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Using Simulation Model to Reduce System Design Time and Cost

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ABSTRACT

This paper presents the findings of a case study in which System Design Simulator, a steady-state system modeling tool, was used to evaluate design options rather than implementing the changes incrementally in a laboratory and evaluating the results of each change. Simulation software has proven to drastically reduce development time and cost by limiting the need for expensive and time consuming laboratory testing. Three different system types were used in this study to show the capability of the model and identify design options for improving system performance.

1. INTRODUCTION

The air-conditioning and refrigeration industry has seen unprecedented regulatory emphasis on energy efficiency improvement over the last decade. Increasingly, researchers around the globe are looking at ways to achieve lower energy consumption, while still maintaining output, reducing carbon footprint and global warming potential. To achieve this objective, engineers must change the product design both at the system level and the component level. System level changes are complex, involving interaction between multiple components such as valves, compressors and heat exchangers. Understanding the effects of these changes traditionally involved trial and error methods, and costly lab experimentation through iterative testing. Here, we present an alternative method using a powerful software tool 'System Design Simulator, (SDS)' to model these changes and predict the outcome before attempting actual tests. Three different systems are used to evaluate the capability of the modeling tool.

3-Ton Air Source Heat Pump (ASHP): It was selected as it is the most common system using conventional round tube finned heat exchangers in Indoor and Outdoor Units.

5-Ton Heat Pump Pool Heater: This unit was selected to show SDS's ability to model other system types. It uses coaxial heat exchanger in the Outdoor Unit and round tube finned heat exchanger in the Indoor Unit. Such systems are common in the coastal regions of Southern US.

3-Ton Residential Split Air-Conditioner with a Microchannel Condenser: Unit is equipped with Microchannel condenser in the Outdoor Unit whereas the Indoor Unit has round tube finned heat exchanger.

2. METHODOLOGY

This paper presents the results of modeling exercise of 3 systems: (i) 3-Ton ASHP, (ii) 3-Ton AC Split System, and the (iii) 5-Ton Pool Heater using SDS. Simulation results were broadly divided into two categories, validation results and findings from the design optimization exercise. Validation was completed for both the Pool Heater and the ASHP and Split Systems. Design optimization results are presented only here for the 3-Ton ASHP, wherein the following design options were considered:

1. Optimize refrigerant circuits in indoor and outdoor coils.
2. Optimize refrigerant charge by managing compressor superheat and condenser sub-cooling.
3. Change to higher efficiency fan motors in indoor and outdoor units.
4. Optimizing the air flow rate in indoor and outdoor units.
5. Effect of smaller displacement compressor.
6. Evaluate effect of two-capacity compressor.

Breakdown of gain from each design option considered above will be presented in this paper. Using the simulation tool to model the system and analyze numerous design changes eliminated several weeks of laboratory testing and evaluation. While the real cost of engineering time varies by organization, it can safely be shown that there was a significant cost saving associated with using the simulation tool. It also offers opportunity to streamline the product development process and speed of the time it takes to get new products to market.

3. SYSTEM DESIGN SIMULATOR TOOL

The System Design Simulator was used for the analysis presented in this paper. This tool is based on the modeling engine developed by Oak Ridge National Labs. The simulation tool has since been enhanced with several key features which are briefly listed below.

- SDS is a hardware-based model with rapid processing speed and Windows interface. Included is a built-in database of over 10,000 compressor models
- Capability to simulate performance of Air Source Heat Pump using round tube / finned heat exchanger, Air-Conditioner with Microchannel heat exchanger and Water Source Heat Pump using Tube-in-Tube heat exchanger
- Refrigerant selection choices are: R-22, R-134a, R-404A, R-507, R-410A, R-32, R-407C and R-290
- SEER and HSPF capability
- Parametric performance mapping of selected design variables (Compressor Superheat, Subcooling, Air Flow Rate, Refrigerant Charge, etc.)
- Evaluate effect of pressure drop of system accessories (e.g. Reversing Valve, Accumulator, etc.)
- Integrates several complementary tools: Re-rate Compressor Performance at user specified conditions, Psychrometric Chart/Calculator, Refrigerant Properties lookup, Stand alone Microchannel condenser model, etc.
- Database of coaxial heat exchangers for water source heat pump modeling function

3. VALIDATION RESULTS

3.1 5-Ton Pool Heater

A commercially available 5-Ton, pool heater system was chosen for the validation task. It uses coaxial heat exchanger in the Outdoor Unit and round tube finned heat exchanger in the Indoor Unit. The compressor is a scroll model running on single phase power.

3.1.1 5-Ton pool heater test

The schematic of the test set up is shown in Figure 1, with the required instrumentation.

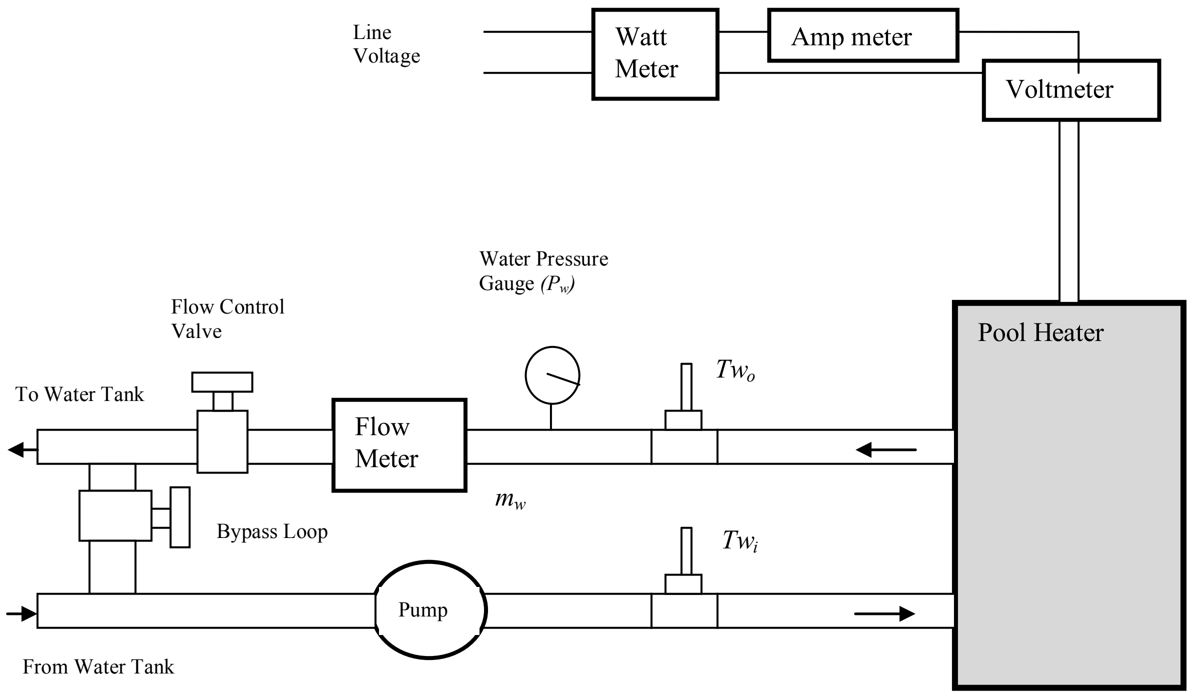

Figure 1: Pool heater test setup

Two standardized tests were completed. These tests are adapted from the standard AHRI 1160. Tests shown in Table 1 represent 'Standard Rating Tests' for Pool Heaters. The unit was tested at the name plate voltage, at the factory setting of Thermal Expansion Valve with a refrigerant charge amount of 3 lb 11 oz. Refrigerant R-410A was used in the tests. The unit was operated at its name plate voltage. Data scans were taken at 10 second intervals, measurements were made with calibrated instruments per ISO17025 standards. The following refrigerant side measurements were made:

T_1 – Compressor discharge temperature (°F)
T_2 – Liquid line temperature (°F)
T_3 – Thermal Expansion Valve outlet (°F)
T_4 – Compressor suction temperature (°F)
T_{ei} – Every Evaporator inlet circuit temperature (°F)
T_{eo} – Every Evaporator outlet circuit temperature (°F)
P_1 – Discharge Pressure (Psi)
P_2 – Liquid Line (Psi)
P_3 – Compressor suction (Psi)
P_w – Water Pressure (Psi)

Table 1: Standard rating tests for pool heaters, AHRI 1160

	Air Temperature Surrounding Unit		Water Temperature Entering Unit	Water Flow Rate (or Less if Specified by the Manufacturer)	
	Dry-bulb °F [°C]	Wet-bulb °F [°C]	°F [°C]	gpm	L/s
High Air Temperature - Mid Humidity (62% RH)	80.6 [27.0]	70.7 [21.5]	80.0 [26.7]	0.450 per 1000 Btu/h	0.028 per 293.1 Watts
Low Air Temperature - Mid Humidity (63% RH)	50.0 [10.0]	44.6 [7.0]	80.0 [26.7]	Same flow rate as established in High Air Temperature - Mid Humidity (62% RH)	

For each of the standard rating test conditions, the following formulations are used in computing Capacity and COP of the system under test.

SST	T_{sat} based on suction pressure
SLT	T_{sat} based on liquid pressure)
SSH	$T_{suction} - SST$
SC	$SLT - Condenser\ refrigerant.\ outlet\ temperature$)
Water Delta	$\Delta T = T_{w,out} - T_{w,in}$
Water Flow [m^3/s]	$(m_w[lb/h]*0.0283)/(62.4*3600)$ Where m_w is the mass flow rate of water
Gross Capacity	$Q = m_w * 0.9991 * (T_{w,out} - T_{w,in})$
Net Capacity	$Q_{net} = Q_{gross} + ((mw[m^3/s]*P_w)*3.412)/0.3$
COP	$COP = Q_{gross} * 0.2928104 /(E_{total} + (m_w*P_w/0.3)$ Where E_{total} is total energy

We show the measured test results below for the high air temperature test at 80.6°F dry bulb and 70.7 °F wet bulb, with an entering water temperature controlled to 80 °F.

Table 2: Standard rating measured test result for 5-ton pool heater

	System Capacity (Btu/hr)	Compressor Power (W)	Total Power (W)	System COP
Tested	60,086	2,604	2,909	6.05

3.1.2 5-Ton pool heater modeling

The simulation effort required preparing detailed inputs for SDS. The heat exchanger geometries were obtained by carefully checking and measuring the physical attributes of the actual hardware which included, number of rows, tubes, their diameters and spacing, smooth / rifled tubing, refrigerant circuits, fin geometry, connecting tubing geometries, estimates of line heat transfer, actual air / water flow rates, fan / pump power inputs, inlet air / water conditions and so on. The compressor performance is based on the ten-term coefficients for refrigerant mass flow rate and power of the compressor along with its rated condition (Compressor Superheat, Subcooling).

3.1.3 5-Ton pool heater SDS validation results

Validation results are shown in Figure 2 below for the high air temperature standard rating test shown in Table 1, compared against the measured test data for the same condition shown in Table 2.

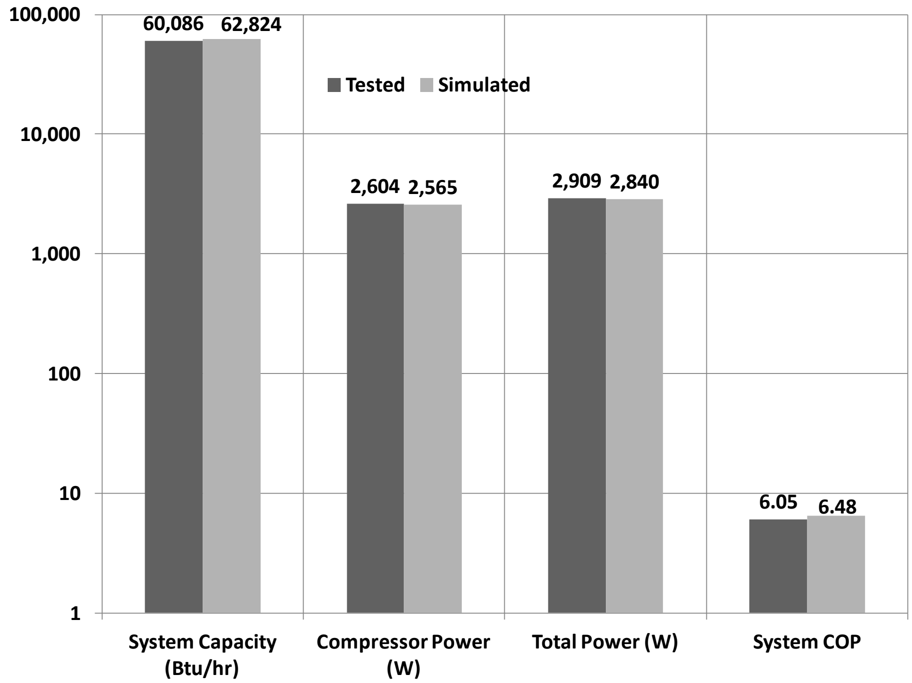

Figure 2: Pool heater simulation results-high temperature standard rating

Table 3 shows the percent error in the predicted versus the actual data. Simulated data for capacity, and COP were higher than actual tested values, while power predictions were lower.

Table 3: Prediction errors for 5-ton pool heater with SDS tool

	System Capacity (Btu/hr)	Compressor Power (W)	Total Power (W)	System COP
% Error	+5%	-1%	-2%	+7%

These validation errors may appear to be high upon first examination particularly for the COP. In practice, compressor manufacturers provide their published data that is typically in the ± 5% range. Taking this into account, difference for capacity appears to be reasonable.

3.2 3-Ton Residential Heat Pump with Fin/Tube and 3-Ton Split Air-Conditioner with Microchannel Condenser

A three ton Residential Split AC System and a 3-Ton Air Source Heat Pump (ASHP) was selected for validation work. The Split AC system is equipped with Microchannel condenser in the Outdoor Unit whereas the Indoor Unit has conventional Round Tube Finned heat exchanger. The ASHP has conventional Round Tube Finned heat exchangers on both Indoor and Outdoor sections. This type of system is commonly used in many residential or commercial buildings for comfort cooling or heating. Both these systems use Scroll compressors operated with R-410A refrigerant, very commonly used in many residential applications.

3.2.1 Test set up and results

Test set up was completed according to ASHRAE 37 and the AHRI 210/240 test standards. These standards are widely used in the industry and are required to be followed by many regulatory agencies.

Table 4: Test conditions for the 3-ton residential system

Test Description	Air Entering Indoor Unit Temperature		Air Entering Outdoor Unit Temperature		Cooling Air Volume Rate
	Dry-Bulb (ºF)	Wet-Bulb (ºF)	Dry-Bulb (ºF)	Wet-Bulb (ºF)	
A Test	80.0	67.0	95.0	75.0	Cooling Full Load
B Test	80.0	67.0	82.0	65.0	Cooling Full Load
C Test	80.0	≤ 57.0	82.0	-	Cooling Full Load
D Test	80.0	≤ 57.0	82.0	-	Airflow Nozzle(s) Static Pressure Difference Same As During C

A & B test points shown in Table 4 above were run. Measured data for these conditions is shown below in Table 5 for the ASHP and for the 3-Ton split system with micro channel condenser.

Table 5: Test results for the 3-ton systems (Test point B only shown)

	Net System Capacity (Btu/hr)	Compressor Power (W)	Total Power (W)	System EER (Btu/Wh)
3-Ton ASHP	36,982	2,169	2,590	14.28
3-Ton Split MCHX	36,262	2,062	2,645	13.71

3.2.2 3-Ton residential system SDS validation results

The simulation model was set up using the SDS software. The inputs were prepared as described in section 3.1.2 by measuring the physical attributes of the actual system hardware. The compressor data consisting of the ten-term coefficients of mass flow rate and power were obtained from the built-in compressor database.

The validation results are shown in Figures 3 and 4 below for the two 3-Ton systems.

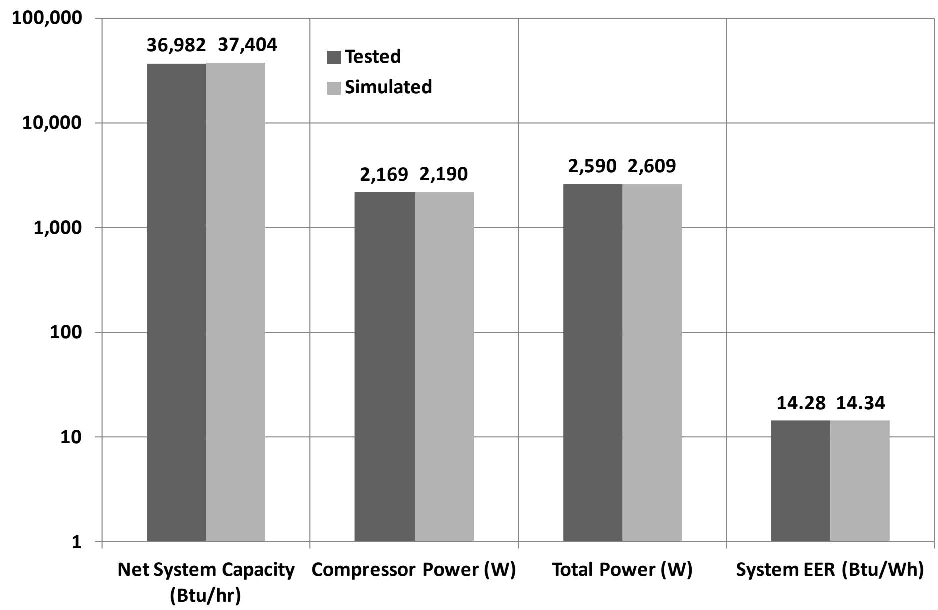

Figure 3: Validation results for the 3-Ton ASHP

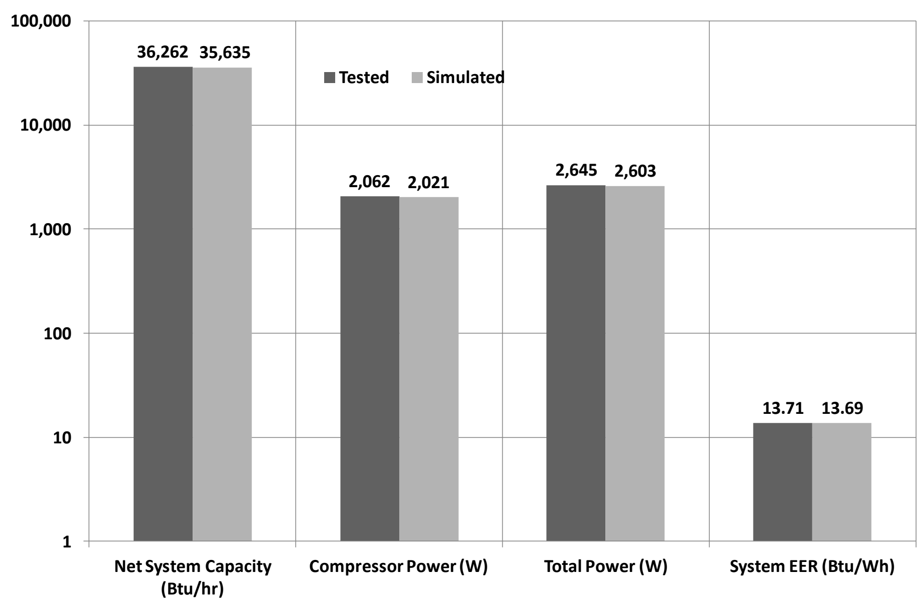

Figure 4: Validation results for the 3-ton split air-conditioner with MCHX

Table 6 below shows the prediction errors for both the 3-Ton systems used for validation.

Table 6: Prediction errors for 3-ton split air-conditioner and ASHP

	Net System Capacity (Btu/hr)	Compressor Power (W)	Total Power (W)	System EER (Btu/Wh)
3-Ton ASHP (% Error)	1.14%	0.97%	0.73%	0.40%
3-Ton Split AC with MCHX (% Error)	-1.73%	-1.99%	-1.59%	-0.14%

As can be seen from Table 6, the test data and simulation results co-relate very well for the two models considered. As mentioned before, typical component performance variation from one system to another will cause tested system results to vary, thus affecting the validation results. Accurate entry of system configuration and compressor information plays a significant role in the accuracy of the predictions.

4. OPTIMIZATION

In this section, we evaluated the capability of the SDS tool to predict the effect of component changes on the overall system performance. We choose the 3-Ton ASHP system validated above, which has the conventional heat exchanger arrangements on both the outdoor and indoor units. This type of system is very commonly used in many residential and small office buildings, and is therefore chosen as the test case. The following hardware changes were considered to optimize the system performance.

1. Optimize refrigerant circuits in indoor and outdoor coils
2. Optimize refrigerant charge by managing compressor superheat and condenser subcooling
3. Change to higher efficiency fan motors in indoor and outdoor units
4. Optimizing the air flow rate in indoor and outdoor units
5. Effect of smaller displacement compressor
6. Evaluate effect of two-capacity compressor

4.1 3-Ton ASHP Optimization Results

For each of the six optimization cases considered above, we started with the base model, for which validation results were presented in section 3. Each change was treated incrementally, so that we may isolate and understand its' affect on system performance. Refrigerant remained R-410A. Table 7 below summarizes the results and impact of each of these changes. All data is reported at the B test point.

Table 7: Design optimization results of 3-ton ASHP model (B test point)

Record No.	Design Change	Incremental Gain (%)	Total Gain in SEER (%)	SEER (Btu/Wh)
1	Optimize Number of Refrigerant Circuits In Indoor and Outdoor Units	1.0	1.0	13.75
2	Optimize Sub-cooling and Superheat	0.0	0.0	13.75
3	BPM Indoor Fan Motor & Higher Efficiency Outdoor Fan Motor	3.3	4.3	14.21
4	Blower Operation -Reduce Air Flow Rate to 70%	2.5	6.8	14.54
5	Lower Displacement Compressor	3.1	9.9	14.97
6	Two-Capacity Compressor	6.3	16.2	15.83

From Table 7, we see that the simulation tool predicts the best incremental SEER improvement of 6.3% may be obtained by changing to a two-capacity compressor model. For the purpose of the simulation, a Two-Step compressor model ZPS31K5E-PFVwas used. No incremental improvement was seen altering the charge amount, in effect changing the Superheat and Subcooling. The charge amount would remain at 9 lb 4 oz as in the base case. The cumulative SEER gain when all the above changes are incorporated was 16.2%, with the final SEER at 15.83.

5. Concluding Remarks

For many companies and research facilities, the only way to predict the outcome of design change is to implement the change and conduct an actual test in psychrometric room. To test a battery of changes as shown in section 4, we would need extensive test facility time and labor to make the hardware changes. In our estimation, the test time could be as much as 12 weeks to iteratively change and test each configuration. SDS provides an estimation of the effect of the various design options, without once going to the test facility. Validation results presented in this paper show the software tools can be a viable alternative to rigorous and costly testing. Once the simulation model was set up, we found it relatively quick to evaluate the various changes and predict outcomes. With these predictions in hand, we can now go to the test facility and only test those configurations that provide the most efficiency gains.

NOMENCLATURE

ASHP	Air Source Heat Pump
MCHX	Microchannel Heat Exchanger
SEER	Seasonal Energy Efficiency Ratio
SST	Saturated Suction Temperature
SLT	Saturated Liquid Temperature
SSH	Suction Superheat
SC	Liquid Subcooling

REFERENCES

ARI standard 210/240 *Standard for Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment* Air-Conditioning, Heating and Refrigeration Institute, 2008; 2111 Wilson Blvd, Suite 500, Arlington, VA 22201, U.S.A.

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