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Cost-Optimized CCHP Model Developed Using HPDM Meeting Performance Targets – FY19 1st Quarter Milestone Report

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12/31/2018

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OAK RIDGE NATIONAL LABORATORY
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Cost-Optimized CCHP Model Developed Using HPDM Meeting Performance Targets

(Regular Milestone)

Executive Summary

This report documents the equipment design and analysis work to meet the first project milestone (M1). Our industry partner, Emerson Commercial & Residential Solutions recently developed a 3-stage scroll compressor product, able to deliver capacity at three levels without using an inverter. Based on ORNL's previous cold climate heat pump (CCHP) prototype and the preliminary compressor performance data, the CCHP was redesigned for cost reduction and better performance. The 3-stage compressor, using the middle capacity level to deliver the rated capacity of the heat pump (HP), will replace the previous tandem compressors, using one single-speed compressor to deliver the rated capacity. In this way, it will reduce the compressor footprint to fit in most residential HPs' chassis. The compressor replacement increases the nominal capacity by 17%. Approximately, 30% cost reduction in the compressor and 17% reduction in the heat exchangers per nominal tonnage can be achieved relative to the prior tandem compressor design. Advanced features using an electronic expansion valve for head pressure control and an Ebm-papst variable-speed, backward-curved indoor blower can be adopted for higher efficiency.

The DOE/ORNL Heat Pump Design Model (HPDM) [2] was used to model and redesign the CCHP. Following AHRI 210/240 [1], heating and cooling performance results were predicted at the conditions for certifying a 2-stage CCHP, regarding various design scenarios. The predicted HSPF reaches 12.0, and predicted SEER reaches16.2.

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1. Background

Cold climate heat pumps (CCHP) expand the heat pump (HP) market to northern climates where heating demand is dominant. Electrically-driven CCHPs are most promising where natural gas is unavailable. They can achieve > 70% energy saving in comparison to electric resistance heating and much better economics than using tank-stored propane. A high efficiency heat pump with HSPF (heating seasonal performance factor, defined in AHRI 210/240 [1]) > 10.0 would be more efficient than gas heating in terms of source energy.

However, there are challenges for developing a cost-effective CCHP. A typical single-speed, air-source heat pump (ASHP) having an HSPF of 7.7, as shown in Figure 1, doesn't work well under cold outdoor temperature conditions typical of cold climate locations for four major reasons:

1. Too high discharge temperature: low suction pressure and high compression pressure ratio at low ambient temperatures cause significantly high compressor discharge temperatures, in excess of the maximum limit for many of the current compressors on the market. Furthermore, system charge of a heat pump is usually optimized in cooling mode, which leads to overcharge conditions in heating mode, further increasing the discharge temperature.
2. Insufficient heating capacity if sized to meet the building design cooling load: heating capacity of a single-speed heat pump decreases with ambient temperature. As illustrated in Figure 1, the heating capacity at -13°F (-25°C) typically decreases to 20% to 40% of the rated heating capacity at 47°F (8.3°C) (~equivalent to the rated cooling capacity at 95°F (35°C)). As such, a single-speed heat pump, typically sized to match the building design cooling load, is not able to provide adequate heating capacity to match the building heating load at low ambient temperatures, and supplemental resistance heat has to be used.
3. Significant cyclic loss if sized to meet the building design heating load: if a single-speed heat pump is sized to meet the heating load, it will be significantly oversized relative to the cooling load in many cold climates. This will cause excessive on/off cyclic loss during the cooling operation and heating operation at moderately low ambient temperatures. Thus, capacity modulation capability, e.g. using a variable-speed or multi-speed compressor, is necessary for a CCHP, which uses its full capacity to meet the peak heating load and partial capacity to meet the cooling and part-load heating loads.
4. Low COP: heating COP degrades significantly at low ambient temperatures, due to the elevated temperature difference between the source side and demand side.

A target CCHP should be sized to match the building design heating load while minimizing the cyclic loss for cooling operation and heating operation at moderately low ambient temperatures.

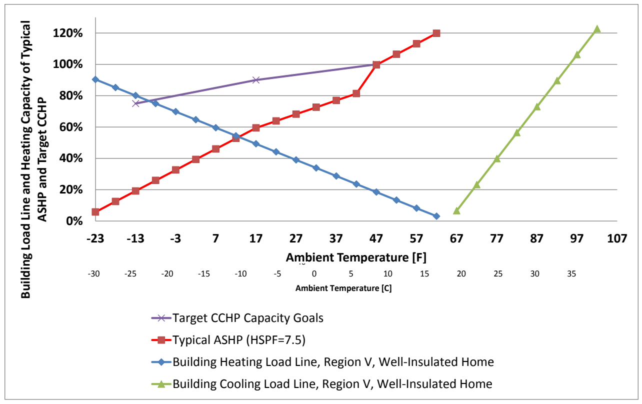

Figure 1: Building heating load in Region V (DHRmin) compared to heating capacity of a typical ASHP with 7.5 HSPF and a target CCHP having equivalent nominal heating capacity at 47°F.

2. Pervious ORNL CCHP Development

For the CCHP development, cost-effective solutions should be identified to address the above issues. From 2010 to 2015, ORNL successfully developed a CCHP prototype, which uses two identical scroll compressors in parallel (tandem). The system configuration is shown in Figure 2. The design considerations are summarized as below:

1) Most CCHPs on the market use premium variable-speed (VS) compressors and inverters. All VS HPs require special controls and thermostats made by individual OEMs (original equipment manufacturers), which increases the cost further. We aimed to develop more cost-effective CCHPs using multi-stage (>= 2) compressor(s) eliminating the need for an inverter and reducing the cost of controls. The multi-stage HPs will directly work with off-the-shelf 2-stage thermostats widely available on the market.
2) The two equally-sized, single-speed compressors were obtained from Emerson Commercial & Residential Solutions. These featured special "heating application" design features, which allow the compressor to operate at higher discharge temperatures than most typical compressors (up to 280°F [137.8°C] compared to 230°F [110°C]). This enables the heat pump to work at extremely low ambient temperatures.
3) The heat pump operates a single compressor to meet the building cooling load and operates both the compressors to meet the heating load at low ambient temperatures. Current two-stage heat pumps on the market use a single, two-stage compressor having a displacement volume split ratio of 100% to 67%. In comparison, the tandem compressors have a volume split ratio of 100% to 50%, which provides a larger over-capacity potential, when the heat pump nominal COP and capacity ratings are established for the low capacity (e.g. one compressor) level. That is the reason that the heat pump using the tandem compressors reached >75% capacity at -13°F (-25°C).

4) The CCHP is sized to match a 3-ton building cooling load using a single compressor. The system uses heat exchangers of a typical 4.5-ton heat pump. When running a single compressor in cooling mode and at moderate temperatures in heating mode, the heat exchangers are unloaded, and this provides higher efficiency. That is the key that enabled the CCHP lab prototypes to reach a COP > 4.0 at 47°F (8.3°C).
5) For CCHPs, the compressor(s) shall be well insulated and placed outside the outdoor air flow stream, as to minimize the shell heat loss. Insulating the compressors impairs the cooling performance; however, this effect is negligible, since the condenser (outdoor heat exchanger) is now oversized for cooling mode operation with only one compressor.
6) Heating mode discharge pressure control, which uses an EXV, coupled with a suction line accumulator, is intended to optimize the active charge in the system while pursuing optimum discharge pressure target as a function of the ambient temperature and compressor stage, over an extensive operation range. This also mitigates the typical charge imbalance problem between cooling and heating modes. A standard thermostatic expansion valve (TXV) is used for cooling mode.

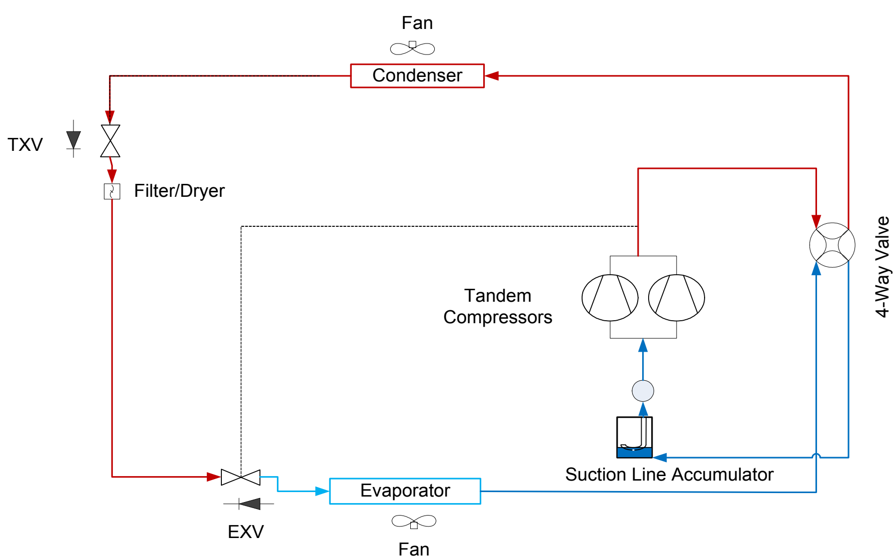

Figure 2: CCHP using tandem, single-speed compressors and an EXV for discharge pressure control in heating mode. The tandem compressors will be replaced by a single 3-stage compressor in the new design.

The lab prototype using the tandem single-speed compressors reached 4.24 COP at 47°F (8.3°C); 76% heating capacity and 1.9 COP at -13°F (-25°C), and 2.9 COP at 17°F(-8.3°C), having a rated HSPF of 11.2. Using the same tandem compressors in a breadboard HP, a field investigation was conducted in an occupied home in Ohio. The field HP operated successfully for 3 years. During the first heating season, the field HSPF was 10.8, and the HP was able to operate down to -13°F (-25°C) and eliminate resistance heat use. 40% energy reduction vs. previous HP with similar average monthly temperatures of 20°F (-6.7°C) was achieved. The HP maintained an acceptable comfort level through the heating season.

3. CCHP Redesign

Although ORNL's previous CCHP delivered the performance metrics required by DOE, further improvements are needed to reduce the cost and improve customers' satisfaction. Since 2015, we've received feedback from OEMs regarding the previous tandem design. They pointed out, a) The tandem compressors require too large of a footprint to be fit in some residential outdoor units; b) We used a single-speed compressor for cooling season, however, it is still desirable to allow capacity modulation to some degree for higher SEER; c) Customers have varied requests of comfort levels, and some may demand higher supply air temperatures than those designed for efficient operation, and we should provide an option of "comfort mode" to raise the supply temperature via reducing the indoor air flow rate as needed, as a result, this will sacrifice the energy efficiency for better comfort; d) Additionally, there is always room for cost reduction, for example, reduce the heat exchanger size per nominal tonnage, use a single multi-stage compressor instead of two parallel compressors, etc.

While keeping the most useful features stated in Section 2, we will redesign the CCHP having the modifications below,

1. A single 3-stage compressor versus the tandem compressors: Our industry partner, Emerson Commercial & Residential Solutions, recently developed an advanced multi-stage scroll compressor product, which varies the displacement volume at three levels, i.e. 50%, 67% and 100% without needing an inverter. We intend to use a single 3-stage compressor to replace the tandem compressors.
2. Capacity modulation strategy: the 3-stage compressor will be used to configure a 2-stage HP, which responds to a typical 2-stage thermostat. In order to allow capacity modulation in cooling mode and overcapacity at low ambient temperatures in heating mode, the 67% displacement volume is selected to deliver the nominal (rated) cooling capacity for the unit at 95°F ambient temperature, 80°F/67°F indoor dry bulb and wet bulb temperatures, as well as the rated heating capacity at 47°F/44°F ambient dry bulb and wet bulb temperatures and 70°F indoor dry bulb temperature. Thus, during the cooling operation below 90°F and heating operation at moderately low ambient, the unit can decrease the capacity to 75% of the rated value (= 50%/67%). For enhanced heating or dehumidification, the unit can deliver overcapacity to 149% (= 100%/67%). The Low (Y1) signal from a 2-stage thermostat will always call the 50% compressor capacity. On the other hand, the High (Y2) signal will activate the 67% compressor capacity at ambient temperatures above 20°F; at ambient temperatures below 20°F, the 100% compressor capacity will respond to the Y2 call.
3. Compressor sizing: we will use a 3-stage compressor having top rated capacity of 51 kBtu/hr to replace the tandem compressors having top capacity of 62 kBtu/hr. 67% capacity for rating the heat pump using

the 3-stage compressor gives 3.5-ton nominal capacity. 50% capacity for rating the heat pump using the tandem compressors leads to 3-ton nominal capacity. This will reduce the total compressor size, yet, increase the HP's nominal capacity. Overall, this can reduce the compressor cost per nominal tonnage by 30%.

4. Heat exchanger size reduction: in comparison to the previous design, the same indoor and outdoor heat exchangers will be used for a heat pump with the rated capacity of 3.5-ton instead of 3-ton, which results in 17% heat exchanger cost reduction per nominal tonnage.
5. Control the indoor air flow rate between "Economy" and "Comfort" mode: To accommodate typical duct size relative to the building design load, heat pumps tend to circulate indoor air flow rate of 350 cfm to 450 cfm per nominal tonnage. Therefore, a 3.5-ton heat pump allows its indoor air flow rate up to 1575 cfm. For the "Comfort" mode, the indoor air flow rate will be decreased to increase the supply air temperature. There are two options for the indoor air blower: 1) a 2-speed ECM (electronically commutated motor) blower (running a low air flow rate of 1200 cfm for better comfort), the ECM blower has its fan efficiency around 30%; 2) a full variable-speed blower with backward-curved impeller, made by Ebm-papst company. The Ebm-papst blower is able to control the supply air temperature more precisely, and it has higher fan efficiency around 50%.
6. Customized options for efficiency and product cost: Better efficiency always comes with higher cost, for example, using an EXV for head pressure control is more efficient but also more expensive than using a TXV to control the suction superheat degree, and an Ebm-papst blower is more efficient but also more expensive than a common ECM blower. Based on the basic features of 3-stage scroll compressor and heat exchangers, the customers can customize the expansion device and the indoor blower.

4. Predicted heating and cooling performance of the improved CCHP

Based on the preliminary 3-stage compressor performance data from the manufacturer, we used the DOE/ORNL Heat Pump Design Model [2] to design and model the improved CCHP. The simulated heating performance results are given in Table 1 for three design scenarios: a) Use a TXV for throttling and a 2-speed ECM blower in heating mode; b) Use an EXV for head pressure control and a 2-speed ECM blower; c) Use an EXV and an Ebm-papst backward-curved blower. The predicted performance results are given respectively responding to the High (Y2) and Low (Y1) calls of a 2-stage thermostat. To calculate the HSPF of a two-speed HP, AHRI 210/240 [1] requires the performance results at 47°F, 35°F and 17°F ambient and two speed levels. Under all the conditions in Table 1, the indoor air flow rate was set at 1500 CFM with 0.2

inH2O external static head. The outdoor fan uses an ECM motor which drives 3350 CFM air flow rate with 300 Watts power. It should be mentioned that the EXV will not only facilitate optimum head pressure control but also accelerate the heat pump start-up by totally shutting off the refrigerant flow at the end of one cycle, to prevent the system pressures from equalizing. As thus, it was assumed that the cyclic degradation coefficient (C_d) is 0.15 when using a TXV; the C_d can be reduced to 0.1 when using an EXV. The standard procedure of AHRI 210/240 was followed to calculate the HSPFs of the three scenarios. It can be seen that combination of the TXV and ECM blower results an HSPF (in region IV) of 11.1; use of the EXV and Ebm-papst blower leads to an HSPF of 12.0.

Table 1: Predicted heating performance of the improved CCHP

		Ambient Temp	Capacity	COP	Compressor Volume	Compressor Power	Blower Power	Supply Temp
	Speed	[F]	[Btu/hr]	[W/W]	Percent	[W]	[W]	[F]
TXV+ECM blower; assume C_d = 0.15; **HSPF = 11.1**	High (Y2)	47	42836	3.93	67%	2663	235	97.7
		35	36017	3.41	67%	2564	235	93.1
		17	37411	2.91	100%	3239	235	94.1
	Low (Y1)	62	41527	5.05	50%	1876	235	96.9
		47	33704	4.20	50%	1814	235	91.6
		35	27846	3.53	50%	1778	235	87.7
		17	20313	2.62	50%	1736	235	82.8
EXV+ECM blower; assume C_d = 0.1; **HSPF = 11.6**	High (Y2)	47	42907	3.93	67%	2680	235	97.8
		35	36495	3.43	67%	2582	235	93.5
		17	37894	2.92	100%	3271	235	94.4
	Low (Y1)	62	41687	5.04	50%	1887	235	97.0
		47	34272	4.25	50%	1826	235	92.0
		35	28951	3.65	50%	1791	235	88.4
		17	21871	2.81	50%	1749	235	83.8
EXV+Ebm-papst blower; assume C_d = 0.1; **HSPF = 12.0**	High (Y2)	47	42587	4.00	67%	2680	141	97.6
		35	36174	3.51	67%	2582	141	93.2
		17	37573	2.97	100%	3271	141	94.2
	Low (Y1)	62	41366	5.21	50%	1887	141	96.7
		47	33951	4.39	50%	1826	141	91.8
		35	28630	3.76	50%	1791	141	88.2
		17	21550	2.88	50%	1749	141	83.6

Table 2 presents predicted performance indices at the required conditions to calculate SEER for a two-speed air conditioner, at 95°F and 82°F ambient temperatures, 80°F/67°F indoor dry bulb and wet bulb temperatures. The results are given respectively responding to the Low (Y1) and High (Y2) calls of a 2-

stage thermostat. At the high stage, 67% compressor volume and 1500 CFM indoor air flow rate are used. At the low stage, 50% compressor volume and 1200 CFM indoor air flow rate are used. A TXV is used for the cooling mode, while assuming the cooling degradation coefficient C_d = 0.1. The resultant SEER with using the two-speed ECM blower is 15.6, and the SEER using the Ebm-papst blower is 16.2.

Table 2: Predicted cooling performance of the improved CCHP

		Ambient Temp	Capacity	EER	Compressor Volume	Compressor Power	Blower Power	SHR
	Unit	[F]	[Btu/hr]	[Btu/hr/W]	Percent	[W]	[W]	Percent
TXV+ECM blower; **SEER = 15.6**	High (Y2)	95	43124	12.46	67%	2926	235	77%
		82	46304	15.60	67%	2433	235	75%
	Low (Y1)	95	33440	13.09	50%	2074	180	78%
		82	35854	16.36	50%	1712	180	76%
TXV+Ebm-papst blower; **SEER = 16.2**	High (Y2)	95	43445	12.90	67%	2926	141	77%
		82	46625	16.23	67%	2431	141	75%
	Low (Y1)	95	33687	13.58	50%	2073	108	77%
		82	36100	17.04	50%	1711	108	75%

5. References

[1] AHRI 2017. ANSI/AHRI Standard 210/240-2017, "Performance Rating of Unitary Air-Conditioning and Air Source Heat Pump Equipment," Air-Conditioning, Heating, and Refrigeration Institute, Arlington, VA, USA.
[2] DOE/ORNL Heat Pump Design Model: https://hpdmflex.ornl.gov