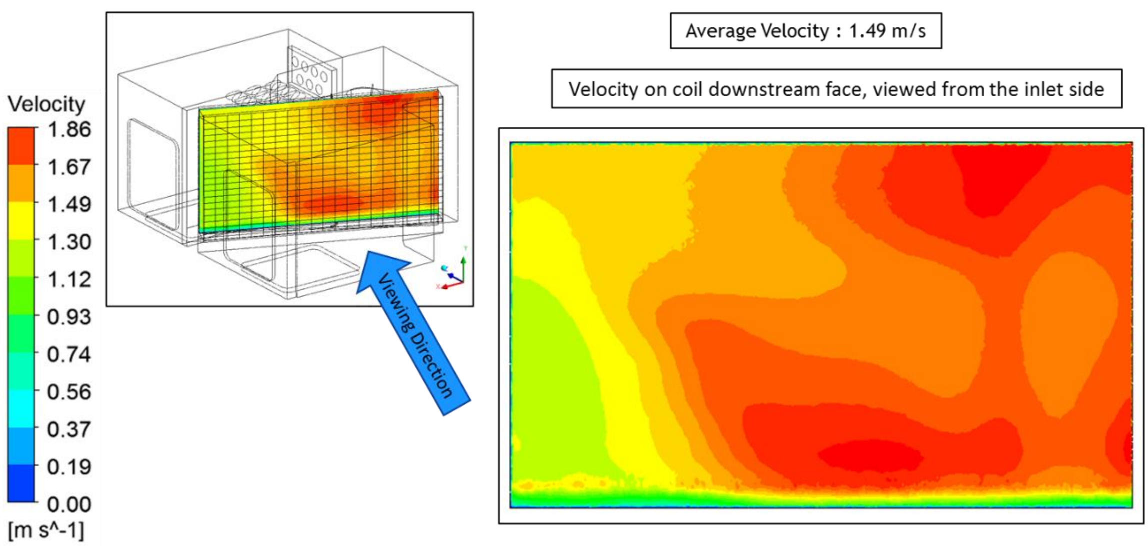


Figure 4.4 Velocity Contours: Coil Downstream

In the figures 4.3 and 4.4, the orientation of the coil can be seen to be at 45^0 angle to the airflow, and most of the air is concentrated at the front part of the coil, thus not utilizing the complete face of the coil.

4.2 Psychrometric Test Results

Table 4.1 A-test results

	Average	Max	Min	Range	Std Dev.
Indoor Air Info					
Entering Dry Bulb Temp (°F)	80.01	80.04	79.98	0.06	0.01
Entering Wet Bulb Temp (°F)	66.97	67.05	66.89	0.16	0.03
Leaving Dry Bulb Temp (°F)	57.41	57.45	57.37	0.09	0.02
Leaving Wet Bulb Temp (°F)	56.48	56.57	56.47	0.10	0.02
Dry Bulb Delta T (°F)	22.60	22.66	22.55	0.11	0.02
Wet Bulb Delta T (°F)	10.49	10.60	10.42	0.18	0.01
Entering Grid Temp (°F)	80.13	80.20	80.05	0.14	0.02
Leaving Grid Temp (°F)	58.42	58.51	58.34	0.17	0.02
Static Pressure (inH2O)	0.28	0.29	0.27	0.02	0.00
Airflow (SCFM)	1802	1815	1794	21	1.5
Outdoor Air Info					
Entering Dry Bulb Temp (°F)	95.01	95.07	94.97	0.10	0.01
Entering Wet Bulb Temp (°F)	71.50	71.57	71.45	0.12	0.02
Refrigerant Info					
Compressor Disch Press (psig)	396.21	397.46	394.71	2.75	0.17
Outdoor Liq SC (°F)	9.25	9.42	9.07	0.35	0.02
Indoor Liq SC (°F)	9.78	10.00	9.54	0.45	0.06
Indoor Suct SH (°F)	11.62	12.31	11.22	1.08	0.07
Compressor Suct SH (°F)	9.9	10.7	9.5	1.2	0.1
Capacity / Efficiency					
ID Capacity (Btu / hr)	61,008	61,590	60,478	1,112	86
ID EER/COP (Btu / W-hr)	13.02	13.17	12.91	0.26	0.02

Refer to table A1 in the appendix for details

Given the ‘A test’ results, the focus was on some key points like the 1802 cfm, as it meets the requirements of over 350 cfm per ton. Given the reliability aspect, the focus was on the compressor discharge pressure to be under 400 psi and it is seen that the average pressure was 396.21 psi.

The sub-cool in the system is over 9^0 Fahrenheit which shows a good amount of liquid leaving the condenser coil. Superheat at the evaporator and compressor is over 9^0 Fahrenheit which means that a significant amount of superheat in both locations is observed.

The capacity target of 60,000 btu/hr has been exceeded by over a 1000 btu/hr given the performance of the compressor was rated over 60,000 btu/hr. The energy efficiency ratio is over 13, whereas the requirement is only 11.

Table 4.2 Old vs new unit comparison

	Old Design	New Design
Indoor Air Info		
Entering Dry Bulb Temp (°F)	80.00	80.01
Entering Wet Bulb Temp (°F)	67.04	66.97
Leaving Dry Bulb Temp (°F)	58.18	57.41
Leaving Wet Bulb Temp (°F)	56.50	56.48
Dry Bulb Delta T (°F)	21.82	22.60
Wet Bulb Delta T (°F)	10.54	10.49
Entering Grid Temp (°F)	79.48	80.13
Leaving Grid Temp (°F)	57.62	58.42
Static Pressure (inH2O)	0.28	0.28
Airflow (SCFM)	1717	1802
Outdoor Air Info		
Entering Dry Bulb Temp (°F)	95.00	95.01
Entering Wet Bulb Temp (°F)	71.52	71.50
Refrigerant Info		
Compressor Disch Press (psig)	390.7	396.2
Outdoor Liq SC (°F)	11.1	9.3
Indoor Liq SC (°F)	11.9	9.8
Indoor Suct SH (°F)	9.1	11.6
Compressor Suct SH (°F)	12.0	9.9
Capacity / Efficiency		
ID Capacity (Btu / hr)	58297	61008
ID EER/COP (Btu / W-hr)	12.6	13.0

Refer to table A2 in the appendix for details

The table 4.2 compares the old test data with the new test data. A higher capacity and better EER can be seen, even while seeing a cost reduction. These improvements are due to a higher air flow, robust circuiting and a stable sub-cool and superheat.

Table 4.3 B test results

	Average	Max	Min	Range	Std Dev.
Indoor Air Info					
Entering Dry Bulb Temp (°F)	80.01	80.06	79.96	0.10	0.03
Entering Wet Bulb Temp (°F)	67.00	67.06	66.94	0.13	0.02
Leaving Dry Bulb Temp (°F)	56.37	56.42	56.34	0.08	0.01
Leaving Wet Bulb Temp (°F)	55.57	55.65	55.50	0.15	0.02
Dry Bulb Delta T (°F)	23.64	23.68	23.57	0.11	0.02
Wet Bulb Delta T (°F)	11.43	11.51	11.35	0.16	0.01
Entering Grid Temp (°F)	80.15	80.23	80.06	0.18	0.03
Leaving Grid Temp (°F)	57.44	57.53	57.36	0.17	0.02
Static Pressure (inH2O)	0.28	0.30	0.27	0.03	0.00
Airflow (SCFM)	1804	1816	1793	23	2.3
Outdoor Air Info					
Entering Dry Bulb Temp (°F)	81.94	82.04	81.87	0.16	0.04
Entering Wet Bulb Temp (°F)	65.01	65.11	64.92	0.19	0.05
Refrigerant Info					
Compressor Disch Press (psig)	337.59	338.89	336.64	2.25	0.27
Outdoor Liq SC (°F)	9.19	9.41	9.01	0.40	0.02
Indoor Liq SC (°F)	9.75	10.03	9.53	0.49	0.02
Indoor Suct SH (°F)	12.40	13.00	11.65	1.35	0.07
Compressor Suct SH (°F)	11.1	11.7	10.2	1.5	0.1
Capacity / Efficiency					
ID Capacity (Btu / hr)	66,111	66,524	65,599	925	43
ID EER/COP (Btu / W-hr)	15.96	16.09	15.85	0.23	0.02

Refer to table A3 in the appendix for details

The ‘B test’ is very similar to the ‘A test’ given the conditions and yields a higher capacity and Energy efficiency ratio. The ‘B test’ EER is then used to calculate the SEER value.

Cyclic Test Results

SEER Calculation

Seasonal Efficiency Calculations. Seasonal efficiency descriptors, SEER, HSPF (heating seasonal performance factor), shall be calculated per the equations in this section, using the results from the individual test calculations. Throughout the seasonal efficiency calculations wherever the values 95, 82, 67, 62, 47, 35, and 17 °F are used, they are derived from the outdoor dry bulb temperatures, °F, at test conditions A, B, F, H0, H1, H2, and H3, respectively [21].

11.2.1 SEER.

11.2.1.1 *Single Stage System*. SEER for a Single Stage System shall be calculated as follows.

SEER = PLF (0.5) · EERB, Full 4.1

Where:

PLF (0.5) = 1 – 0.5 · $CDc, Full$ 4.2

$CDc, Full$= {1−$EERD, FullEERC, Full$} 1−$CLFcyc, Full$ 4.3

$CLFcyc, Full$=$q'cyc, D, Full$ ($qC, Full\cdot\theta cyc$) 4.4

Given this standard equation, the unit SEER was calculated to be 14.89.

Table 4.4 Cyclic test results

441 – Cd	0.13	0.11	0.12
441 – SEER	14.89	15.05	15.03
441 – Part Load	0.93	0.94	0.94
441 – B-EER	15.96	15.96	15.96

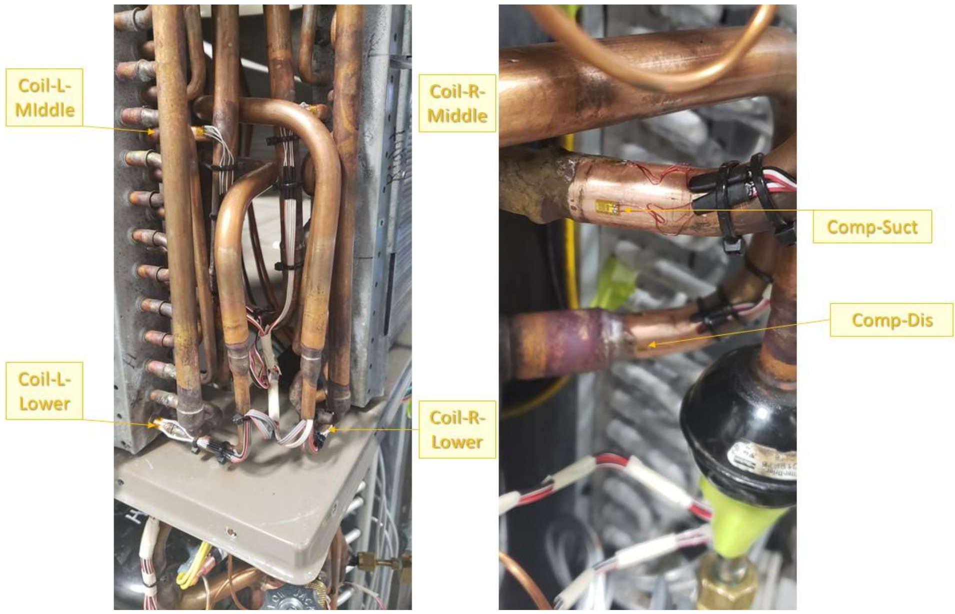
Refer to table A4 in the appendix for details

4.3 Vibration Analysis

Strain gauges were mounted on all the copper tubes to measure the strain of the system. Since the strain is linearly related to stress, the values are converted to psi using the material properties. The passing criteria for vibration analysis was to have stress under 5000 psi.

The first data set was conducted and high stress in tubes were observed in upwards of 9000 psi.

A mass on the copper tube was placed and vibration was achieved by external means. Though the stresses dropped and the mass dampened the vibration, the eventual goal is not to use mass in the system as it can drop off over time.



(a) **(b)**

Figure 4.5 Strain Gauges Condenser

Figure 4.5(a) shows the outdoor condenser coil and figure 4.5(b) shows the indoor section of the package unit. These pictures reflect the drawings from chapter 3.2. The unit was strain gauged to take strain readings.

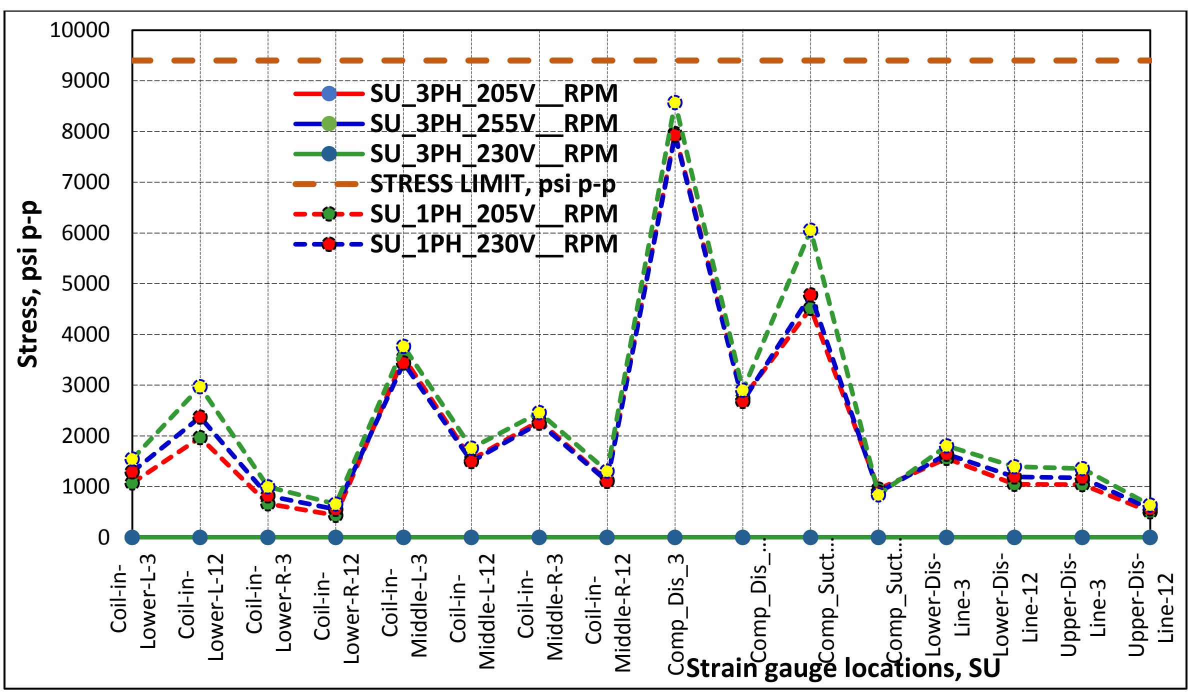


Figure 4.6 Stock Copper Tubing

The chart I figure 4.6 shows high vibration in tubes, in some cases the geometry is not sufficient for a decent reading. The highest stress can be located in the discharge line. We add a mass on the discharge line to reduce the vibration.

Mass

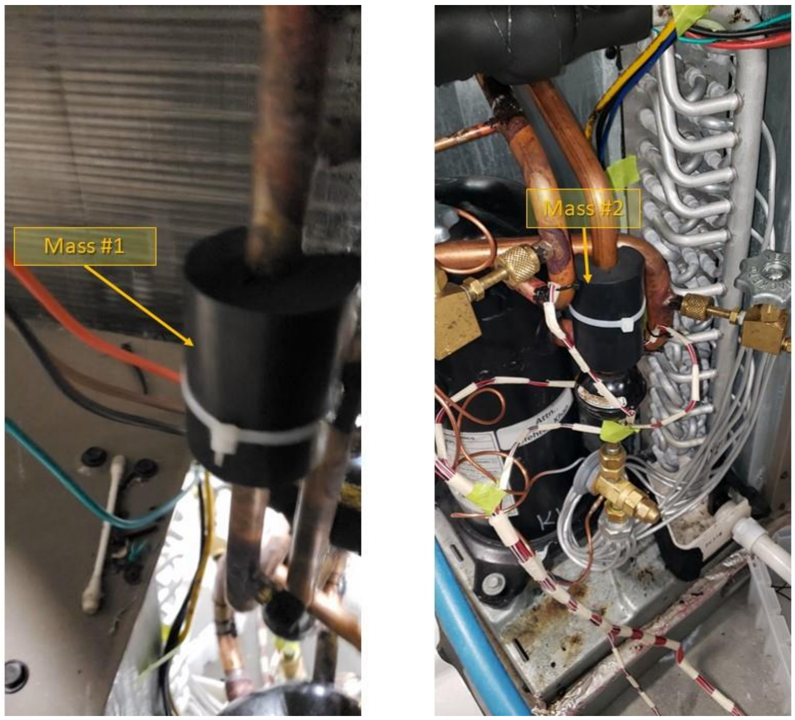


Figure 4.7 Strain Gauges Evaporator with mass

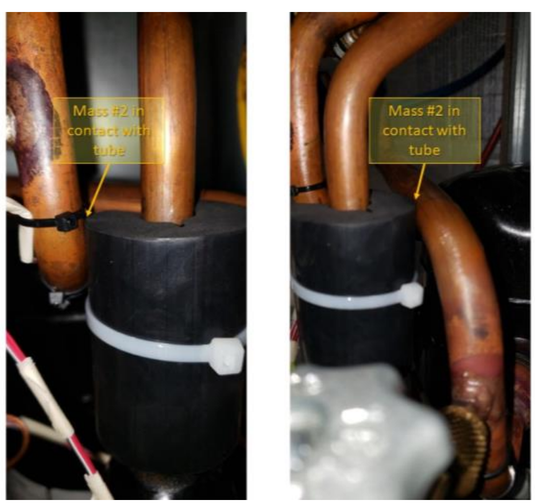


Figure 4.8 Mass addition

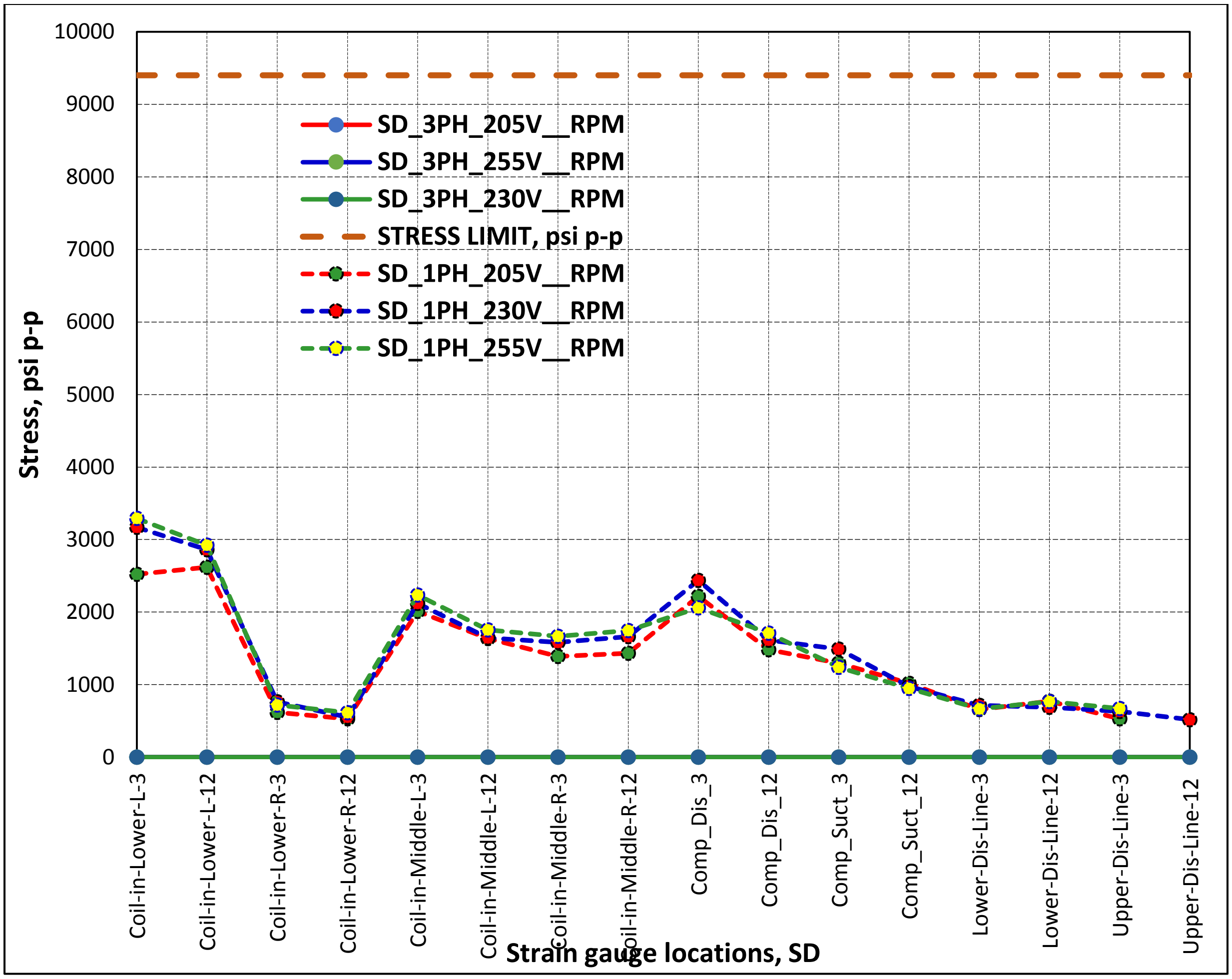


Figure 4.9 Addition of mass to copper tubing

Given the success of the mass on the copper tubing, it was decided to tweak the copper tubing and move the coil location. That introduced a vibration in the condenser coil tubing. A zip tie was added to the right and left side coil to dampen the vibration. It dropped the stress to under 4000 psi and moved failure rate to well below 5000 psi.

Factor of Safety (FOS) = (ultimate stress/ actual stress)

FOS = 5000/3200

FOS = 1.56

Figure 4.10 Zip tie on condenser coil

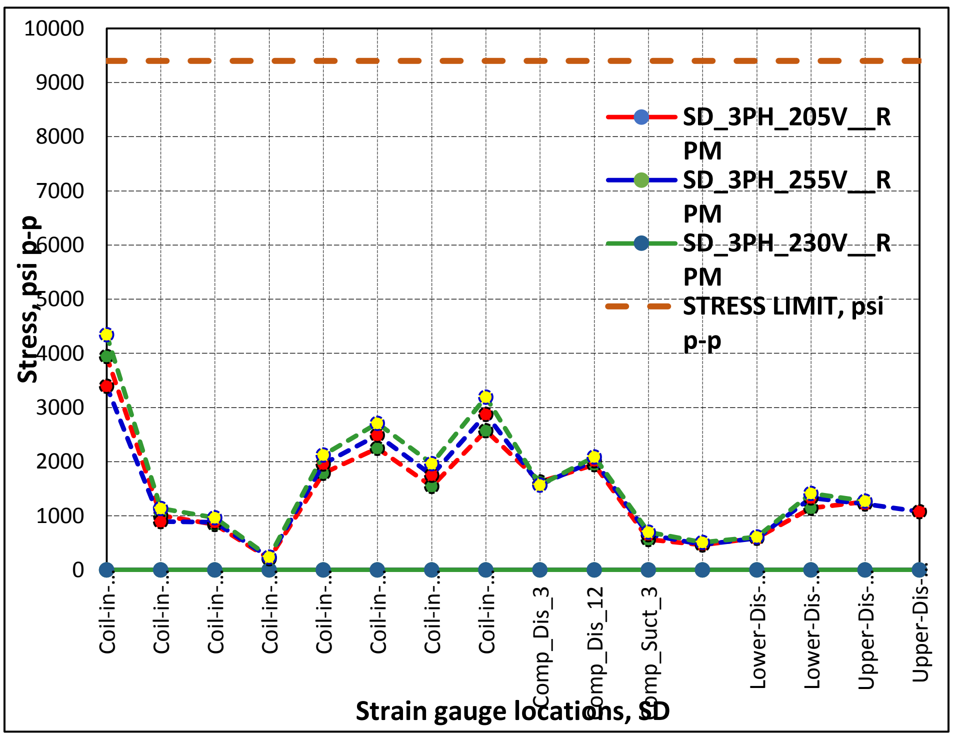


Figure 4.11 Addition of zip tie to copper tubing

Addition of the zip tie is cheap and effective method to reduce vibration in the condenser coil. The factor of safety using a zip tie:

Factor of Safety (FOS) = (ultimate stress/ actual stress)

FOS = 5000/4300

FOS = 1.16

Given the vibration testing results, either masses can be used on the discharge tube or a zip tie at the condenser coil can be used. It was decided to use a zip tie as it is cheaper and more effective. The zip tie was thought to be easier to handle as well. Given the vibration standards set, a factor of safety over 1.10 is acceptable. For both these solutions, a factor of safety over 1.10 was achieved and therefore both options are viable. Vibration in the system does not interfere with system performance but plays a pivotal role in the reliability of the copper tubes and the system a whole.

CHAPTER 5

COSTING

5.1 Cost Saving

Table 5.1 Costing

Old Components	**Cost/$**	**New Components**	**Cost/$**		
Compressor	$235.96	Compressor	$205.30		
ID Coil	$200.41	ID Coil	$90.85		
OD Coil	$256.38	OD Coil	$163.18		
ID Header	$20.08	ID Header	$6.08		
ID Distributor	$7.51	ID Distributor	$8.32		
OD Fan Blade	$12.48	OD Fan	$12.48		
OD Motor	$30.73	OD Motor	$30.73		
OD Header		OD Header			
Capacitor	$4.17	Capacitor	$4.17		
Total	$767.72	Total	$521.11	$256.11	Total Saving

Given the changes in the system, it was noticed that the cost of indoor coil drastically drops. The switch from tin coated copper to aluminum tube signifies the $109.56 cost saving. The effect of outdoor coil height reduction and the use of 9/32 inch tubing warrants a saving of over $90. Given the coil height decrease, a cost savings is seen in parts like the header and compressor.

Table 5.2 Indoor coil assembly BOM (bill of materials)

Description		Cost	Currency
EVAPORATOR SLAB COIL ASSY – CIRCUITED 4 ROW 41			
SUB ASSEMBLY			
SLAB COIL ASSY – EXPANDED 4 ROW 41			
LACING TABLE			
ALUMINUM END SHEET – REAR 4 ROW			

Table 5.2 (continued)

ALUMINUM END SHEET – FRONT 4 ROW			
	110T RF PRESS		
	0ZAX-063-04274,COILT=1.600W=108.6 ALUH32		
	Fab. OH Such.		
COPPER TUBE – HAIRPIN,43.688 TIN COATED		142.240	USD
ALUMINUM TUBE – HAIRPIN,43.688		44.000	USD
FIN-SINE WAVE 4R-22H-16FPI		44.210	USD
FIN-SINE WAVE 4R-22H-16FPI		32.893	USD
	FIN PRESS – ALUM	0.26	USD
	ALUMINUM FIN .0045 X 31.875 7072-O	43.16	USD
	Fab. OH Such.	0.79	USD
EXPANDER – BURR OAK			USD
Fab. OH Such.			USD
RETURN BEND 3/8ODX1.00BC W/R			USD
Fab. OH Such.			USD

The indoor coil assembly BOM reflects the parts used to produce the current assembly. For aluminum, the copper tube was replaced with aluminum tube and sine wave fin to lanced fin. The total cost saving in the evaporator given these changes is approximately $110

CHAPTER-6

DISCUSSION AND CONCLUSIONS

In this study, the beneficial use of aluminum evaporator coils in an air-conditioning unit has been proven beyond a shadow of doubt. Capacity is lost during the heat transfer process of moving from tin coated copper tubes to aluminum tubes, but using a stacked circuit and better surface area fins results is significant gains. The size of the condenser coil was reduced but at the same time the number of tubes were not decreased, and the sub cool loop circuit makes the refrigerant leaving the coil cooler, thus helping to increase the overall capacity. The outdoor fan blade and motor, makes sure that the air effectively goes through the coil and cools the refrigerant.

The drop in compressor (from LG®) adds to the performance of the system, thus resulting in a higher capacity. The blower with a better venture and modified scroll sides make for better airflow through the system. The lower watts and different torque settings on the standard ECM (electronically commutated motor) indoor motor make the system more efficient and yield a better energy efficiency ratio (EER) which in totality helps in maintaining a SEER value of over 14.

The copper tubing was made simpler and easy to manufacture, so that the total yearly consumption of over 5000 units sold are less labor intensive do not take up a huge amount of time to build.

CHAPTER-7

RECOMMENDATIONS AND FUTURE WORK

This research work has resulted in significant data generated that will help future engineers design, more-efficient and cost-effective air systems for 60,000 Btu and other grades of input packaged refrigeration systems. However, there are potential future work that can further enhance these systems. Some of the recommendations are given below.

- This design robust and more compact and could be sold for a higher price
- The smaller size can provide as an advantage to costumers
- Higher capacity can be good selling point in hotter areas
- This unit can be used to study galvanic corrosion
- The effectiveness and reliability of new components can be proved with this unit
- Try using a 2 stage compressor for better efficiency
- Look into using thinner sheet metal to make the unit lighter

Future Work

- Run a transit test on the system for transportation reliability
- Run a rain test to make sure it is IPX 4 compliant
- Field test unit for it to be exposed to all temperatures
- Look into alternates of zip ties to hold coils better (brazing tubes together)
- Given time, copper tubing can be modified to reduce vibration
- Given simulations, Aluminum can be used in the complete unit
- Try reducing coil size even more to reduce cost
- Look into changing coil angle to the airflow for better face velocity
- Use full ECM indoor motor for higher efficiency
- Test furnaces with the new indoor coil and blower

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APPENDIX

APPENDIX

Table A1

	Average	Max	Min	Range	Std Dev.
Indoor Air Info					
Entering Dry Bulb Temp (°F)	80.01	80.04	79.98	0.06	0.01
Entering Wet Bulb Temp (°F)	66.97	67.05	66.89	0.16	0.03
Leaving Dry Bulb Temp (°F)	57.41	57.45	57.37	0.09	0.02
Leaving Wet Bulb Temp (°F)	56.48	56.57	56.47	0.10	0.02
Dry Bulb Delta T (°F)	22.60	22.66	22.55	0.11	0.02
Wet Bulb Delta T (°F)	10.49	10.60	10.42	0.18	0.01
Entering Grid Temp (°F)	80.13	80.20	80.05	0.14	0.02
Leaving Grid Temp (°F)	58.42	58.51	58.34	0.17	0.02
Static Pressure (inH2O)	0.28	0.29	0.27	0.02	0.00
Airflow (SCFM)	1802	1815	1794	21	1.5
Blower Motor (RPM)					
Stable? (Y or N)					
Outdoor Air Info					
Entering Dry Bulb Temp (°F)	95.01	95.07	94.97	0.10	0.01
Entering Wet Bulb Temp (°F)	71.50	71.57	71.45	0.12	0.02
Leaving Dry Bulb Temp (°F)					
Leaving Wet Bulb Temp (°F)		0.00	0.00	0.00	
Dry Bulb Delta T (°F)	95.01	95.07	94.97	0.10	0.01
Wet Bulb Delta T (°F)		0.00	0.00	0.00	
Static Pressure (inH2O)					
Airflow (SCFM)	0.00	0.00	0.00	0.00	0.000
Fan Motor (RPM)					
Stable? (Y or N)					
Power Info					
Voltage (V)	227.56	228.40	226.20	2.20	0.28
Frequency (Hz)					
Total Power (W)	4685.85	4707.24	4668.56	38.68	1.36
Total Current (A)	22.52	22.70	22.41	0.30	0.02
Coefficient Compressor Power (W)	3893.24	3905.31	3878.79	26.51	1.38
Measured Compressor Power (W)	3880.15	3894.26	3865.48	28.78	0.92
Coefficient Compressor Current (A)	17.59	17.64	17.53	0.12	0.01
Measured Compressor Current (A)	17.28	17.43	17.21	0.22	0.02
Fan Motor Power (W)	308.06	309.48	306.40	3.08	0.21
Fan Motor Current (A)	1.43	1.44	1.43	0.01	0.00
Blower Motor Power (W)	497.64	514.32	485.04	29.28	0.73
Blower Motor Current (A)	3.81	3.92	3.72	0.20	0.02

Table A1 (continued)

Blower Motor Efficiency (W / CFM)	27.62	28.43	26.87	1.55	0.02
Fan Motor Efficiency (W / SCFM)	0.00	0.00	0.00	0.00	0.00
Refrigerant Info					
Compressor Disch Press (psig)	396.21	397.46	394.71	2.75	0.17
Stable? (Y or N)					
Compressor Disch Sat Temp (°F)	115.83	116.06	115.54	0.52	0.03
Compressor Disch Temp (°F)	161.20	161.51	160.94	0.57	0.03
Compressor Disch SH (°F)	45.38	45.75	45.00	0.75	0.04
Outdoor Liq Press (psig)	367.43	368.44	366.41	2.03	0.06
Outdoor Liq Sat Temp (°F)	110.07	110.27	109.86	0.40	0.01
Outdoor Liq Temp (°F)	100.82	100.98	100.63	0.35	0.01
Outdoor Liq SC (°F)	9.25	9.42	9.07	0.35	0.02
Indoor Liq Press (psig)	368.97	370.51	367.57	2.93	0.27
Indoor Liq Sat Temp (°F)	110.37	110.68	110.10	0.58	0.05
Indoor Liq Temp (°F)	100.59	100.79	100.43	0.36	0.01
Indoor Liq SC (°F)	9.78	10.00	9.54	0.45	0.06
Indoor Liq Enthalpy (Btu / lbm)	51.96	52.05	51.89	0.16	0.00
Indoor Suct Press (psig)	143.38	143.63	143.12	0.51	0.08
Indoor Suct Sat Temp (°F)	50.03	50.13	49.93	0.20	0.03
Indoor Suct Temp (°F)	61.65	62.27	61.27	1.00	0.05
Indoor Suct SH (°F)	11.62	12.31	11.22	1.08	0.07
Indoor Suct Enthalpy (Btu / lbm)	124.68	124.86	124.58	0.29	0.02
Stable? (Y or N)					
Compressor Suct Press (psig)	144.94	145.34	144.61	0.73	0.11
Compressor Suct Sat Temp (°F)	50.63	50.79	50.51	0.28	0.04
Compressor Suct Temp (°F)	60.57	61.22	60.18	1.04	0.04
Compressor Suct SH (°F)	9.9	10.7	9.5	1.2	0.1
Stable? (Y or N)					
Compressor Dome Temp (°F)	161.70	161.97	161.49	0.48	0.03
Compressor Sump Temp (°F)	98.14	98.21	98.09	0.12	0.02
Compressor Sump SH (°F)	47.51	47.66	47.33	0.33	0.04
Pressure Deltas					
OD Coil Press Delta (psig)	28.78	29.69	27.52	2.16	0.17
Liq Line Press Delta (psig)	-1.54	-0.41	-2.37	1.96	0.26
ID Coil Press Delta (psig)	225.59	227.08	224.18	2.90	0.22
Suct Line Press Delta (psig)	-1.56	-1.29	-1.85	0.55	0.03
Compressor Press Delta (psig)	251.27	252.39	249.84	2.55	0.09
Compression Ratio (psia / psia)	2.57	2.58	2.57	0.02	0.00

Table A1 (continued)

Temperature Deltas (Refrigerant)					
OD Coil Temp Delta (°F)	60.39	60.77	59.99	0.79	0.03
Liq Line Temp Delta (°F)	0.22	0.27	0.17	0.09	0.01
Liq Line SC Delta (°F)	0.53	0.72	0.28	0.44	0.06
ID Coil Temp Delta (°F)	38.95	39.62	38.18	1.44	0.04
Suct Line Temp Delta (°F)	-1.08	-0.92	-1.20	0.28	0.01
Suct Line SH Delta (°F)	1.68	1.86	1.53	0.33	0.01
Compressor Temp Delta (°F)	100.64	101.19	100.22	0.97	0.05
Capacity / Efficiency					
ID Capacity (Btu / hr)	61,008	61,590	60,478	1,112	86
ID EER/COP (Btu / W-hr)	13.02	13.17	12.91	0.26	0.02
OD Capacity (Btu / hr)	0	0	0	0	0
OD EER/COP (Btu / W-hr)	0.00	0.00	0.00	0.00	0.00
OD Heat Balance (%)	0.00	0.00	0.00	0.00	0.00
ID & OD Ave Capacity (Btu / hr)	0	0	0	0	0
Measured Mass Flow (lbm / hr)	78.26	91.30	59.96	31.33	2.47
Stable? (Y or N)					
Coefficient Mass Flow (lbm / hr)	871.79	875.74	867.68	8.06	0.87
FM Capacity (Btu / hr)	3,936	4,873	2,617	2,256	177
FM EER/COP (Btu / W-hr)	0.84	1.04	0.56	0.48	0.04
FM Heat Balance (%)	93.55	95.77	92.01	3.76	0.30
ID & FM Ave Capacity (Btu / hr)	32,472	33,056	31,802	1,254	59
Comp Coefficient Capacity (Btu / hr)	61,065	61,284	60,895	389	52
Comp EER/COP (Btu / W-hr)	13.03	13.11	12.96	0.16	0.01
Comp Heat Balance (%)	-0.09	1.02	-1.01	2.03	0.21
ID & Comp Ave Capacity (Btu / hr)					
Sensible Heat Ratio (%)	0.74	0.74	0.73	0.01	0.00

Table A2 Old vs new unit comparison

	Test # (017)	Test # (033)
Indoor Air Info	(017) AHRI A2-RP	(033) A-Full-RP 0.28
Entering Dry Bulb Temp (°F)	80.00	80.01
Entering Wet Bulb Temp (°F)	67.04	66.97
Leaving Dry Bulb Temp (°F)	58.18	57.41
Leaving Wet Bulb Temp (°F)	56.50	56.48
Dry Bulb Delta T (°F)	21.82	22.60
Wet Bulb Delta T (°F)	10.54	10.49
Entering Grid Temp (°F)	79.48	80.13
Leaving Grid Temp (°F)	57.62	58.42
Static Pressure (inH2O)	0.28	0.28
Airflow (SCFM)	1717	1802
Blower Motor (RPM)		
Stable? (Y or N)		
Outdoor Air Info		
Entering Dry Bulb Temp (°F)	95.00	95.01
Entering Wet Bulb Temp (°F)	71.52	71.50
Leaving Dry Bulb Temp (°F)	102.18	
Leaving Wet Bulb Temp (°F)	73.14	
Dry Bulb Delta T (°F)	95.00	95.01
Wet Bulb Delta T (°F)	1.63	
Static Pressure (inH2O)	-0.01	
Airflow (SCFM)	0	0
Fan Motor (RPM)		
Stable? (Y or N)		
Power Info		
Voltage (V)	229.3	227.6
Frequency (Hz)		
Total Power (W)	4635	4686
Total Current (A)	21.12	22.52
Coefficient Compressor Power (W)	3862	3893
Measured Compressor Power (W)	3854	3880
Coefficient Compressor Current (A)	17.43	17.59
Measured Compressor Current (A)	16.67	17.28
Fan Motor Power (W)	334	308
Fan Motor Current (A)	1.48	1.43
Blower Motor Power (W)	447	498

Table A2 Old vs new unit comparison (continued)

Blower Motor Current (A)	2.98	3.81
Blower Motor Efficiency (W / CFM)	19.4	27.6
Fan Motor Efficiency (W / SCFM)	0.0	0.0
Refrigerant Info		
Compressor Disch Press (psig)	390.7	396.2
Stable? (Y or N)		
Compressor Disch Sat Temp (°F)	114.9	115.8
Compressor Disch Temp (°F)	161.3	161.2
Compressor Disch SH (°F)	46.4	45.4
Condenser Liq Press (psig)	379.7	367.4
Condenser Liq Sat Temp (°F)	112.6	110.1
Condenser Liq Temp (°F)	101.5	100.8
Condenser Liq SC (°F)	11.1	9.3
Evaporator Liq Press (psig)	379.9	369.0
Evaporator Liq Sat Temp (°F)	112.6	110.4
Evaporator Liq Temp (°F)	100.7	100.6
Evaporator Liq SC (°F)	11.9	9.8
Evaporator Liq Enthalpy (Btu / lbm)	52.0	52.0
Evaporator Suct Press (psig)	144.0	143.4
Evaporator Suct Sat Temp (°F)	50.5	50.0
Evaporator Suct Temp (°F)	59.6	61.6
Evaporator Suct SH (°F)	9.1	11.6
Evaporator Suct Enthalpy (Btu / lbm)	124.0	124.7
Stable? (Y or N)		
Compressor Suct Press (psig)	141.8	144.9
Compressor Suct Sat Temp (°F)	49.7	50.6
Compressor Suct Temp (°F)	61.7	60.6
Compressor Suct SH (°F)	12.0	9.9
Stable? (Y or N)		
Compressor Dome Temp (°F)	159.0	161.7
Compressor Sump Temp (°F)	116.3	98.1
Compressor Sump SH (°F)	66.6	47.5
Pressure Deltas		
OD Coil Press Delta (psig)	11.0	28.8
Liq Line Press Delta (psig)	-0.2	-1.5
ID Coil Press Delta (psig)	236.0	225.6
Suct Line Press Delta (psig)	2.1	-1.6
Compressor Press Delta (psig)	248.9	251.3
Compression Ratio (psia / psia)	2.8	2.6

Table A2 Old vs new unit comparison (continued)

Temperature Deltas (Refrigerant)		
OD Coil Temp Delta (°F)	59.8	60.4
Liq Line Temp Delta (°F)	0.8	0.2
Liq Line SC Delta (°F)	0.8	0.5
ID Coil Temp Delta (°F)	41.1	38.9
Suct Line Temp Delta (°F) Suct Line SH Delta (°F)	2.1 2.9	-1.1 1.7
Compressor Temp Delta (°F)	99.6	100.6
Capacity / Efficiency		
ID Capacity (Btu / hr)	58297	61008
ID EER (Btu / W-hr)	12.6	13.0
OD Capacity (Btu / hr)	0	0
OD EER (Btu / W-hr)	0.0	0.0
OD Heat Balance (%)	0.0	0.0
ID & OD Ave Capacity (Btu / hr)	0	0
Measured Mass Flow (lbm / hr)	-1877	78
Stable? (Y or N)		
Coefficient Mass Flow (lbm / hr)	826	872
FM Capacity (Btu / hr)	-135455	3936
FM EER (Btu / W-hr)	-29.2	0.8
FM Heat Balance (%)	332.4	93.5
ID & FM Ave Capacity (Btu / hr)	-38579	32472
Comp Coefficient Capacity (Btu / hr)	57372	61065
Comp EER (Btu / W-hr)	12.38	13.03
Comp Heat Balance (%)	1.58	-0.09
ID & Comp Ave Capacity (Btu / hr)		
Sensible Heat Ratio (%)	0.71	0.74
ID & OD & FM Ave Capacity (Btu / hr)	-25751	
Recommended Rating Capacity (Btu / hr)	0	
Recommended Rating EER (Btu / W-hr)	0	

Table A3 B test results

	Average	Max	Min	Range	Std Dev.
Indoor Air Info					
Entering Dry Bulb Temp (°F)	80.01	80.06	79.96	0.10	0.03
Entering Wet Bulb Temp (°F)	67.00	67.06	66.94	0.13	0.02
Leaving Dry Bulb Temp (°F)	56.37	56.42	56.34	0.08	0.01
Leaving Wet Bulb Temp (°F)	55.57	55.65	55.50	0.15	0.02
Dry Bulb Delta T (°F)	23.64	23.68	23.57	0.11	0.02
Wet Bulb Delta T (°F)	11.43	11.51	11.35	0.16	0.01
Entering Grid Temp (°F)	80.15	80.23	80.06	0.18	0.03
Leaving Grid Temp (°F)	57.44	57.53	57.36	0.17	0.02
Static Pressure (inH2O)	0.28	0.30	0.27	0.03	0.00
Airflow (SCFM)	1804	1816	1793	23	2.3
Blower Motor (RPM)					
Stable? (Y or N)					
Outdoor Air Info					
Entering Dry Bulb Temp (°F)	81.94	82.04	81.87	0.16	0.04
Entering Wet Bulb Temp (°F)	65.01	65.11	64.92	0.19	0.05
Leaving Dry Bulb Temp (°F)					
Leaving Wet Bulb Temp (°F)		0.00	0.00	0.00	
Dry Bulb Delta T (°F)	81.94	82.04	81.87	0.16	0.04
Wet Bulb Delta T (°F)		0.00	0.00	0.00	
Static Pressure (inH2O)					
Airflow (SCFM)	0.00	0.00	0.00	0.00	0.000
Fan Motor (RPM)					
Stable? (Y or N)					
Power Info					
Voltage (V)	227.63	228.50	225.90	2.60	0.29
Frequency (Hz)					
Total Power (W)	4141.91	4160.60	4124.08	36.52	4.46
Total Current (A)	20.15	20.28	19.98	0.29	0.05
Coefficient Compressor Power (W)	3336.27	3347.30	3327.47	19.83	2.58
Measured Compressor Power (W)	3328.07	3343.46	3316.58	26.88	2.64
Coefficient Compressor Current (A)	15.16	15.21	15.12	0.09	0.01
Measured Compressor Current (A)	14.79	14.87	14.73	0.14	0.01
Fan Motor Power (W)	311.81	313.38	309.94	3.44	0.23
Fan Motor Current (A)	1.45	1.45	1.44	0.01	0.00
Blower Motor Power (W)	502.03	520.56	490.40	30.16	3.49
Blower Motor Current (A)	3.91	4.07	3.80	0.26	0.04
Blower Motor Efficiency (W / CFM)	27.83	28.72	27.15	1.57	0.20

Table A3 B test results (continued)

Fan Motor Efficiency (W / SCFM)	0.00	0.00	0.00	0.00	0.00
Refrigerant Info					
Compressor Disch Press (psig)	337.59	338.89	336.64	2.25	0.27
Stable? (Y or N)					
Compressor Disch Sat Temp (°F)	104.17	104.44	103.97	0.48	0.06
Compressor Disch Temp (°F)	146.44	146.80	146.21	0.59	0.06
Compressor Disch SH (°F)	42.27	42.63	41.91	0.73	0.01
Outdoor Liq Press (psig)	305.04	306.30	304.41	1.89	0.22
Outdoor Liq Sat Temp (°F)	96.83	97.12	96.68	0.43	0.05
Outdoor Liq Temp (°F)	87.63	87.89	87.45	0.44	0.04
Outdoor Liq SC (°F)	9.19	9.41	9.01	0.40	0.02
Indoor Liq Press (psig)	307.03	308.42	306.11	2.30	0.19
Indoor Liq Sat Temp (°F)	97.28	97.59	97.07	0.52	0.04
Indoor Liq Temp (°F)	87.53	87.81	87.34	0.47	0.04
Indoor Liq SC (°F)	9.75	10.03	9.53	0.49	0.02
Indoor Liq Enthalpy (Btu / lbm)	46.46	46.58	46.38	0.20	0.02
Indoor Suct Press (psig)	140.59	140.92	140.24	0.68	0.05
Indoor Suct Sat Temp (°F)	48.94	49.06	48.80	0.27	0.02
Indoor Suct Temp (°F)	61.34	61.90	60.84	1.06	0.06
Indoor Suct SH (°F)	12.40	13.00	11.65	1.35	0.07
Indoor Suct Enthalpy (Btu / lbm)	124.82	124.97	124.62	0.36	0.02
Stable? (Y or N)					
Compressor Suct Press (psig)	141.54	141.98	141.10	0.88	0.06
Compressor Suct Sat Temp (°F)	49.31	49.48	49.13	0.35	0.02
Compressor Suct Temp (°F)	60.36	60.98	59.64	1.35	0.05
Compressor Suct SH (°F)	11.1	11.7	10.2	1.5	0.1
Stable? (Y or N)					
Compressor Dome Temp (°F)	146.93	147.26	146.74	0.52	0.05
Compressor Sump Temp (°F)	93.41	93.55	93.32	0.23	0.05
Compressor Sump SH (°F)	44.11	44.29	43.87	0.42	0.04
Pressure Deltas					
OD Coil Press Delta (psig)	32.55	33.55	31.58	1.97	0.31
Liq Line Press Delta (psig)	-1.99	-0.79	-3.08	2.29	0.12
ID Coil Press Delta (psig)	166.44	167.74	165.56	2.17	0.18
Suct Line Press Delta (psig)	-0.94	-0.64	-1.22	0.58	0.02
Compressor Press Delta (psig)	196.06	197.08	195.17	1.91	0.29
Compression Ratio (psia / psia)	2.25	2.26	2.25	0.01	0.00
Temperature Deltas (Refrigerant)					
OD Coil Temp Delta (°F)	58.81	59.16	58.39	0.77	0.06
Liq Line Temp Delta (°F)	0.11	0.15	0.03	0.12	0.01

Table A3 B test results (continued)

Liq Line SC Delta (°F)	0.56	0.81	0.26	0.55	0.03
ID Coil Temp Delta (°F)	26.19	27.06	25.51	1.55	0.08
Suct Line Temp Delta (°F)	-0.98	-0.82	-1.15	0.33	0.01
Suct Line SH Delta (°F)	1.35	1.57	1.18	0.40	0.01
Compressor Temp Delta (°F)	86.08	86.59	85.68	0.91	0.04
Capacity / Efficiency					
ID Capacity (Btu / hr)	66,111	66,524	65,599	925	43
ID EER/COP (Btu / W-hr)	15.96	16.09	15.85	0.23	0.02
OD Capacity (Btu / hr)	0	0	0	0	0
OD EER/COP (Btu / W-hr)	0.00	0.00	0.00	0.00	0.00
OD Heat Balance (%)	0.00	0.00	0.00	0.00	0.00
ID & OD Ave Capacity (Btu / hr)	0	0	0	0	0
Measured Mass Flow (lbm / hr)	86.88	99.74	67.52	32.22	1.36
Stable? (Y or N)					
Coefficient Mass Flow (lbm / hr)	855.82	860.33	851.63	8.70	0.53
FM Capacity (Btu / hr)	5,027	6,033	3,528	2,505	101
FM EER/COP (Btu / W-hr)	1.21	1.46	0.85	0.61	0.02
FM Heat Balance (%)	92.40	94.68	90.84	3.84	0.15
ID & FM Ave Capacity (Btu / hr)	35,569	36,139	34,861	1,278	58
Comp Coefficient Capacity (Btu / hr)	64,677	64,916	64,452	465	33
Comp EER/COP (Btu / W-hr)	15.62	15.72	15.53	0.18	0.02
Comp Heat Balance (%)	2.17	2.89	1.30	1.59	0.07
ID & Comp Ave Capacity (Btu / hr)					
Sensible Heat Ratio (%)	0.71	0.72	0.71	0.01	0.00

Table A4

	Cycle 1	Cycle 2	Cycle 3	Highest Cd
Entering Grid Offset	-0.14	-0.14	-0.14	0.13
Leaving Grid Offset	-1.31	-1.31	-1.31	
Off Mode Watts	6.32	6.34	6.34	
EER ss,dry	14.35	14.35	14.35	
Measured Airflow	1885.26	1885.26	1885.26	
Standard Airflow	1860.15	1860.15	1860.15	
Cp,a	0.24	0.24	0.24	
Gamma	2.63	2.66	2.66	
Vn	13.57	13.57	13.57	
Blower Delay	1.00	1.00	1.00	
q cyc,dry (Gross)	5273.43	5327.94	5335.54	
q cyc,dry (Adj)	0.00	0.00	0.00	
q cyc,dry (Net)	5273.43	5327.94	5335.54	
e cyc,dry (Gross)	412.85	409.52	410.77	
e cyc,dry (Adj)	0.00	0.00	0.00	
e cyc,dry (Net)	412.85	409.52	410.77	
EER cyc,dry	12.77	13.01	12.99	
Q ss,dry	58542.71	58542.71	58542.71	
E ss,dry	4078.84	4078.84	4078.84	
dt cyc,dry	0.50	0.50	0.50	
CLF	0.18	0.18	0.18	
Cd	0.13	0.11	0.12	
SEER	14.89	15.05	15.03	
Part Load	0.93	0.94	0.94	
B-EER	15.96	15.96	15.96	
441 - EER ss,dry	14.35	14.35	14.35	
441 - q cyc,dry (Gross)	5273.43	5327.94	5335.54	
441 - q cyc,dry (Adj)	0.00	0.00	0.00	
441 - q cyc,dry (Net)	5273.43	5327.94	5335.54	
441 - e cyc,dry (Gross)	412.85	409.52	410.77	
441 - e cyc,dry (Adj)	0.00	0.00	0.00	
441 - e cyc,dry (Net)	412.85	409.52	410.77	
441 - EER cyc,dry	12.77	13.01	12.99	
441 - Q ss,dry	58542.71	58542.71	58542.71	
441 - E ss,dry	4078.84	4078.84	4078.84	
441 - CLF	0.18	0.18	0.18	
441 - Cd	0.13	0.11	0.12	
441 - SEER	14.89	15.05	15.03	
441 - Part Load	0.93	0.94	0.94	
441 - B-EER	15.96	15.96	15.96	

Table A5 Definitions - Nomenclature

HBx Heat balance for test x
$\dot{q}tci,x$ Total cooling capacity for test *x*, indoor side data, Btu/h
$\dot{q}tco,x$ Total cooling capacity for test *x*, outdoor side data, Btu/h
$\dot{q}ref,x$ Ttal capacity as measured by the refrigerant enthalpy method, Btu/h
$EERx$ Energy efficiency ratio for test *x*, Btu/W·h
$\dot{q}tci,x$ Total cooling capacity for test *x*, indoor side data, Btu/h
$Ptot,x$ Total power for test *x*, W
$EERx$ Energy efficiency ratio for test *x*, Btu/W·h
$\dot{q}tci,x$ Total cooling capacity for test *x*, indoor side data, Btu/h
$Ptot,x$ Total power for test *x*, W
$\dot{q}x$ Indoor capacity for test *x* before any duct or blower adjustments, Btu/h
$\dot{Q}mi$ Airflow, indoor, measured, cfm
$ha1$ Enthalpy, air entering indoor side, Btu/lbmda
$ha2$ Enthalpy, air leaving indoor side, Btu/lbmda
$v'n$ Specific volume of air at the nozzle, ft3/lbm of air-water vapor mixture
Wn Humidity ratio at the nozzle, lbmWV/lbmda
SEER – Seasonal Energy Efficiency Ratio
$PLF(0.5)$ Part Load Factor for SEER
$EERx$ Energy efficiency ratio for test *x*, Btu/W·h