DEVELOPMENT OF A COLD CLIMATE HEAT PUMP USING TWO-STAGE COMPRESSION

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ABSTRACT

This paper presents a simulation study using the ORNL Heat Pump Design Model to investigate the effectiveness of two-stage economizing cycle for cold climate heat pump applications that uses two variable-speed compressors. The high stage compressor was modelled using a compressor map, while the performance of the low stage compressor was evaluated experimentally using calorimeter testing. A single-stage heat pump system was also modelled as the baseline case. The simulated performance of the two-stage system was compared with the single-stage baseline system. Special considerations for designing a cold climate heat pump were addressed at both the system and component levels.

1. INTRODUCTION

In the U. S., there are approximately 4.4 million dwellings that use electricity for heating in very cold regions 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 (> 70% 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. For cold climate heat pumps (CCHP), the primary market segment consists of existing and new residential buildings in cold climate regions using electricity as the primary heating source. Electric resistance furnaces are used in cold climates with limited access to natural gas to provide heating as the result of the severe capacity loss and energy performance degradation experienced by conventional heat pumps in extreme ambient conditions. Resistance heating is limited to a maximum COP of 1.0. CCHPs have the capability to provide significantly higher energy efficiency. Presently, current technical and economic barriers limit market penetration of heat pumps in cold climates. R&D efforts can be employed to overcome these barriers and develop high performance CCHPs that minimize, or even eliminate, the need for backup strip heating.

For developing CCHPs, the major R&D challenges are to limit the heat pump heating capacity degradation and enhance the operation efficiency as much as possible at extremely low ambient temperatures. The heating capacity of conventional air source heat pumps (ASHP) using single-speed compressors decreases drastically as the ambient temperature drops, while the building heating demand increases. Consequently they cannot match building heating demand at extremely low ambient temperatures. Therefore, significant supplemental resistance heat has to be used, which decreases the heat pump annual operation efficiency. In addition, due to the high pressure ratio and low suction pressure at low ambient temperatures, the compressor discharge temperature could exceed maximum compressor operating limits thus limiting the working range of the heat pump [Bertsch et al. (2005)]. Also, the significantly high pressure ratio of a single-speed compressor running at extremely low ambient temperatures degrades the compressor efficiencies dramatically.

It is difficult to size a CCHP properly - if the rated heating capacity matches house load at low ambient temperatures, excessive cycling and efficiency losses occur at moderate ambient temperatures. Shen (2014) pointed out that using multi-capacity compressor(s) is necessary for CCHPs, for example, using a variable-speed compressor or parallel compressors in one system. The system operates at partial capacity of the compressor(s) at moderately low ambient temperatures, for example 8.3°C; and at higher or full capacity to meet the building load at extremely low ambient temperatures. Although the use of multi-capacity compressor(s) in a single-stage system can provide sufficient heating capacity, it does not enhance the compression efficiency at high pressure ratio where high compressor discharge temperature limits its working range.

Researchers at Purdue University identified a number of cycle concepts that could be useful for ASHPs in cold climate applications (Bertsch et al., 2006). Of these, three were seen to have the highest relative efficiency and relative heat output with low or acceptable discharge temperatures (see Figure 1). These were selected for detailed comparison in this study – the two-stage using an intercooler, the two-stage using an economizer, and the cascade cycle. System simulation models using Engineering Equation Solver (EES) were created for each of the three technologies to simulate the heating capacity and performance comparison. The heating air supply temperature for each was fixed to 50°C. Both the intercooler and economizer cycles showed similar performance at temperatures above 0°C, while the cascade cycle performed relatively better at colder temperatures. All cycles showed COPs above 2 at the low outdoor temperatures indicating reasonably good efficiency during these extremes. All three cycles had similar capacities at the low ambient temperatures. The only difference between these technologies was noticed at the warmer ambient temperatures. The cascade cycle COP is considerably lower than that of the economizer and intercooler cycles, and this is most likely due to the sizing selected for the high-stage cycle. Bertsch et al. (2006) assumed that the cascade cycle has an additional outdoor HX to allow for the high side cycle to operate without the low side cycle. Overall, all three cycles were predicted to meet the heating load and that the two-stage economizer cycle would be the best choice for an ASHP in colder climates.

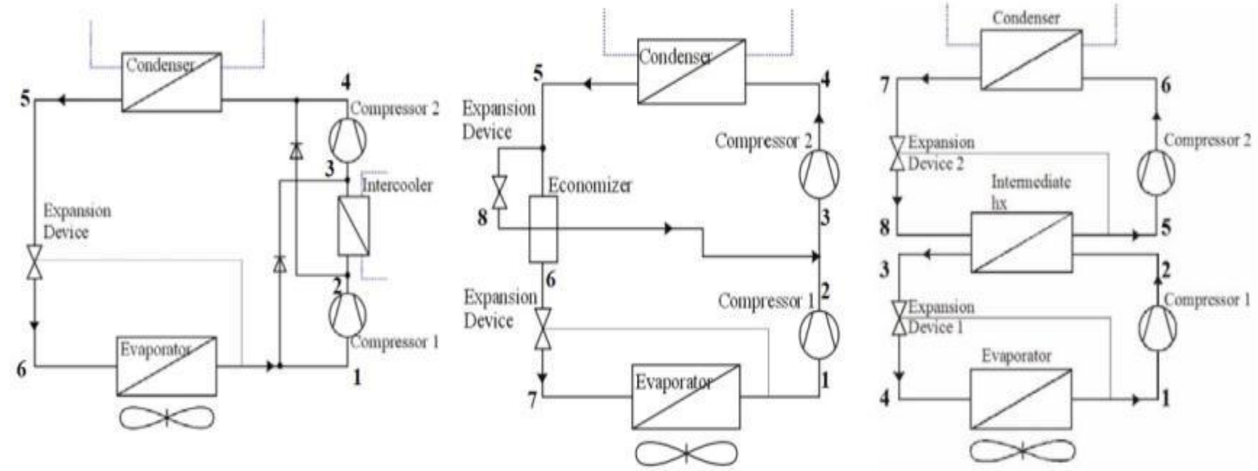

Figure 1: Heat pump cycle schematics - Intercooler (left), Economizer (middle), Cascade (right) (Bertsch et al., 2006)

Two-stage economizer cycle (middle – Fig. 1) uses two or more compressors in series, i.e. low and high stages, the configuration results in lower pressure ratios in the separate stages. An intermediate economizer separates liquid and vapour at the condenser exit, where the liquid goes to the evaporator, and the leaving vapour or two-phase flow is mixed with the discharge vapour exiting the low stage compressor(s). If an expansion device controls the superheat degree entering the high stage compressor, for example around 6 K, two-phase refrigerant will exit from the economizer's vapour port and further reduce the mixture's enthalpy entering the high stage compressor, and thus, decrease the discharge temperature effectively. Moreover, due to the phase separation by the economizer, the evaporator inlet enthalpy and quality are lower than those in a single-stage compression system at low ambient temperatures, which enlarges the evaporating capacity per volumetric flow rate driven by the low stage compressor(s). Consequently, the two-stage economizer cycle is a superior concept for CCHPs

This paper extends the analytical work of Bertsch et al. (2006) to an engineering design practice. A CCHP, having a nominal heating capacity of 17.6 kW, is investigated. The design practice is based on the system heat pump design model or HPDM (Rice et al. 2005). The CCHP uses R410A as the refrigerant since it offers a balanced choice for both the heating and cooling modes, and its compressors are readily available. The paper will fundamentally simulate the two-stage economizer cycle, and predict heating capacities and COPs at the key design conditions. Variable-speed compressors are chosen for the heat pump, in order to reduce cyclic losses at moderate temperatures and satisfy the significantly augmented building heating load at low ambient temperatures.

2. EQUIPMENT MODEL DESCRIPTIONS

The HPDM is a hardware-based, steady-state system simulation model library that covers most categories of residential and light commercial space cooling, space heating and water heating components, like compressors, heat exchangers, pumps, fans, etc. It has a component-based modelling framework that uses Newton-Raphson method to solve simultaneous system equations. The component-based modelling framework allows connecting steady-state component models in any manner. All the component models are able to calculate charge inventory, as well as heat transfer and pressure drop. Details of some of the component models used to model a CCHP are described below.

Compressor: HPDM provides multiple choices for modelling single-speed and variable-speed compressors; however, the AHRI 10-coefficient compressor map (AHRI standard 540 [ANSI/AHRI (2007)]) is the most common and accurate option. AHRI Standard 540 uses a bivariate cubic polynomial with cross-terms (Equation 1) to describe the mass flow rate and the power input as a function of saturated evaporating and condensing temperatures with a standard compressor suction superheat of 20°F (11.1 K).

$$F(S,D) = \lambda_1 + \lambda_2 S + \lambda_3 D + \lambda_4 S^2 + \lambda_5 DS + \lambda_6 D^2 + \lambda_7 S^3 + \lambda_8 DS^2 + \lambda_9 SD^2 + \lambda_{10} D^3 \quad (1)$$

Where: $\lambda_1 - \lambda_{10}$ are the map coefficients per AHRI Standard 540,
S and D are the compressor suction and discharge saturation temperatures (°F)

The compressor model simulates energy balance from inlet to outlet using the calculated power and given shell heat loss ratio; and it also considers the actual suction state to correct the map mass flow predictions. With respect to modelling variable-speed compressors, HPDM accepts multiple sets of compressor map coefficients at individual speed levels, and performs linear interpolation between neighbouring speeds to predict the compressor performance. For the working conditions where a compressor map is not available, values of volumetric efficiency, isentropic efficiency, compressor displacement volume and rotational speed can be used to model the compressor fundamentally.

Heat Exchangers: A segment-to-segment modeling approach has been used here for modelling fin-and-tube heat exchangers, where each tube segment has individual air side and refrigerant side entering states, and may have possible phase transition. Within each segment, an ε-NTU approach is used for heat transfer calculations and the air-side fin is simplified as an equivalent annular fin. Both refrigerant and air-side heat transfer and pressure drops are considered. The coil model can simulate arbitrary tube and fin geometries and circuitries and any entering and exit states of refrigerant, misdistribution, two-dimensional air side temperature, and local inputs of humidity and velocity. The segment-to-segment modelling approach is also capable of simulating the dehumidification process of water condensing on a heat exchanger coil (i.e. evaporator) by following Braun et al. (1989) methodology, where the driving potential for heat and mass transfer is the enthalpy difference between the inlet air and the saturated air at the refrigerant temperature.

It should be mentioned, effects of frost/defrosting on the outdoor evaporator were neglected, since the study focuses on steady state performance predictions and selecting compressors for a two-stage heat pump. Furthermore, at very low ambient temperatures, the humidity level is low and the frost/defrosting losses are negligible.

System Balance Points: Expansion device models usually estimate refrigerant mass flow rate, and the model can be replaced by either a given superheat value and subcooling value. In this study, the degree of superheat at the compressor suction was explicitly specified. There are two ways to specify the system balance point at the high side. One is to impose the amount of refrigerant in the equipment, and the other is to explicitly specify the condenser exit subcooling degree. Both ways were used in the simulation study.

Fans and Blowers: For a given airflow rate, the model uses the fan curve to simulate static head, power consumption, and calculate air-side temperature increment from inlet to outlet. For this study, we didn’t use a fan curve. Instead, we directly used the air flow rate and power consumption measured in the experiment.

Economizer: The intermediate economizer was modelled using Effectiveness-NTU approach to simulate the component as a whole.

3. CONSIDERATIONS AT COMPONENT LEVEL

The objective is to design a heat pump that can perform down to -25°C, and have sufficient heating capacity (>75% relative to the rated heating capacity at 8.3°C). The R410A heat pump will have a suction saturation temperature around -32°C (pressure of 252 kPa) and discharge saturation temperature around 38°C (pressure of 2289 kPa). These saturation temperatures correspond to a pressure ratio of 9.0, which exceeds the working range of most compressors. This is the reason why two-stage compression is necessary for CCHPs.

Regarding the two-stage configuration, typical AHRI compressor map conditions cover the high stage compressor's working range. However, the low stage compressor has very low condensing pressures, which could go below 28°C and beyond the testing conditions that a compressor manufacturer would normally provide. As a result, there are no compressor maps available for the low stage compressors. To overcome this barrier, a variable-speed compressor was tested in a compressor calorimeter, specifically for the working conditions of the low stage compressor. The compressor mass flow rates and power consumptions were measured with controlling the suction superheat degree and condenser subcooling degree, according to the AHRI standard. From the testing data, isentropic efficiencies and volumetric efficiencies were calculated and plotted in Figures 2 and 3, as a function of the suction saturation temperature and the ratio between the discharge pressure and suction pressure. It can be seen in Figure 2, the isentropic efficiency has a strong dependence on the pressure ratio. Around the pressure ratio of 3.0, the isentropic efficiency reaches 70%, which is a typical value for a conventional compressor at its rated condition. The isentropic efficiency contour plot indicates that the displacement volumes of the low and high stage compressors should be sized properly to maintain reasonable pressure ratios at both stages, i.e. > 2.5; otherwise, low pressure ratio will result in low isentropic efficiency, which offsets of the theoretical efficiency benefit of the two-stage compression. Figure 3 indicates that the volumetric efficiency is very high and relatively constant, approaching 100%, because the refrigerant vapour has very small specific volume at the low suction pressures and is insensitive to the heat transfer inside the compressor.

To continue the discussion on the big vapour specific volume, when changing the suction saturation temperature from 1.7°C (corresponding to 8.3°C ambient temperature) to -31.7°C (corresponding to -25°C ambient temperature), with a constant superheat degree of 6 K, the suction specific volume increases by 3.2 times. It means that the compressor needs a displacement volume 3 times as large, to drive the refrigerant mass flow rate at -25°C, as compared to 8.3°C. Certainly, the intermediate economizing reduces the refrigerant enthalpy entering the evaporator and mitigates the need for a much larger compressor displacement volume. But, increasing the displacement volume significantly at low ambient temperatures is still indispensable for a CCHP to meet the heating load.

Another consideration for CCHP is how to determine the evaporator circuit number and maintain a reasonable refrigerant pressure drop at low ambient temperatures. The frictional pressure drops in the two-phase and vapour phase regions increase linearly with the refrigerant specific volume at the same mass flux. It means that the evaporator will see much larger pressure drop at the -25°C ambient temperature than the 8.3°C ambient temperature. The refrigerant pressure drop leads to saturation temperature difference across the evaporator. The larger saturation temperature drop, the bigger reduction of log mean temperature difference in the evaporator occurs, with the same compressor suction pressure and entering air temperature. To make the situation worse, at low ambient temperatures, the saturation temperature change is more sensitive to the pressure drop. Figure 4 illustrate saturation temperature deviation changing with the evaporator pressure drop, at suction saturation temperature of -31.7°C and 0°C, respectively. One can see that a CCHP needs to split refrigerant flow to more evaporator circuits, to restrict the pressure drop and prevent the heat transfer degradation at low ambient temperatures.

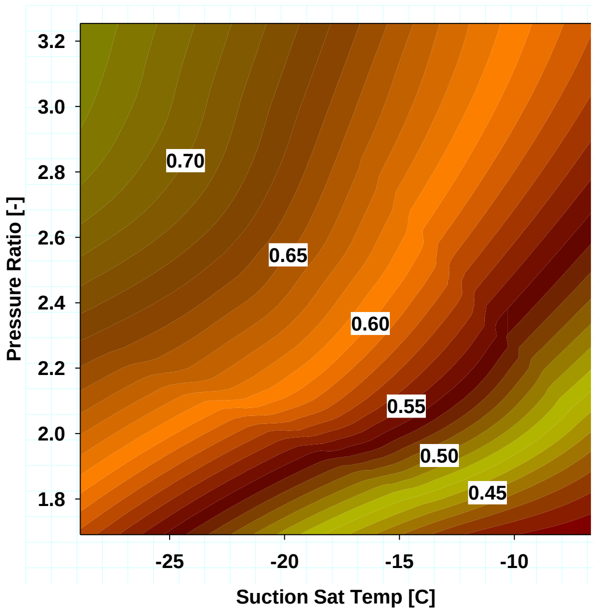

Figure 2: Isentropic efficiency plot of the low stage compressor

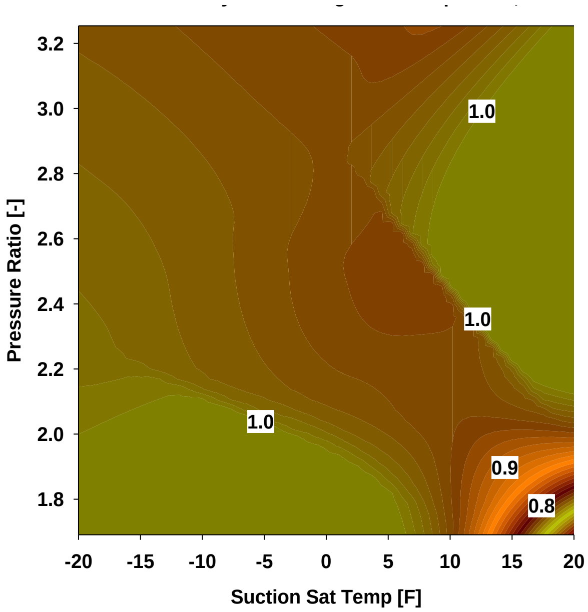

Figure 3: Volumetric efficiency plot of the low stage compressor

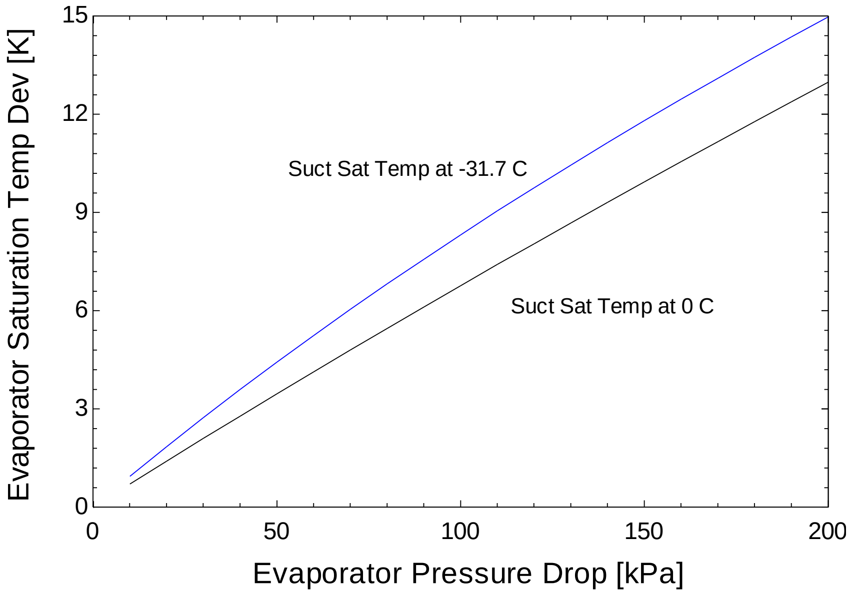

Figure 4: Evaporator Saturation Temperature Deviation Changing with Evaporator Pressure Drop at Different Suction Saturation Temperatures

4. BASELINE SINGLE-STAGE SYSTEM MODELLING

Before modelling the two-stage economizing cycle, one may want to see how a regular, single-stage ASHP performs under low ambient temperatures. A single-stage ASHP was modelled as the baseline, using a variable-speed compressor that runs at 4500 RPM at 8.3°C ambient temperature to achieve the rated heating capacity of 17.6 kW. Some component parameters of the baseline heat pump are given in Table 1.

Heat losses of refrigerant lines and compressor shell were obtained empirically by laboratory testing, all treated as linear functions of the ambient temperature. It is assumed that the compressor shell heat loss, relative to the compressor power, is 14% at 8.3°C and 40% at -25°C; the discharge line temperature drop is 7.2 K at 8.3°C and 15.6 K at -25°C; the liquid line temperature drop is 2.2 K at 8.3°C and 8.3 K at -25°C; the suction line temperature gain is 2.2 K at 8.3°C, and 5.6 K at -25°C. A TXV is used to control the evaporator exit superheat degree at 5.6 K. The system charge was optimized in the cooling mode, corresponding to a subcooling degree of 5.6 K, with the compressor running 4500 RPM at 35°C ambient temperature, and

26.7°C indoor dry bulb/19.6°C indoor wet bulb; in heating mode, the same system charge leads to 11.2 K subcooling degree with the compressor runs 4500 RPM at 8.3°C ambient temperature, and 21.1°C indoor air temperature.

Table 1: Components of Baseline Single-Stage Heat Pump

Parameters	Fin-&-Tube Condenser Coil	Fin-&-Tube Evaporator Coil
Face area [m^2]	0.307	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	551 fins/ meter	866.1 fins/ meter
	Indoor Blower	Outdoor Fan
Flow Rate [CFM]	1674	3500
Power [W]	322 (indoor blower)	300 (outdoor fan)
	Variable-Speed Compressor	
Speed Range [RPM]	800 to 7000	
Rated Speed Level [RPM]	4500 RPM at 8.3°C ambient to achieve 17.6 kW heating capacity	

System charge of a heat pump is usually optimized at the rated cooling condition. After switching to heating mode, the condenser is changed from the larger outdoor heat exchanger to the smaller indoor heat exchanger, and thus, the heat pump system is overcharged in heating mode and has much larger subcooling levels than in cooling mode. At low ambient temperatures, the baseline ASHP over-speeds to meet the building heating load, which further increases the subcooling degree and compressor discharge temperature. As illustrated in Figures 5 and 6, the compressor discharge temperature is plotted against the ambient temperature and the indoor return air temperature for two scenarios, with the compressor running at the top speed of 7000 RPM. The first scenario has a fixed system charge optimized for cooling mode, and uses a TXV to control the compressor superheat degree; the second scenario is to control a constant condenser subcooling degree of 5.6 K and use a suction line accumulator to store the extra charge, i.e. zero superheat degree entering the compressor. The case with the fixed charge has a very high discharge temperature, up to 138°C at -25°C ambient temperature and 21.1°C indoor air temperature. On the other hand, as shown in Figure 6, using the suction line accumulator for charge management and maintaining a low condenser subcooling degree is able to decrease the discharge temperature effectively.

Figure 7 compares the building heating load of a DHRmax home (having a design cooling load of 17.6 kW) to the heating capacity provided by the baseline heat pump with a suction line accumulator and constant subcooling degree. The required compressor speed for matching the building heating load is overlaid on the plot. The home location is US climate Region V, e.g. Wisconsin, Minnesota, etc. DHRmax means that the home is relatively inadequately insulated with a load profile as given by the AHRI standard 210/240. It can be seen that the required compressor speed increases with decreasing the ambient temperature, and at -5.6°C, the compressor runs at the top speed of 7000 RPM. As the ambient temperature decreases further, the heat pump heating capacity decreases and auxiliary heat has to be turned to meet the building heating load. At -25°C ambient temperature and 21.1°C indoor air temperature, the heat pump capacity is 10.4 kW, equal to 60% of the rated capacity of 17.6 kW, and the heat pump heating COP is 1.87.

5. TWO-STAGE SYSTEM MODELLING

The system model of the two-stage economizing cycle uses the same set of components as the single-stage baseline system, and the selected inter-stage economizer leads to a 5.6 K temperature approach between its exit liquid and saturation vapour, working at -25°C ambient temperature. A TXV is used to control the superheat degree entering the high stage compressor at 5.6 K; another TXV is used to control the low stage compressor suction superheat degree at 5.6 K. The tube number of the evaporator is kept the same, but the circuit number is doubled. The isentropic and volumetric efficiency of the low stage compressor were obtained from Figures 2 and 3.

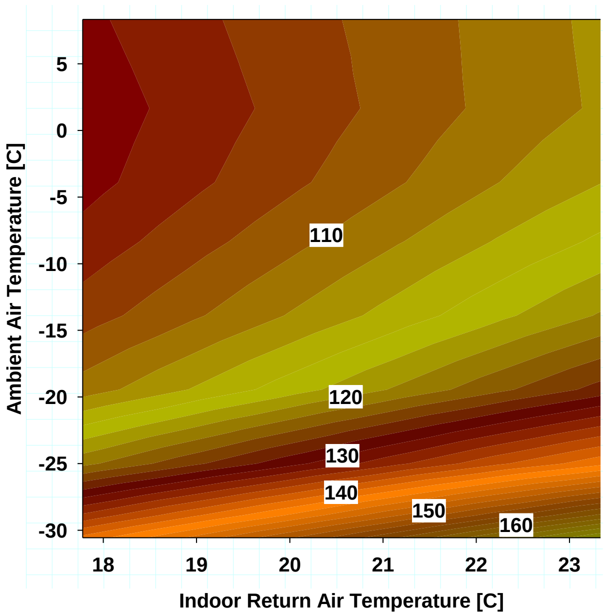

Figure 5: Compressor discharge temperature of baseline ASHP with fixed system charge and TXV control

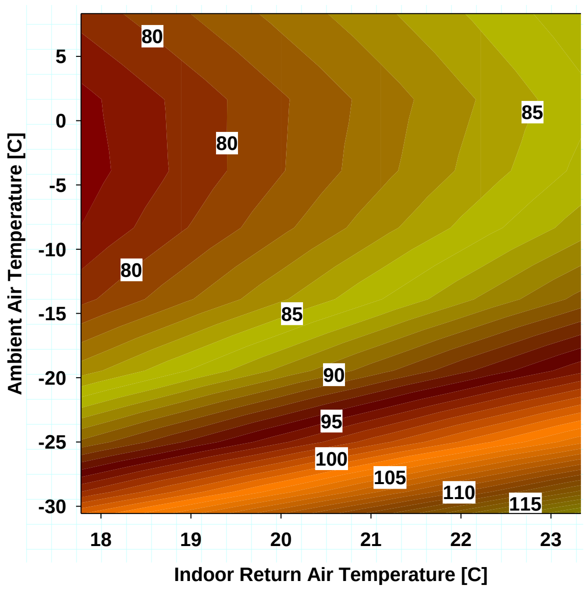

Figure 6: Compressor discharge temperature of baseline ASHP with suction line accumulator and 5.6 K constant condenser subcooling degree

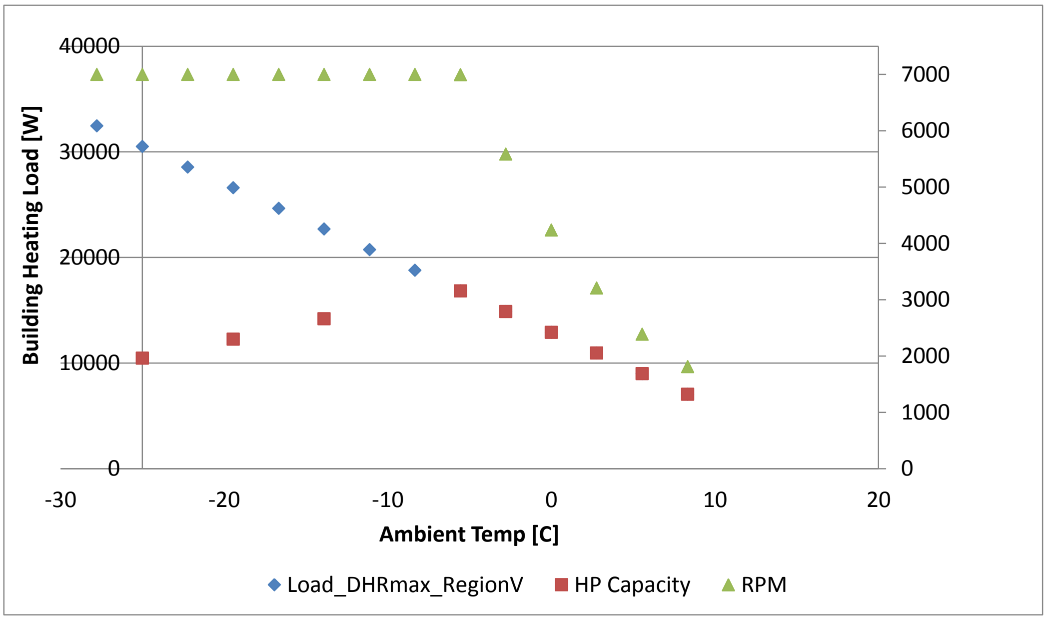

Figure 7: Comparison between Building Heating Load in Region V and HP Heating Capacity of the Baseline ASHP with Suction Line Accumulator and Constant Subcooling Degree

It is assumed that the low stage variable-speed compressor runs at the same RPM as the high stage VS compressor, i.e. using a single inverter to control the speed of both compressors. From the two-stage system performance predictions at -25°C ambient temperature and 21.1°C indoor air temperature, Figure 8 shows the resultant discharge temperature of the high stage compressor as a function of the high stage compressor RPM and the low stage compressor displacement volume (normalized to the high stage VS compressor's displacement volume). It can be seen that the two-stage economizing cycle controls a very low discharge temperature, in comparison to the baseline single-stage heat pump. Figure 9 illustrates the contours of heating capacities (relative to the rated capacity of 17.6 kW) and COPs. The two-stage compression is able to provide 60% to 140% capacity at -25°C versus at 8.3°C; the heating capacity can be augmented by increasing the compressor speed and the displacement volume of the low stage compressor. Heating COP decreases with increasing heating capacity, because higher heating capacities overload the heat exchangers. When reaching 60% heating capacity of 17.6 kW at -25°C, the two-stage system's heating COP is 2.25, about 20% higher than the baseline single-stage heat pump having the same heating capacity.

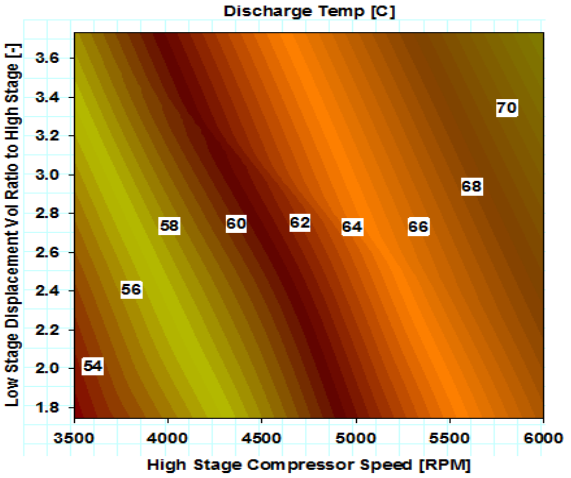

Figure 8: Compressor discharge temperature vs. speed of high stage compressor and displacement volume of low stage compressor

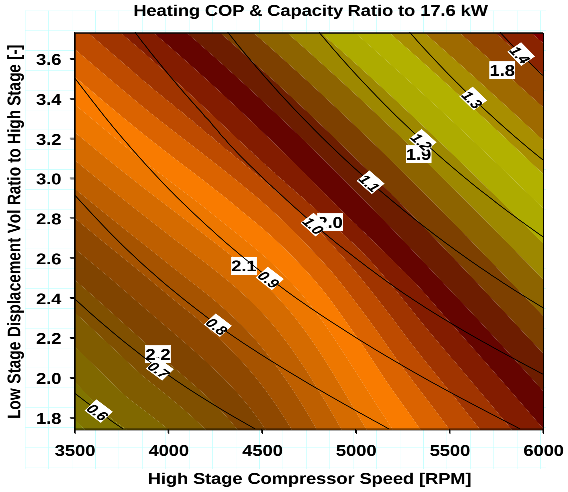

Figure 9: Heating COP and Capacity vs. speed of high stage compressor and displacement volume of low stage compressor

6. CONCLUSIONS

The system modelling reveals that application of the two-stage economizing cycle to an ASHP system can decrease compressor discharge temperature significantly compared to that of a baseline single-stage ASHP - by 40 K at -25°C ambient and 21°C indoor air temperature. This feature allows an ASHP, a larger working envelope and enables efficient operation down to much lower outdoor temperatures. The two-stage configuration provides up to 140% heating capacity and up to 20% higher efficiency than the baseline single-stage heat pump. Certainly, the advantage comes at the expense of using more compressors, complicated system configuration, and potential oil return risk for putting two compressors in series.

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8. ACKNOWLEDGEMENTS

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