Cold Climate Heat Pumps Using Tandem Compressors

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ABSTRACT

In cold climate zones, e.g. ASHRAE climate regions IV and V, conventional electric air-source heat pumps (ASHP) do not work well, due to high compressor discharge temperatures, large pressure ratios and inadequate heating capacities at low ambient temperatures. Consequently, significant use of auxiliary strip heating is required to meet the building heating load. We introduce innovative ASHP technologies as part of continuing efforts to eliminate auxiliary strip heat use and maximize heating COP with acceptable cost-effectiveness and reliability. These innovative ASHP were developed using tandem compressors, which are capable of augmenting heating capacity at low temperatures and maintain superior part-load operation efficiency at moderate temperatures. Two options of tandem compressors were studied; the first employs two identical, single-speed compressors, and the second employs two identical, vapor-injection compressors. The investigations were based on system modeling and laboratory evaluation. Both designs have successfully met the performance criteria. Laboratory evaluation showed that the tandem, single-speed compressor ASHP system is able to achieve heating COP = 4.2 at 47°F (8.3°C), COP = 2.9 at 17°F (-8.3°C), and 76% rated capacity and COP = 1.9 at -13°F (-25°C). This yields a HSPF = 11.0 (per AHRI 210/240). The tandem, vapor-injection ASHP is able to reach heating COP = 4.4 at 47°F (8.3°C), COP = 3.1 at 17°F (-8.3°C), and 88% rated capacity and COP = 2.0 at -13°F (-25°C). This yields a HSPF = 12.0. The system modeling and further laboratory evaluation are presented in the paper.

INTRODUCTION

As described by Khowailed et al. (2011), in the U. S., the primary target market for cold climate heat pumps is the 2.6 million U.S. homes using electric furnaces and heat pumps in the cold/very cold region, with an annual energy consumption of 0.16 quads (0.17 EJ). A high performance air-source cold climate heat pump (CCHP) would result in significant savings over current technologies (> 60% compared to strip heating). It can result in an annual primary energy savings of 0.1 Quads (0.1055 EJ) when fully deployed, which is equivalent to 5.9 million tons (5.35 million MT) of annual CO_2 emissions reduction. In cold climate areas with limited access to natural gas, conventional electric ASHPs or electric resistance furnaces can be used to provide heating. During very cold periods, the ASHPs tend to use almost as much energy as the electric furnaces due to their severe capacity loss and efficiency degradation. Presently, technical and economic barriers limit market penetration of heat pumps in cold climates. R&D efforts should be employed to overcome these barriers and develop high performance CCHPs that minimize, or even eliminate, the need for backup strip heating.

A typical single-speed heat pump having a HSPF of 7.5 (HSPF is heating seasonal performance factor [Btu/h/W], defined in AHRI 210/240), as shown in Figure 1, doesn't work well under cold outdoor temperature conditions typical of cold climate locations for three major reasons:

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1. Too high discharge temperature: low suction pressure and high pressure ratio at low ambient temperatures cause significantly high compressor discharge temperatures, in excess of the maximum limit for many 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: 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, sized to match the building 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. Low COP: heating COP degrades significantly at low ambient temperatures, due to the elevated temperature difference between the source side and demand side.

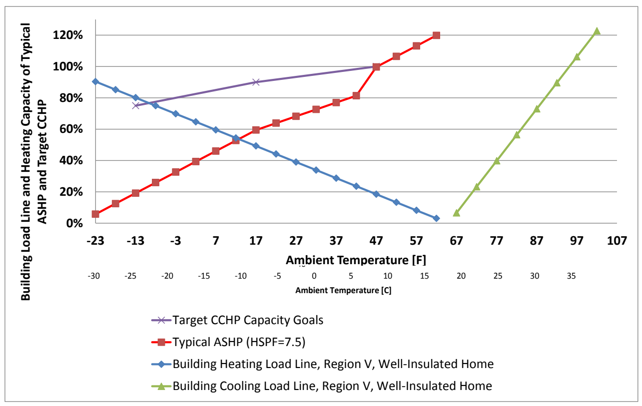
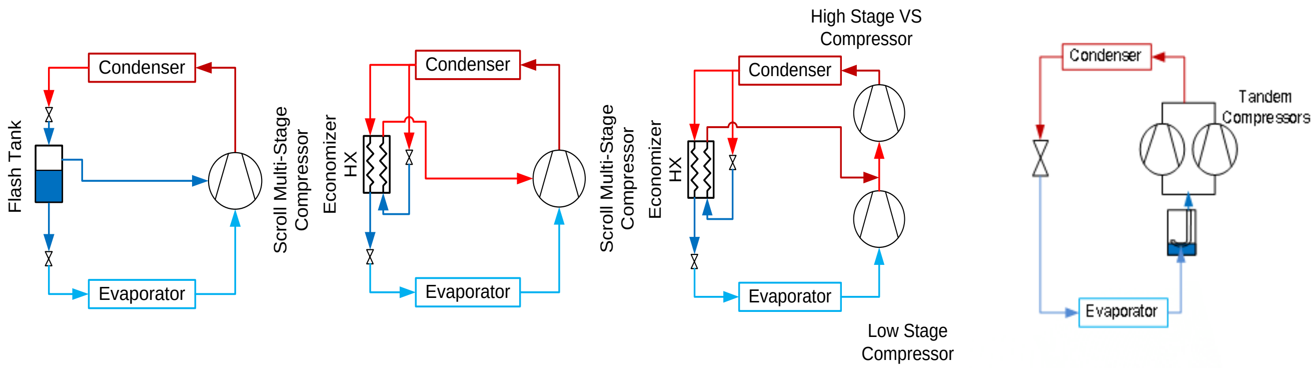


Figure 1 Building heating load in Region V (DHRmin) compared to heating capacity of a typical ASHP with 7.5 HSPF and a target CCHP having equivalent nominal heating capacity at 47°F (8.3°C).

For the CCHP development, cost-effective solutions should be identified to tackle the three issues as above. DOE's performance targets for CCHPs are 1) to matain heating capacity at -13°F (-25°C) greater than 75% of the rated heating capacity at 47°F (8.3°C), and 2) heating COP at 47°F (8.3°C) greater than 4.0. The 75% capacity criterion would result in a heat pump capacity approximately equal to the building heating load for a well-insulated home at -13°F (-25°C) in Region V (assumed to be the DHRmin load condition as defined by AHRI Standard 210/240 for Region V), where the building heating load at -13°F (-25°C) is 80% of the building cooling design load at 95°F (35°C) ambient temperature.

Researchers have investigated several cycle configurations for CCHP. Wang et al. (2009) studied advanced vapor injection (VI) cycles. A VI cycle uses a vapor injection compressor. In a VI system, liquid from the condenser is expanded to a middle stage between the condensing and evaporating pressures, after phase separation, the vapor is injected to the compressor injection port, and the liquid is further expanded and goes to the evaporator. VI cycles can be classified into two fundamental configurations: (a) Flash tank cycle and (b) Economizing heat exchanger cycle. Figure 2 (a and b) shows the schematics of a VI cycle for each configuration. In a VI cycle with flash tank, two-phase refrigerant is separated into saturated liquid and vapor by a flash tank after the first expansion. It has the advantage of feeding 100% of saturated vapor to the compressor injection port. The two-stage cycle with economizing heat exchanger allows part of the liquid refrigerant at the condenser outlet to pass through an expansion valve before entering the economizer HX to further subcool the mainstream refrigerant coming from the condenser. The superheated intermediate pressure refrigerant leaving the economizer HX enters the intermediate compressor port. As a result, the separation with economizer HX will never be 100% as compared to the flash tank separation due to the limited surface area involved. The refrigerant flow rate and pressure entering the intermediate compressor port can be easily controlled using thermostatic expansion valves. The VI cycles are able to reduce compressor discharge temperature effectively by directly injecting low enthalpy refrigerant vapor into the compressor compression cylinder. They also increase the evaporating and heating capacities due to the lower enthalpy liquid refrigerant entering the evaporator after the inter-stage phase separation.

Bertsch (2005) and Bertsch and Groll (2008) studied two-stage compression, as shown in Figure 2c, which used compressors in series, i.e. low stage compressor (booster compressor) and high stage compressor, and an inter-stage economizing heat exchanger. The two-stage compression with inter-stage economizing usually runs both stages at low ambient temperatures and only runs the high stage at moderate ambient temperatures. This configuration effectively lowers the discharge temperature and maintains a good efficiency at low ambient temperatures. It has a larger capacity modulation potential than a single VI compressor, but the cycle is more complex and auxiliary oil management is needed to ensure adequate oil return to all compressors.



(a) (b) (c) (d)

Figure 2 Candidate technologies for CCHPs

As described in Shen (2014), to augment heating capacity at low ambient temperatures, another option is to use multi-capacity compressor(s), i.e. a variable-speed (VS) compressor or put two compressors in parallel (tandem compressors, as shown in Figure 2d). Only partial capacity of the multi-capacity compressor(s) options is used to meet the building cooling load and provide heating at moderate temperatures.(such as a VS compressor at low speed, or a single compressor in a tandem set). Full capacity operation is used to boost heating capacity at low ambient temperatures. For multi-capacity compressor(s) in a single-stage system (Figure 2d) the heat exchangers must be sized adequately for full capacity operation and the discharge temperature must be managed to avoid exceeding maximum limits.

HEAT PUMP EQUIPMENT MODELING

The DOE/ORNL heat pump design model (HPDM) was used for the analytical evaluations (Shen and Rice, 2014). Ten-coefficient compressor maps (AHRI-540) were used to calculate mass flow rate and power consumption. The model also considers the actual compressor suction state to correct the map mass flow prediction. For heat exchanger modeling, a segment-by-segment modeling approach is used. Each tube segment has individual air side and refrigerant side entering states, and considers possible phase transition. An ε-NTU approach is used for heat transfer calculations within each segment. Air-side fins are simplified as equivalent annular fins. Both refrigerant and air-side heat transfer and pressure drop are considered. The heat exchanger model can simulate arbitrary tube and fin geometries and circuitries, any refrigerant side entering and exit states, maldistribution, and accept two-dimensional air side temperature, humidity and velocity local inputs. It is capable of modeling both condenser and evaporator.

A number of candidate system technologies were simulated, including a baseline single-speed heat pump, multi-capacity compressor(s) options, and vapor injection compressor option with inter-stage economizer (VI+EcHX) or flash tank (VI+FlashTank), as shown in Figure 2. The single-speed heat pump uses a compressor having a nominal cooling capacity of 5-ton/17.6 kW. The VI+EcHX and VI+FlashTank options also use a VI compressor having a nominal cooling capacity of 5-ton/17.6 kW. Two multi-capacity compressor(s) options were evaluated. One included a VS compressor, with its speed ranging from 800 to 7200 RPM and having a nominal cooling capacity of 5-ton/17.6 kW rated at 4500 RPM. The other featured a tandem compressor pair having two equal size, single-speed compressors, or two equal size, VI compressors. Each compressor in the tandem systems has a nominal cooling capacity of 2.5-ton/8.8 kW. All the compressor technologies were simulated with the same set of indoor and outdoor units, which were originally used for a 5-ton heat pump, as given in Table 1. For the system modeling in heating mode, the condenser exit subcooling degree was set at 10 R (5.6 K), i.e. assuming optimized charge control; the evaporator exit was assumed to be saturated vapor, i.e. from use of a suction line accumulator. When using a VI compressor with an economizer, the economizer exit superheat degree was set at 10 R (5.6 K) and its heat transfer effectiveness was assumed as 70%. The indoor return air temperature was always set at 70°F (21.1°C).

Table 1: Parameters of Indoor and Outdoor Units

Parameters (heating mode)	Indoor Fin-&-Tube Coil	Outdoor Fin-&-Tube Coil
Face area, ft^2 (m^2)	3.30 (0.307)	22.3 (2.07)
Total Tube Number	84	64
Number of rows	3 (cross counter-flow)	2 (cross counter-flow)
Number of parallel circuits	9	6
Fin density, fins/ft (fins/m)	168 (551)	264 (866)
	Indoor Blower (High/Low[1])	Outdoor Fan
Flow Rate, cfm (m^3/s)	1670/1380 (0.790/0.653)	3500 (1.652)
Power [W]	322/203	300

1 The indoor blower has two speed levels. For ≤50% compressor capacity or at -13°F/-25°C ambient temperature, the lower indoor air flow rate and blower power were used; otherwise, the higher air flow rate and blower power were used.

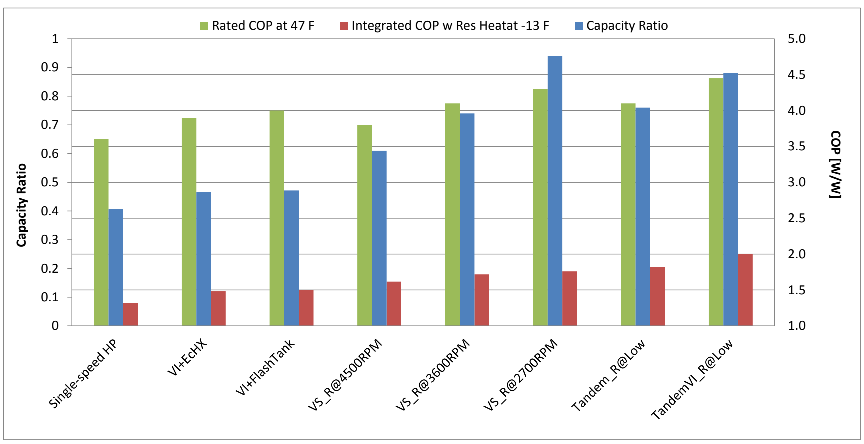


Figure 3 Ratios of heating capacity relative to 47°F, COP at 47°F and integrated COP at -13°F

The heat pump rated capacity at 47°F (8.3°C), approximately the rated cooling capacity at 95°F (35°C) is usually the value used to match a building cooling design load for the sizing selection. Regarding the multi-capacity compressor(s), VS_R@4500RPM, 3600RPM, and 2700RPM mean getting the rated heating capacity at 47°F (8.3°C) by running the VS compressor speed at 4500, 3600, and 2700 RPM, respectively. Tandem_R@Low means achieving the rated heating capacity at 47°F (8.3°C), by running a single compressor. The simulation results in Figure 3 compare heating COPs at 47°F (8.3°C), integrated COPs at -13°F (-25°C), and ratios of heat pump heating capacity at -13°F vs. 47°F rating point. The ratios of heating capacity were defined as heat pump capacity running at full speed at -13°F vs. the rated capacity at 47°F. The integrated COPs at -13°F (-25°C) were calculated by including the supplemental resistance heat needed to match 80% rated heating capacity at 47°F (8.3°C), i.e. the building heating load for a well-insulated home at -13°F (-25°C) in Region V. If no resistance heat was needed, the heat pump COP was used as the integrated COP. It can be seen that over-capacity is the key to match the 75% capacity goal at -13°F (-25°C) and provide higher integrated COP due to the elimination of resistance heat use. Four options in Figure 3 are able to reach the DOE capacity goal at -13°F (-25°C), i.e. >75% relative to the rated capacity at 47°F (8.3°C) (VS_R@4500RPM, 3600RPM, Tandem_R@Low and TandemVI_R@Low). Tandem_R@Low has a higher integrated COP than the VS options, since the VS compressor has an efficiency drop when running at the top speed of 7200 RPM. The tandem compressors with vapor injection and inter-stage economizing result in the highest integrated COP and the second highest capacity. Based on the analysis, two prototypes were selected for laboratory evaluation. First was a 'most cost-effective' design, i.e. using equal tandem, single-speed compressors with an electronic expansion valve (EXV) for discharge temperature control. The other is a 'Premium' design, i.e. using equal tandem, VI compressors with inter-stage economizing and discharge temperature control.

'MOST COST-EFFECTIVE' DESIGN - EQUAL TANDEM, SINGLE-SPEED COMPRESSORS

The 'most cost-effective' design using two equal, single-speed compressors is shown in Figure 4. The design considerations are summarized as below:

1. The two equal, single-speed compressors were provided with special "heating application" design features that allow the compressors to operate at higher discharge temperatures than most typical compressors (up to 280°F (138°C)). This enables the heat pump to operate at extremely low ambient temperatures.
2. Current two-speed heat pumps on the market use a single, two-stage compressor having a typical displacement volume split ratio of 100%/67%. In comparison, the tandem compressors have a volume split ratio of 100%/50%, which provides a larger extended-capacity potential, if the heat pump nominal COP and capacity ratings are established for one compressor. That is the reason that the heat pump using the tandem compressors can reach >75% capacity at -13°F (-25°C).
3. The CCHP is sized to match a 3-ton/10.6 kW building cooling load using a single compressor. The system uses heat exchangers of a typical 5-ton heat pump. With a single compressor running (cooling mode and moderate temperatures in heating mode), the heat exchangers are under-loaded, and this provides higher efficiency. That is the key that enabled the CCHP laboratory prototypes to reach a COP > 4.0 at 47°F (8.3°C).
4. The compressor(s) and discharge line are well insulated and placed outside the outdoor air flow stream, so as to minimize the shell heat loss, as shown in Figure 4. Insulating the compressors impairs the cooling performance; however, its effect is negligible, since the condenser (outdoor heat exchanger) has been oversized for cooling mode.
5. Heating mode discharge temperature control, which uses an EXV, coupled with a suction line accumulator, is intended to optimize the active charge in the system over an extended operating range. It mitigates the typical charge imbalance problem between cooling and heating modes. A standard thermostatic expansion valve (TXV) is used for cooling mode.

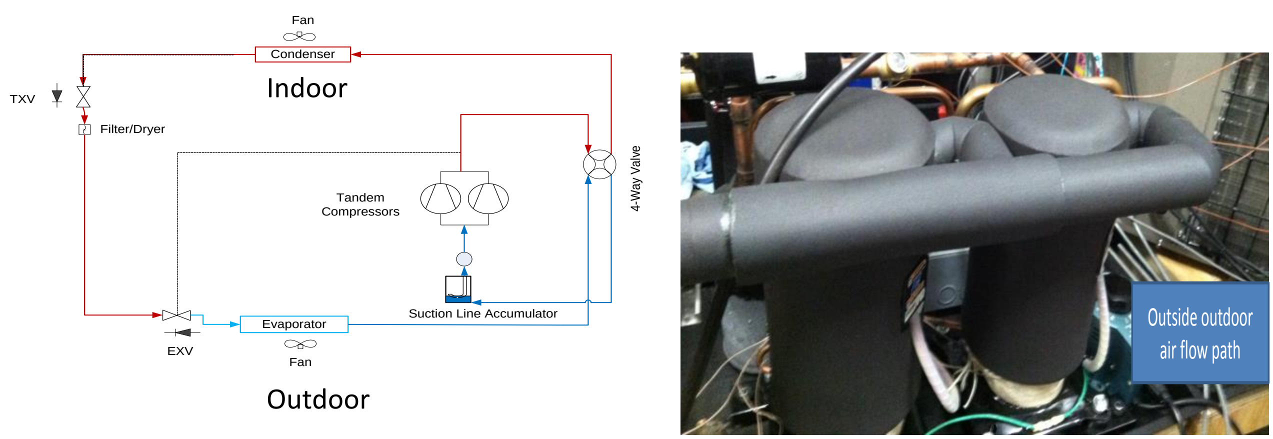


Figure 4 Left: CCHP using tandem, single-speed compressors and an EXV for discharge temperature control in heating mode, right: Insulated tandem, single-speed compressors

We tested two samples of tandem compressors 1) a pair of typical compressors optimized for cooling mode, and 2) a specially-made pair optimized for heating mode. The latter tandem pair provided better heating performance under all the operating conditions. The comparisons are given in Table 2 and 3 below. Table 2 shows measured performance indices at key ambient temperatures, i.e. 47°F (8.3°C), 17°F (-8.3°C) and -13°F (-25°C), with one or two compressors operating. Table 3 shows HSPFs (as estimated per AHRI Standard 210/240 using the measured performance results) in Region IV and V, with DHRmin and DHRmax building loads, respectively.

Table 2. Performance indices of CCHPs using tandem single-speed compressors.

	Ambient/Comp(s)	47°F, 1 Comp	17°F, 2 Comp	17°F, 1 Comp	-13°F, 2 Comp
Optimized for cooling mode	COP [-]	4.09	2.76	2.89	1.85
	Capacity, kBtu/h (kW)	37.96 (11.13)	50.46 (14.79)	25.86 (7.58)	30.04 (8.80)
	Capacity Ratio to 47°F	100%	133%	68%	79%
	Discharge Temp, °F (°C)	122 (50.0)	183 (83.9)	131 (55.0)	257 (125)
Optimized for heating mode	COP [-]	4.24	2.80	2.97	1.94
	Capacity, kBtu/h (kW)	39.72 (11.64)	50.92 (14.92)	25.92 (7.60)	30.25 (8.86)
	Capacity Ratio to 47°F	100%	128%	65%	76%
	Discharge Temp, °F (°C)	124 (51.1)	181 (82.8)	124 (51.1)	213 (100.6)
%COP Increment		3.7%	1.4%	2.8%	4.9%

Table 3. Heating Seasonal Performance Factors of CCHPs using tandem single-speed and VI compressors.

Case	HSPF/cooling optimized	HSPF/heating optimized	HSPF/tandem VI compressors
		Heating Season Ratings, Region: IV	
Based on DHRmin	11.04	11.21	11.84
Based on DHRmax	10.90	10.95	11.80
		Heating Season Ratings, Region: V	
Based on DHRmin	9.90	10.03	10.68
Based on DHRmax	9.51	9.59	10.10

'PREMIUM' DESIGN - EQUAL TANDEM, VAPOR INJECTION COMPRESSORS

Use of tandem VI compressors resulted in increases in both the heating capacity and efficiency. We tested a sample of tandem VI compressors in the same breadboard unit as the 'most cost-effective' configuration. The tandem VI compressors were coupled with an inter-stage flash tank and investigated in three scenarios. The first used a TXV to control the evaporator exit superheat; the second (Figure 5) used an EXV to control the compressor discharge temperature; the third coupled the discharge temperature control with a suction line heat exchanger (SLHX).

It was observed that using an EXV for discharge temperature control led to better performance than using a TXV for compressor suction superheat control. We tested the CCHP using the tandem VI compressors, with and without the SLHX, over extensive ambient temperatures. The SLHX addition didn't show any positive effects on the heat pump COPs and heating capacities. It was observed to increase the compressor suction superheat degree and discharge temperature, which increased the heating capacity per unit refrigerant mass flow rate. However, the increased suction superheat also decreased the suction density, and reduced the compressor mass flow rate. In addition, the compressor efficiency was found to decrease due to elevated suction and discharge temperatures. Consequently, neither capacity nor efficiency gain was observed with the SLHX. Therefore, this feature was not selected for the final design. The final system configuration, having the tandem VI compressors and discharge temperature control, achieved 5% better COPs than the tandem, single-speed compressors (optimized for heating mode) at various ambient conditions. It achieved 88% capacity and 2.0 COP at -13°F (-25°C), 4.4 COP and 40 kBtu/h (11.7 kW) rated capacity at 47°F (8.3°C). When delivering 90% capacity at 17°F (-8.3°C), it achieved 3.1 COP. Figure 6 compares the heating capacities of the tandem single-speed compressors and the tandem VI compressors, as a function of the ambient temperature. Figure 7 compares the heating COPs. Table 3 reports the lab-measured HSPFs of the system using the tandem VI compressors and discharge temperature control.

CONCLUSION

Based on the system modeling and laboratory investigations, two CCHP system configurations are recommended. One is a 'most cost-effective' design using tandem, single-speed compressors (optimized for heating performance), and the other is a 'premium' configuration using tandem VI compressors. Both configurations achieved the CCHP performance targets.

Due to the significant heating capacity reduction of a typical single-speed ASHP, a properly sized ASHP to match a building cooling design load is inadequate for the building heating load under extremely low ambient temperature conditions. Consequently, multi-capacity compressor(s) are needed to provide proper load matching for both cooling and heating seasons, i.e. using partial capacity to match the building cooling design load, and using the full capacity to match the building heating load at low ambient temperatures. This facilitates a good balance between reducing the cyclic loss and eliminating the supplemental resistance heat use. Among the multi-capacity compressor(s), using the tandem single-speed compressors is a more cost-effective option than using the VS compressors, since the tandem compressors are less expensive and do not need an inverter. In addition, the tandem compressors have a simpler control and no need to be equipped with variable-speed, indoor and outdoor fans, and a specially-made thermostat. A CCHP requires its compressor(s) to work at quite high discharge temperatures, necessitating discharge temperature management (e.g., using an EXV for discharge temperature control, or optimizing the charge for heating mode). VI cycles are able to lower the discharge temperature effectively; however, a single VI compressor can't reach the 75% capacity goal at -13°F (-25°C). Therefore, the tandem VI compressors were used to facilitate the capacity goal, and achieve the highest COPs, albeit with increased cycle complexity and cost.

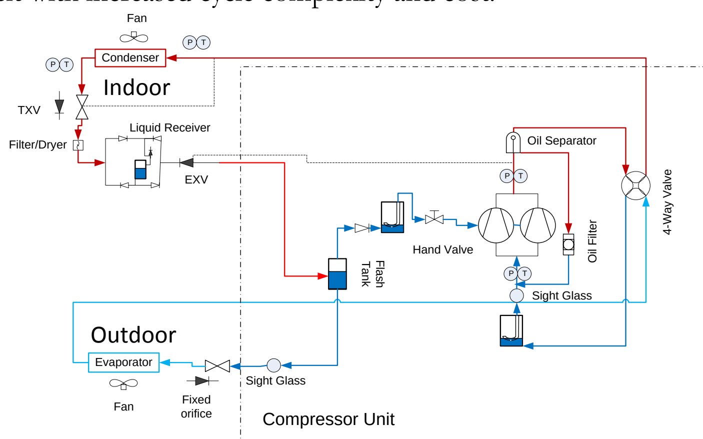


Figure 5 CCHP using tandem VI compressors and an EXV for discharge temperature control in heating mode

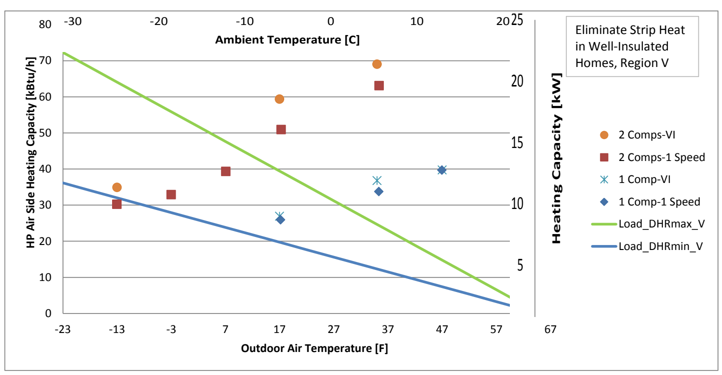


Figure 6 Heating capacity vs. ambient temperature, for tandem single-speed compressors and tandem VI compressors

As compared to a typical, single-speed, 7.5 HSPF heat pump, the option using tandem single-speed compressors will use one more identical compressor, and enlarge the indoor and outdoor heat exchangers' surface area by 60%. Since the heat exchangers are oversized for cooling mode, they lead to better cooling performance than the 7.5 HSPF heat pump, i.e. 14.0 SEER. At start-up moment in heating mode, one compressor will be turned on first. If it fails to match the temperature setting, the second compressor will be used to augment the heating capacity. Since the two compressors will never be started simultaneously, it won't increase the in-rush current significantly on the power distribution facility.

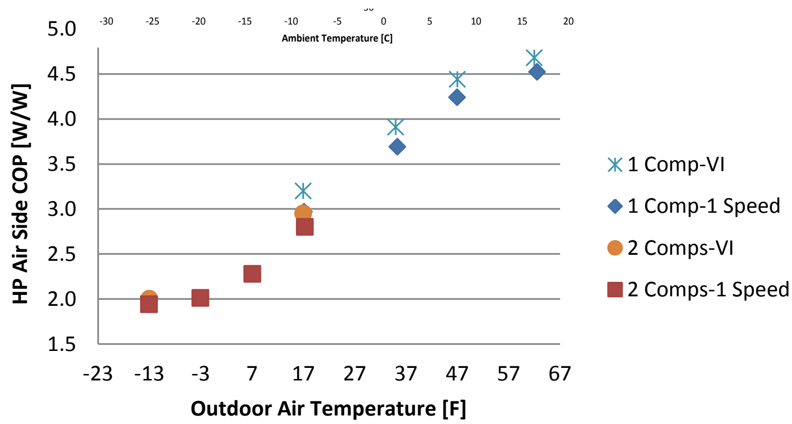


Figure 7 Heating COP vs. ambient temperature, for tandem single-speed compressors and tandem VI compressors

ACKNOWLEDGMENTS

The authors thank Mr. Antonio Bouza, Technology Development Manager for HVAC, WH, and Appliances, Emerging Technologies Program, Buildings Technology Office at the U.S. Department of Energy for supporting this research project.

DISCLAIMERS

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