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ABSTRACT

This paper presents the findings of a case study in which System Design Simulator, a steady-state system simulation software was used to evaluate design options rather than implementing the various changes incrementally in a laboratory and evaluating the results of each change. System modeling software has proven to significantly reduce development time and cost by limiting the need for expensive and time-consuming laboratory testing. In this investigation we used a nominal fixed capacity 5-ton, 14 SEER Heat Pump using Refrigerant R-410a and identified opportunities that were relatively easy to implement.

1. INTRODUCTION

The HVAC industry has seen unprecedented regulatory emphasis on energy efficiency improvement over the last decade. Increasingly, engineers are looking for opportunities to achieve lower energy usage, while still maintaining system performance, comfort, reducing carbon footprint and global warming potential. To achieve this objective, engineers need to modify the product design both at the system level and the component level. Interaction of system components such as expansion device, compressor, heat exchangers along with the changes in indoor/outdoor air condition are complex. Understanding their contribution in a system context traditionally involved trial and error methods, and costly laboratory experimentation through iterative testing. In this study we used Emerson's steady-state, hardware-based system simulation model 'System Design Simulator, SDS' to evaluate the various options to determine the final system design that can deliver our performance target before we committed the system for actual laboratory validation of the performance.

2. METHODOLOGY

As a first step, we tested the baseline fixed capacity, 14.0 SEER heat pump in the laboratory in its "As Received" state to verify the published cooling performance. Published heating mode seasonal performance of the selected system has 8 HSPF (Btu/Wh). Next, we validated the simulation model before proceeding to evaluate the various design options for improving its performance. The simulation effort required preparing detailed inputs for the model. 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 indoor and outdoor air flow rates, fan power inputs, inlet air conditions and so on. The compressor performance is based on the ten-term AHRI coefficients (ANSI/AHRI Standard 540 (2015) for refrigerant mass flow rate and power of the compressor for each compressor speed along with its rated compressor superheat and subcooling.

Our findings showed the system model provides excellent accuracy (about ± 5%) between measured and simulated values. System level accuracy of ± 5% is excellent keeping in perspective that system's critical component i.e.

compressor's rated performance published by manufacturers have an accuracy of ± 5%. Having confirmed the accuracy of the model we moved to next the task of finding opportunities for improving system's cooling and heating performance. It is important to note that from the practical standpoint only those design options that were simple to implement were considered. These were: Variable Speed Compressor, Flow Control, Higher Efficiency Condenser and Evaporator Fans and Optimization of Refrigerant Charge.

We selected a nominal 5-ton speed compressor and matching variable speed drive to achieve the target design capacity. Next, we used the system model to evaluate several combinations of Indoor and Outdoor Air Flow Rates, Condenser Exit Subcooling, Compressor Superheat and Refrigerant Charge for each test listed in ARI Standard 210/240-2008 (refer to Tables 1 and 2). This standard is widely used in the industry and is required to be followed by system manufacturers. An additional benefit of the Indoor Fan Strategy is improved dehumidification during cooling mode and higher delivered temperature during heating mode operation. Once we identified the optimum cooling and heating system configuration we proceeded to the testing phase. Results are summarized in Figures 1 - 8. Note, heating performance at $H2_2$ and $H2_v$ test points were not simulated as these tests have a defrost component and thus transient in behavior making it outside the capability of the steady-state model. Additionally, we conducted laboratory tests to accurately determine the Coefficient of Degradation C_d in Cooling (Tests: G_1 and I_1) and Heating Modes (Test: $H0C_1$) needed for computing seasonal Cooling (SEER) and Heating performances (HSPF).

Using the simulation tool to model the system and analyze numerous design changes eliminated several weeks of time consuming and expensive laboratory testing and evaluation. While the real cost of engineering time may vary by organization, it can safely be shown that there was a significant cost saving associated in use of the simulation model. It also offers opportunity to streamline the product development process and speed of the time it takes to get new products to market.

Table 1: Cooling Mode Test Conditions for Units with Variable-Speed Compressor

	Temperature of Air Entering Indoor Unit		**Temperature of Air Entering Outdoor Unit**		**Compressor Speed**
Test Description	Dry Bulb Temp. (ºF)	Wet Bulb Temp. (ºF)	Dry Bulb Temp. (ºF)	Wet Bulb Temp. (ºF)	
A2 (Steady-State-State, Wet Coil)	80	67	95	75	Maximum
B2 (Steady-State, Wet Coil)	80	67	82	65	Maximum
Ev (Steady-State, Wet Coil)	80	67	87	69	Intermediate
B1 (Steady-State, Wet Coil)	80	67	82	65	Minimum
F1 (Steady-State, Wet Coil)	80	67	67	53.5	Minimum
G1 (Optional, Steady-State Dry Coil) [3]	80	(1)	67	---	Minimum
I_1 (Cyclic, Dry Coil) [3]	80	(1)	67	---	Minimum

[3] G_1 and I_1 are cyclical tests used to determine the Cooling Mode Coefficient of Degradation C_d.

Table 2: Heating Mode Test Conditions for Units with Variable-Speed Compressor

Test Description	Temperature of Air Entering Indoor Unit		Temperature of Air Entering Outdoor Unit		Compressor Speed
	Dry Bulb Temp. (ºF)	Wet Bulb Temp. (ºF)	Dry Bulb Temp. (ºF)	Wet Bulb Temp. (ºF)	
H0$_1$ (Steady-State)	70	60	62	56.5	Minimum
H0C$_1$ (Optional, Cyclic) [4]	70	60	62	56.5	
H1$_2$ (Steady-State)	70	60	47	43	Maximum
H1$_1$ (Steady-State)	70	60	47	43	Minimum
H1$_N$ (Optional)	70	60	47	43	Nominal
H2$_2$ (Optional)	70	60	35	33	Maximum
H2$_V$ (Required)	70	60	35	33	Intermediate
H3$_2$ (Steady-State)	70	60	17	15	Maximum

[4] H0C$_1$ is a cyclical test used to determine the Heating Mode Coefficient of Degradation C_d.

3. VALIDATION RESULTS

3.1 Test Setup

The schematic of the test set up is shown in Figure 1, with all required measurements of temperatures, pressure, mass flow rate and electrical power. We followed the standard AHRI 210/240-2008 in conducting the tests and unit was operated at its name plate voltage. Data scans were taken at 10 second intervals and all measurements were made with calibrated instrumentation per ISO17025 standards. Code Tester is used for accurate measurement of air flow rates.

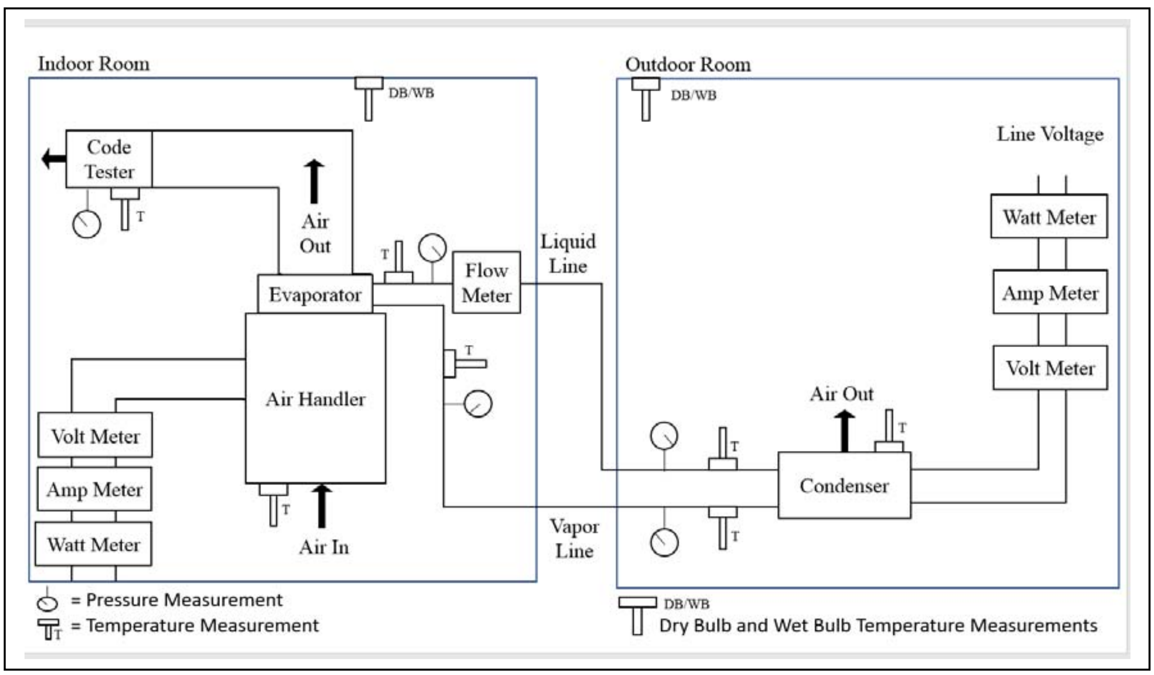

Figure 1: Setup Used for Conducting System Tests

3.2 Cooling Mode Results

We used the cooling mode system configuration shown in Table 3 for validation of simulation results in laboratory. Note test conditions are per ARI Standard 210/240-2008. Validation results for the cooling mode operation are shown in Figures 1 to 4 below for Capacity, Power, System Efficiency and Error for simulated vs. tested data. As may be seen the test data and simulation results co-relate quite well and the error ranged between 0.8% to 5.3% for wide range of system conditions. Realistically, we should not expect simulation accuracy lower than 5% consistently over a wide range since the accuracy of published compressor has an accuracy tolerance of ± 5%. Improving the entry of system configuration and compressor information plays a significant role in the accuracy of the predictions.

Table 3: System Configuration for Cooling Mode Validation

ARI 210/240 Test Point	Compressor Speed (RPM)	Indoor Air Flow Rate (CFM)	Indoor Fan Power (W)	Outdoor Air Flow Rate (CFM)	Outdoor Fan Power (W)
A_2	4,500	1,740	469	4,215	184
B_2	4,500	1,740	469	4,215	184
Ev	2,883	1,185	148	2,227	128
B_1	2,075	895	108	2,985	74
F_1	2,075	895	108	2,985	74

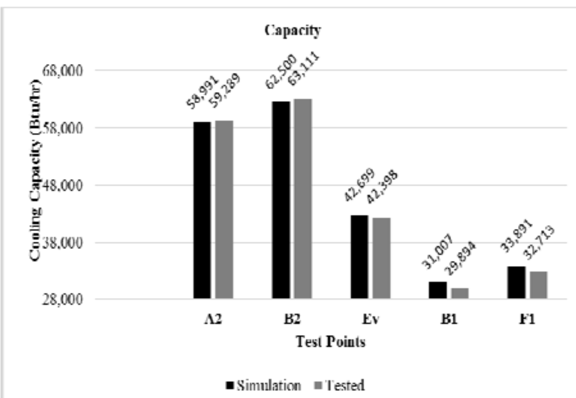

Figure 1: Capacity

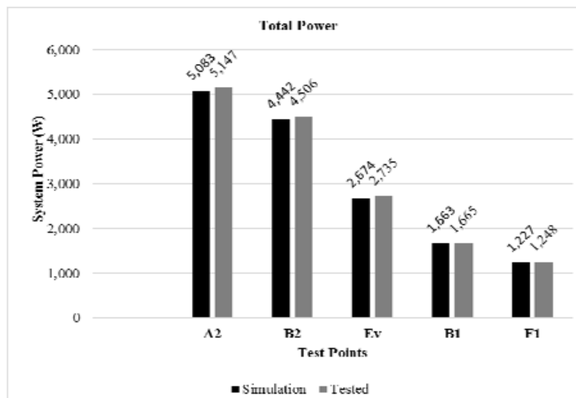

Figure 2: System Power

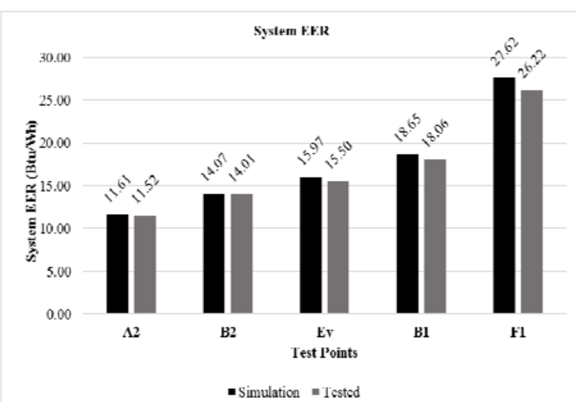

Figure 3: System Efficiency

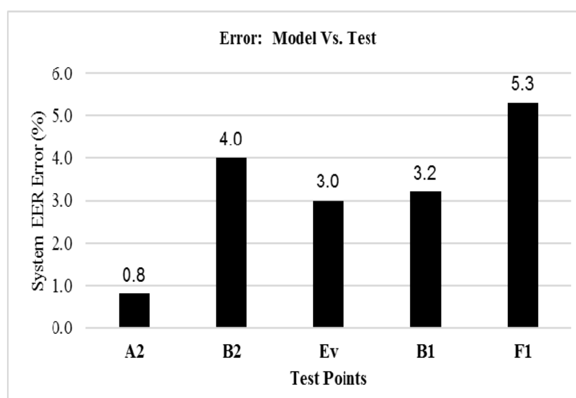

Figure 4: Prediction - Error (%)

3.3 Heating Mode Results

We used heating mode system configuration shown in Table 4 for validation of simulation results in laboratory. Note, tests $H2_2$ and $H2_v$ were not included in our validation effort as these are transient in nature because of frost build up and thus outside the capability of our steady-state model. Validation results are shown in Figures 5 to 8 below for Capacity, Power, System Efficiency and Error for simulated vs. tested data. Heating mode error was smaller vs. cooling mode and the maximum error was 3%.

Table 4: System Configuration for Heating Mode Validation

ARI 210/240 Test Point	**Compressor Speed (RPM)**	**Indoor Air Flow Rate (CFM)**	**Indoor Fan Power (W)**	**Outdoor Air Flow Rate (CFM)**	**Outdoor Fan Power (W)**
$H1_2$	4,500	1,870	460	4,215	184
$H3_2$	4,500	1,870	460	4,215	184
$H1_1$	2,075	1,180	123	1,880	25
HO_1	2,075	1,180	123	1,880	23

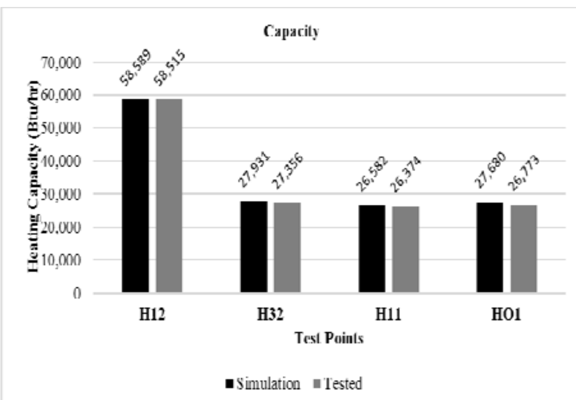

Figure 5: Capacity

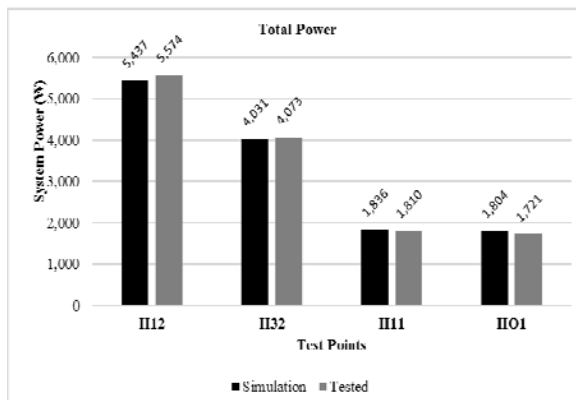

Figure 6: System Power

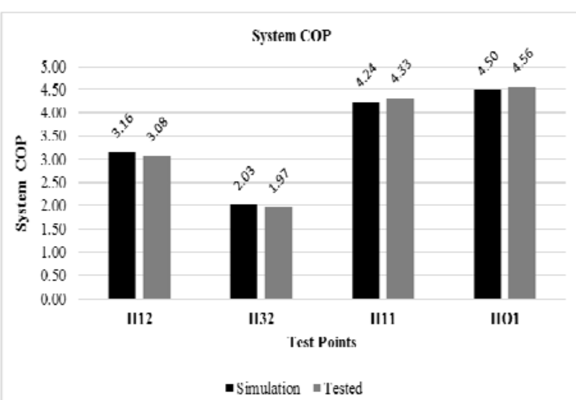

Figure 7: System Efficiency

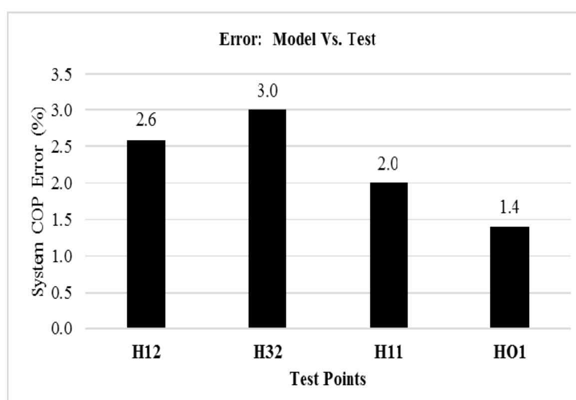

Figure 8: Prediction - Error (%)

4. IMPOROVEMENT IN PERFORMANCE

As indicated earlier we evaluated only those design changes in system that were simple to implement. These were: Variable Speed Compressor, Flow Control, Higher Efficiency Condenser and Evaporator Fans and Optimization of Refrigerant Charge. Changes in Heat Exchanger were not considered. Our investigation showed that selected design changes evaluated in this study offered significant improvements in efficiencies in cooling and heating modes. Results are summarized in Table 5.

Table 5: Improvement in Cooling and Heating Efficiencies

Description	SEER (Btu/Wh)	HSPF (Btu/Wh)
Baseline Fixed Capacity 5-ton Heat Pump in "As Received" State	14.00	8.00
System with Selected Design Changes	16.79	9.19
Improvement (%)	**19.90**	**14.90**

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 10 weeks to iteratively change and test each configuration. SDS can quickly provide an estimation of the effect of the various design options with minimal use of 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 along with excellent accuracy both for cooling and heating modes of steady-state system operation. 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

SEER	Seasonal Energy Efficiency Ratio (Btu/Wh). It is measure of system efficiency in Cooling mode
HSPF	Seasonal Energy Efficiency Ratio (Btu/Wh). It is measure of system efficiency in Cooling mode

REFERENCES

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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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