Integration of Thermoelectric Modules to Vapor Compression Systems

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ABSTRACT

This paper describes the technology of a thermoelectric (TE) integrated heat pump. It uses a TE heat pump to subcool the refrigerant liquid before entering the evaporator to enlarge the evaporating capacity; and elevate the heat in the refrigerant flow to a supplemental heating coil and discharge the capacity to the indoor air flow. Using the DOE/ORNL Heat Pump Design Model, four system configurations were modelled on a baseline two-speed heat pump, the baseline heat pump with adding a supplemental coil, the TE integrated heat pump having one stage TE heating to control a 20R liquid refrigerant temperature drop; and having two stages TE heating to control up to 40R temperature drop. The system models were used to produce performance curves for EnergyPlus building energy simulations. We selected template residential buildings, poorly insulated: pre-1990 buildings, and well-insulated: IECC2021 buildings; in five cold climate zones: 5A, 5B, 6A, 6B, 7. The building energy simulations demonstrated energy saving potentials, utility cost reductions and estimated payback periods.

1. INTRODUCTION

Electricity-driven heat pumps using a vapor compression cycle are an energy-efficient solution to replace fossil fuel burning and reduce greenhouse gas emissions. However, most heat pumps being sold and installed in U.S. homes and small commercial buildings are air-source heat pumps that either meet or slightly exceed the federal minimum efficiency standards. When the outdoor air temperature drops below about –8.8°C (17°F), their heating capacity and Coefficient of Performance (COP) degrade significantly due to the temperature difference between the source side and demand sides in the cycle. The heating capacity of a single-speed heat pump at -25°C (-13°F) outdoor temperature decreases to 40% of the rated heating capacity at 8°C (47°F). Additionally, conventional heat pumps tend to be sized and optimized for cooling operation. Consequently, in the colder months of the year, they must be supplemented by another, less efficient heating source, such as electric resistance or a natural gas furnace, to provide additional warmth which increases the overall equipment and energy costs. Due to these issues, residents and building owners, particularly those living in colder climates, are reluctant to switch from natural gas, propane, or other fossil fuel furnaces to electric heat pumps for space heating. The use of conventional all-electric heat pumps also causes concern for electric utilities due to an increased winter peak demand arising from the use of supplemental electric resistance heaters. Another challenge with heat pumps stems from the fact that designing them to satisfy extreme winter heating needs could negate the part-load efficiency benefits that they may otherwise provide at moderate outside air temperatures, e.g.: >4°-15°C (40°-60°F). Since the heating load at moderate temperatures is much lower than at colder times of the year, the heat pump frequently cycles on and off, which causes low operation efficiency.

The DOE is promoting research in non-vapor-compression HVAC technologies. A potentially revolutionary non-vapor-compression mechanism to use thermoelectric technology that can use DC electricity to pump heat from a low temperature heat source to a high temperature sink via Peltier effect (Figure 1). Thermoelectric (TE) heat pumps are widely applied in the electronics cooling industry with COPs > 3.0 for 10K (18R) ΔT (Figure 2). TE elements are compact, low-cost, widely available in the market. For example, a 150 W TE module costs less than $7.01. Their recently proven ability for other heating uses such as thermoelectric clothes dryers (Patel and Gluesenkamp, 2017) and their confirmed technical feasibility for all space-heating and cooling applications for residential/commercial buildings are very promising. Hence, we believe that designing a TE based cascaded cold climate heat pump can help overcome the shortcomings of a regular heat pump.

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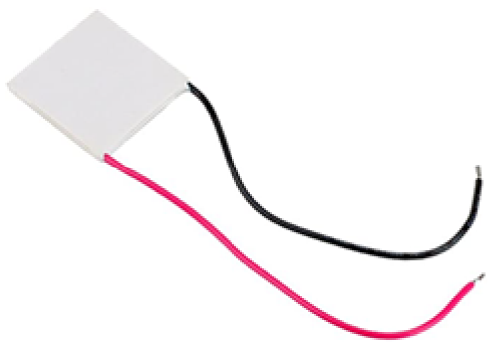

Figure 1. Thermo-electric module having a max cooling capacity of 150 Watts, Dimensions: 40mm x 40mm x 3.2mm

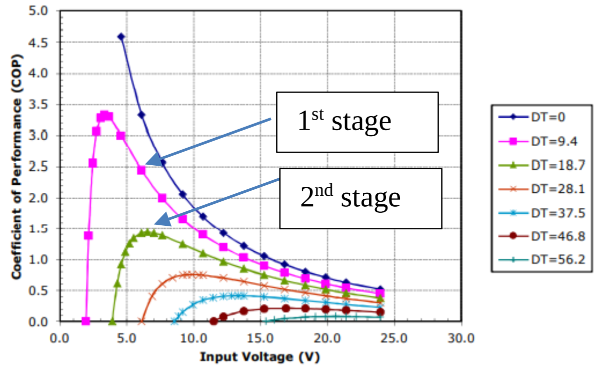

Figure 2. Cooling COP of a TE element changing with temperature difference between the hot side and cold side (Performance data from TE Technology Inc.)

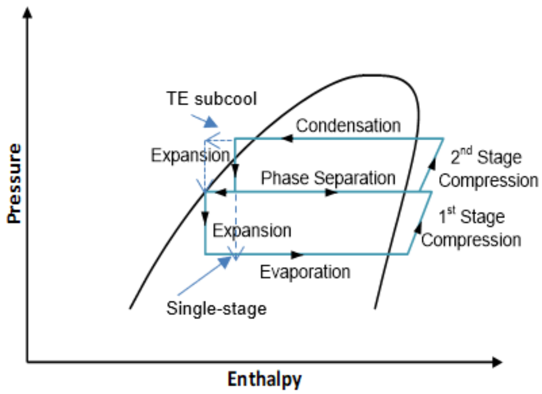

Figure 3: Pressure-enthalpy diagram depicting evaporator entering enthalpy, resulted from a single-stage heat pump, vapor-injected two-stage compression, and TE subcooler.

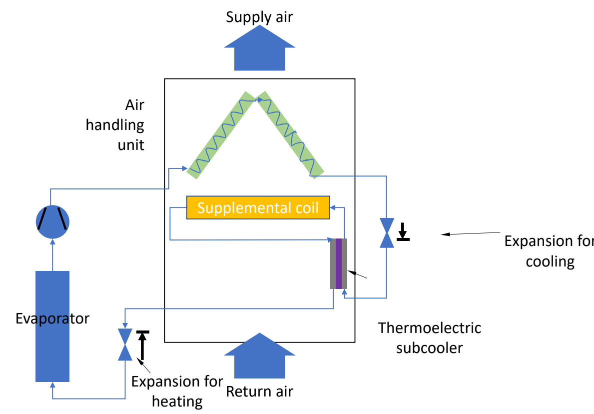

Figure 4: System view of installing a TE heat pump in the refrigerant liquid line and a supplemental (subcooler) coil in the supply duct.

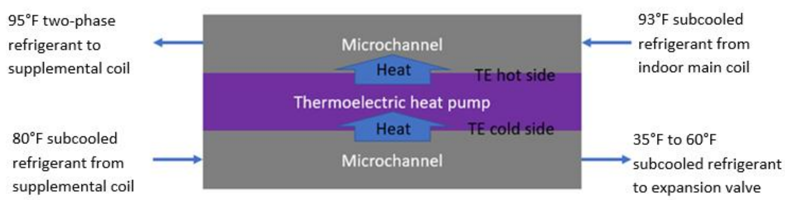

Figure 5: Detail of thermoelectric subcooler component

2. TECHNOLOGY DESCRIPTION

To improve the efficiency of a CCHP, one key is to subcool the liquid refrigerant further before entering the evaporator coil. As shown in Figure 3, a vapor injection compressor coupled with an economizer decreases the liquid refrigerant temperature by expanding part of the refrigerant liquid to an intermediate pressure/temperature to absorb heat from the liquid refrigerant. The expanded refrigerant stream comes back to the compressor injection port. The remaining refrigerant enters the evaporator with lower quality. Therefore, this process increases the overall efficiency by compressing refrigerant flow with higher volumetric evaporating capacity.

Our invention places a thermoelectric heat pump device (TE), Figure 4, inside the indoor air handling (AHU) unit of a heat pump for building space heating. This can achieve the same effect of reducing the evaporator inlet enthalpy without needing the complicated vapor-injection configuration, phase separation and associated control devices through the following innovations.

We will apply a similar structure having refrigerant flowing in both sides. The TE is powered by DC electricity. The TE assembly consists of TE modules sandwiched between microchannels, shown in Figure 5. The microchannels transport refrigerant and the heat exchange arrangement is counterflow. The TE assembly maintains good efficiency by working at a relatively low temperature lift (2-20K, or 4-40R). It accomplishes this by extracting unusable heat and boosting it to a useable temperature and location. The cold side of the TE extracts heat from subcooled (i.e. depleted) refrigerant as it leaves a supplemental heat exchanger (Figure 4, yellow coil) located downstream of a conventional heat pump's AHU coil (Figure 4, green coil). The hot side of the TE rejects heat to the refrigerant exiting the conventional heat pump's indoor condenser coil. Within the hot side of the microchannel heat exchanger, some of the refrigerant is evaporated, raising its quality. This higher enthalpy refrigerant flows to the supplemental coil to further heat the supply air. The TE refrigerant-to-refrigerant heat pump will be extremely compact by utilizing refrigerant convective heat transfer on both sides of the TE heat exchanger.

As depicted in Figure 4, the system boosts capacity due to reducing the evaporator inlet quality, thus increasing the evaporating capacity, with the same compressor size. The system clearly improves efficiency when the average heating COP of the TE heat pump exceeds the heating COP of the vapor compression system (VCS). On the other hand, when the TE heat pumps are designed to subcool the refrigerant sufficiently to reduce the COPTE below COPVCS, there can still be system efficiency benefits by reducing reliance on electric resistance backup heat (COP≤1). The integrated COP of the overall system in this case will be higher because the decreased enthalpy and quality of refrigerant entering the evaporator can enhance the COPVCS. In addition, the system is highly controllable. By varying the current supplied to the TEs, various operation points are accessible.

3. SYSTEM MODELLING

The DOE/ORNL heat pump design model (HPDM) was used for analytical evaluations (Shen, 2019). 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. It is capable of modeling both condenser and evaporator.

<u>Baseline heat pump:</u>

We selected a baseline heat pump having a 5-ton rated capacity at the 47˚F ambient running the compressor top speed. The baseline heat pump uses a two-speed scroll compressor, with a high efficiency motor. The two-speed compressor map was provided by Emerson, i.e. the compressor manufacturer. The heat exchanger and fan parameters are described in Table 1.

For the baseline system modeling in heating mode, the condenser exit subcooling degree was set at 20 R (11.2 K); the evaporator exit was assumed to be 10R (5.6 K).

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	1675/1172	3350

The outdoor fan has a constant air flow rate, 3350 CFM, and consumes a power of 250 watts. It is assumed that the indoor blower has a fan efficiency of 30%, it drives air flow having a total fan head of the total coil air side pressure drop plus a 0.2 inH2O external head. The indoor blow power = fan efficiency * indoor air flow rate [kg/s] * total fan head [pa], which consumes 216 Watts at the high flow rate, and 127 watts at the low air flow rate.

The performance maps of the baseline heat pump at the low and high stages are presented in contour plots, as below.

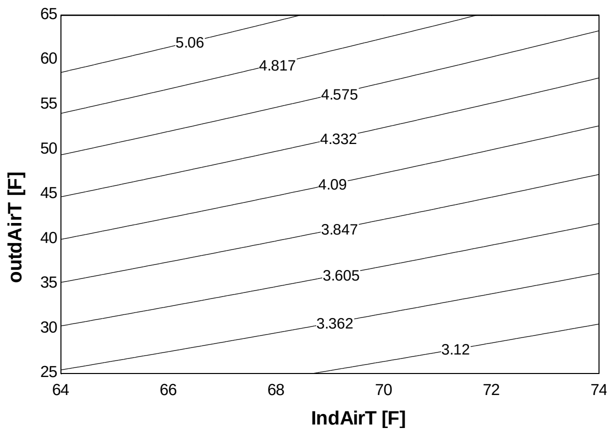

Figure 6: COP map at the compressor low stage as a function of indoor and outdoor air temperatures

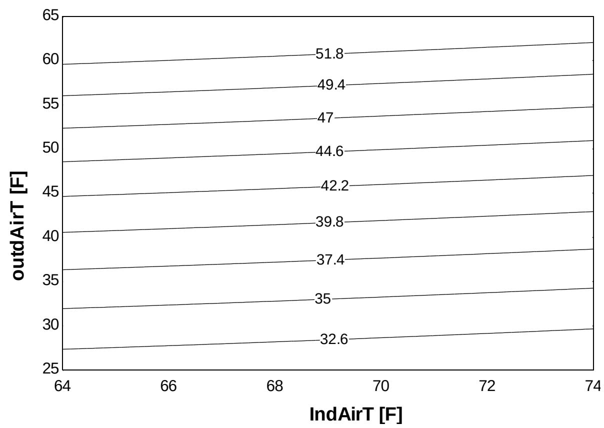

Figure 7: Capacity map at the compressor low stage as a function of indoor and outdoor air temperatures

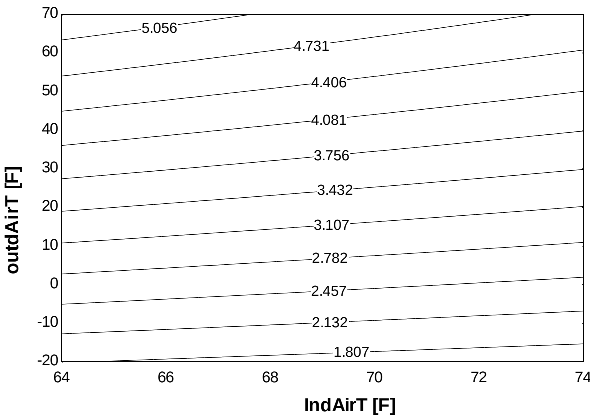

Figure 8: COP map at the compressor high stage as a function of indoor and outdoor air temperatures

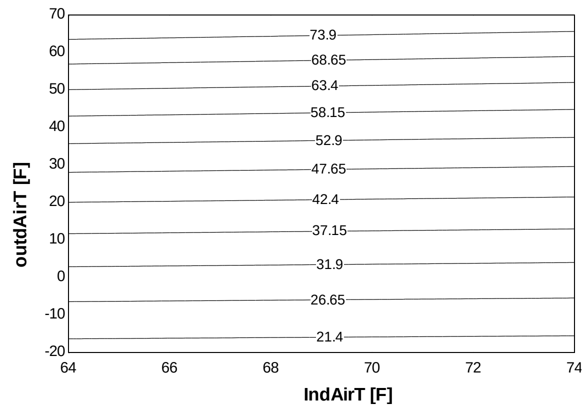

Figure 9: Capacity map at the compressor high stage as a function of indoor and outdoor air temperatures

The resultant HSPF (heating seasonal performance factor) in region IV, DHR_min building is 10.5, as the rating conditions for a two-speed heat pump required by AHRI 210/240.

<u>Baseline heat pump + Subcooler:</u>

For the TE integrated subcooler, we added a one row microchannel heat exchanger having the same frontal flow area as the baseline indoor fin-and-tube condenser. The microchannel fin density is 216 fins/ft, and tube pitch is 0.4-inch, tube width is 0.63-inch, tube length is 17 inches. It includes total 70 microchannel tubes, with each tube having 18 miniports. The microchannel coil added 60% air side flow resistance of the indoor fin-and-tube coil, increasing the blower power to 269 Watts and 151 Watts at the high and low air flow rate, respectively.

Due to the subcooler addition, a suction line accumulator is used to balance the charge in various conditions, which controls saturated vapor entering the compressor. The subcooler has a saturation liquid entrance, i.e. quality of zero. Because the subcooler augments the indoor heat exchanger surface area, it results in the liquid exit temperature about 2R higher than the return air temperature. It increases total condensing capacity and heating efficiency.

The figures below present the COP and capacity increment ratios relative to the baseline heat pump. The simulations results show, at the low stage, the subcooler increases the heating capacity from 4% to 6%, while increasing the heat pump COP up to 10%; at the high stage, the subcooler increases the capacity around 10% and increases the COP from 3% to 8%.

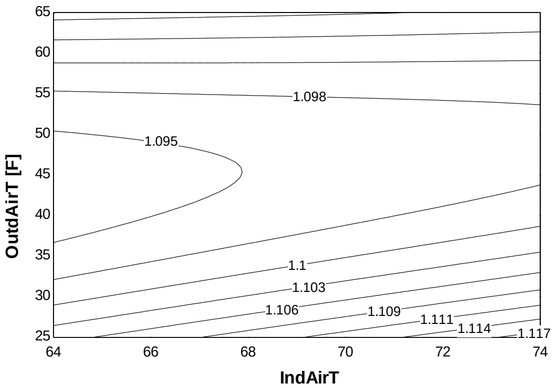

Figure 10: COP ratios at the compressor low stage as a function of indoor and outdoor air temperatures

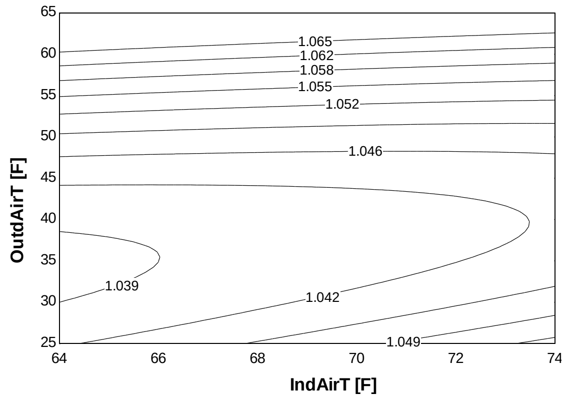

Figure 11: Capacity ratios at the compressor low stage as a function of indoor and outdoor air temperatures

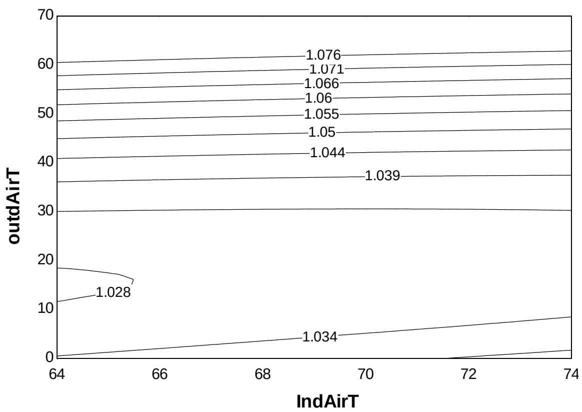

Figure 12: COP ratios at the high stage as a function of indoor and outdoor air temperatures

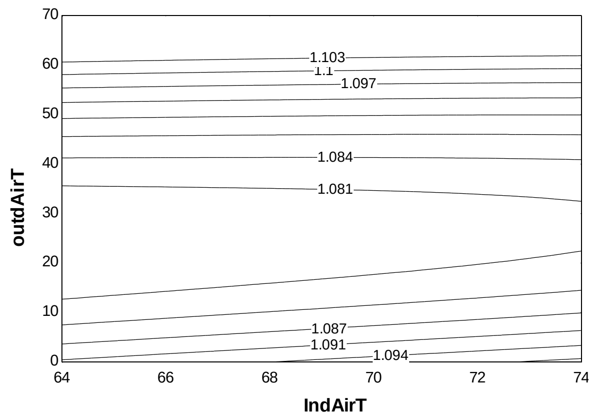

Figure 13: Capacity ratios at the high stage as a function of indoor and outdoor air temperatures

The resultant HSPF with the subcooler in region IV, DHR_min building is 11.4. The subcooler addition and the related measures increase the HSPF (minimum supplement Heat, DHR_min, Region IV, 210/240) from 10.5 to 11.4, i.e., 8.6%.

<u>Baseline heat pump + Subcooler + Thermoelectric heating:</u>
In the case that the heat pump high stage can't meet the indoor heating load, a thermostat would call supplemental heating. The thermoelectric modules will provide two more stages of heating, the first TE stage reduces the liquid line temperature by 20R, which corresponds a TE cooling COP of 3.0, as shown in Figure 2; the second TE stage would reduce the liquid line temperature by 40R, corresponding to a TE cooling COP of 1.5. Figure below illustrates the capacity and COP ratios, with turning on the TE subcooler relative to the case running only the subcooler without the TE, respectively for the 20R and 40R liquid line temperature drop. Running the TE to subcool the liquid refrigerant by 20R results in similar heating COP with an increase of heating capacity by 10%; running the TE subcooling the liquid by 40R, leads to 20% higher heating capacity, however, reduces the total heating COP by 10%. The figure below presents the capacity and COP ratios turning the TE modules relative the case with the subcooler but no TE.

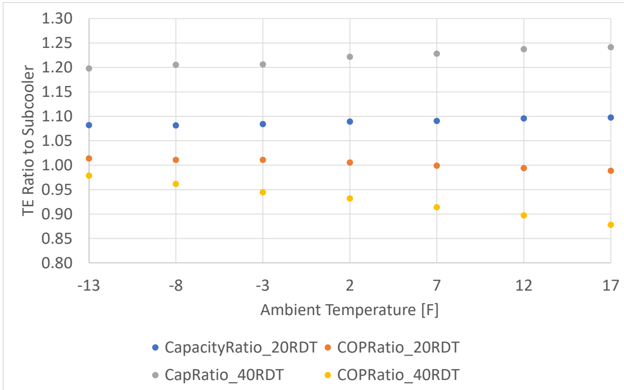

Figure 14: COP and Capacity ratios running the TE heat pump relative to no running the TE, as a function of the ambient temperature.

4. BUILDING ENERGY SIMULATION

EnergyPlus 9.5 is used to simulate the baseline and TE integrated heat pumps. EnergyPlus uses performance curves to represent heat pump operations under a wide range of ambient and indoor conditions, which are bi-quadratic curves as a function of indoor and outdoor temperatures. Part-load performance is considered by inputting degradation coefficient to a part-load correction curve. EnergyPlus can model variable-speed or multi-speed cooling and heating coils, with inputting curves, rated capacities, COPs, and air flow rates for individual speeds. We incorporated the performance curves as given by the contour plots in Figures 6 to 13, to represent the low and high stage of the baseline heat pump and the heat pump with the subcooler, separately. In addition to the subcooler performance curves, the TE stage 1, controlling a 20R liquid line temperature drop, was modelled as the third stage of the TE integrated heat pump, having the same rated COP but 10% higher rated capacity than the subcooler heat pump's high stage without the TE; the TE stage 2, controlling a 40R liquid line temperature drop, was modelled as the fourth stage of the TE integrated heat pump, having 10% lower rated COP but 20% higher rated capacity than the subcooler heat pump's high stage.

To study the heat pump types in more building types, an EnergyPlus model of the DOE prototype residential building for single family detached home was used for this study. Two scenarios were considered for building envelopes: 1) Pre-1990 house with no insulation on walls and floor and minimum insulation on attic, 2) improved windows, added insulations, and improved airtightness to meet the IECC 2021 requirements.

The whole matrix of envelope, equipment and climate zones is given in Table 3. Every combination of 4 HVAC types, 5 climate zone, and 2 building vintages were simulated for the single-family detached residential model to quantify technical savings potential of heat pumps in cold climates.

Table 3: Simulation Matrix of Equipment, Climate Zones, and Envelope Types

HVAC Types	Climate Zones (Representative Cities)	Vintages
1. Baseline 2-stage heat pump 2. Baseline heat pump + Subcooler wo TE (subcooler) 3. TE 1-stage integration controlling 20R liquid temperature drop, without the second TE stage (TE 1 + subcooler) 4. TE 2-stage integration controlling 20R and 40R liquid temperature drops, responding to the building heating load (TE 2 + subcooler)	1. 5A (Buffalo, NY) 2. 5B (Denver, CO) 3. 6A (Rochester, MN) 4. 5B (Great Falls, MT) 5. 7 (International Falls, MN)	1. Pre-1990 2. 2021

The table below shows the simulation results in pre-1990 buildings. In the table, HPCOP means seasonal heat pump heating COP, without the supplemental resistance heat; HPElec means the electricity consumed by the heat pump and TE modules. TotCOP is the total delivered heating energy (TotDeliver) divided by the total electricity consumption (TotConsump) including the supplemental resistance heat. SupRatio means the total supplemental resistance electricity consumption divided by the total heating energy delivered (TotDeliver).

Table 4: Pre-1990 Building Energy Simulation Results of Equipment, Climate Zones, and Envelope Types

HPSizing [Btu/hr]	45148.25	49066.45	46314.56	48503.48	46950.32
Baseline HP	5A	5B	6A	6B	7
HPCOP	3.34	3.25	3.14	3.21	2.98
TotCOP	2.31	2.46	1.77	2.06	1.60
SupRatio	19%	14%	36%	25%	43%
TotDeliver [kwh]	39104.55	30800.63	51583.47	43856.41	59418.49
TotConsump [kwh]	16918.11	12543.23	29118.90	21316.01	37110.28
HPElec [kwh]	9473.75	8131.12	10474.47	10185.90	11288.23
Subcooler	5A	5B	6A	6B	7
HPCOP	3.60	3.52	3.39	3.49	3.23
TotCOP	2.38	2.56	1.80	2.12	1.63
SupRatio	20%	15%	37%	26%	44%
TotDeliver [kwh]	39104.55	30800.63	51583.47	43856.40	59418.49
TotConsump [kwh]	16453.39	12050.89	28599.78	20641.50	36412.25
HPElec [kwh]	8727.07	7453.64	9617.85	9326.85	10298.23
EnergySaving	3%	4%	2%	3%	2%
Cost Saving	46.47	49.23	51.91	67.45	69.80
TE 1+Subcooler	5A	5B	6A	6B	7
HPCOP	3.55	3.47	3.33	3.44	3.17
TotCOP	2.50	2.67	1.87	2.19	1.68
SupRatio	17%	12%	33%	23%	41%
TotDeliver [kwh]	39104.54	30800.62	51630.44	43859.80	59447.76
TotConsump [kwh]	15671.39	11547.71	27577.63	20027.70	35389.37
HPElec [kwh]	9204.95	7780.10	10324.28	9759.47	11071.10
EnergySaving	7%	8%	5%	6%	5%
Cost Saving	124.67	99.55	154.13	128.83	172.09
TE 2+Subcooler	5A	5B	6A	6B	7
HPCOP	3.40	3.36	3.15	3.31	2.99
TotCOP	2.57	2.73	1.92	2.23	1.71
SupRatio	13%	10%	30%	21%	38%
TotDeliver [kwh]	39104.54	30800.62	51641.90	43860.27	59453.71
TotConsump [kwh]	15187.06	11274.93	26947.39	19687.19	34782.54
HPElec [kwh]	9971.19738	8269.11542	11490.90	10461.42	12404.69
EnergySaving	10.2%	10.1%	7.5%	7.6%	6.3%
Cost Saving	173.1	126.8	217.2	162.9	232.8

The table below shows the simulation results in IECC2021 buildings,

Table 5: IECC2021 Building Energy Simulation Results of Equipment, Climate Zones, and Envelope Types

HPSizing [Btu/hr]	24878.36	27569.87	25392.67	26865.91	25651.30
Baseline HP	5A	5B	6A	6B	7
HPCOP	3.33	3.13	3.13	3.18	2.94
TotCOP	2.15	2.38	1.67	1.98	1.51
SupRatio	23%	15%	41%	28%	49%
TotDeliver [kwh]	22528.26	15123.13	30362.59	24138.69	34735.46
TotConsump [kwh]	10468.43	6341.53	18195.63	12218.64	22935.11
HPElec [kwh]	5178.23	4125.68	5710.14	5478.46	6074.45
Subcooler	5A	5B	6A	6B	7
HPCOP	3.57	3.41	3.37	3.45	3.20
TotCOP	2.20	2.50	1.69	2.04	1.54
SupRatio	24%	15%	42%	28%	49%
TotDeliver [kwh]	22528.26	15123.13	30362.59	24138.69	34735.46
TotConsump [kwh]	10238.49	6059.09	17931.90	11848.74	22569.14
HPElec [kwh]	4774.73	3759.76	5243.03	5007.27	5539.30
EnergySaving	2%	4%	1%	3%	2%
Cost Saving	22.99	28.24	26.37	36.99	36.60
TE 1+Subcooler	5A	5B	6A	6B	7
HPCOP	3.52	3.37	3.31	3.41	3.13
TotCOP	2.31	2.59	1.75	2.10	1.58
SupRatio	21%	13%	38%	26%	46%
TotDeliver [kwh]	22530.12	15123.20	30456.60	24172.52	34869.28
TotConsump [kwh]	9741.51	5829.41	17371.62	11536.88	22095.94
HPElec [kwh]	5074.36	3914.04	5668.56	5249.28	5985.90
EnergySaving	7%	8%	5%	6%	4%
Cost Saving	72.69	51.21	82.40	68.18	83.92
TE 2+Subcooler	5A	5B	6A	6B	7
HPCOP	3.36	3.27	3.11	3.27	2.94
TotCOP	2.39	2.65	1.80	2.13	1.60
SupRatio	17%	10%	35%	24%	43%
TotDeliver [kwh]	22530.28	15123.20	30484.35	24181.35	34907.15
TotConsump [kwh]	9424.49	5698.76	16981.35	11351.52	21767.01
HPElec [kwh]	5561.73	4157.45	6393.07	5644.26	6768.40
EnergySaving	10.0%	10.1%	6.7%	7.1%	5.1%
Cost Saving	104.4	64.3	121.4	86.7	116.8

Figures 15 and 16 illustrate the electricity reduction ratios, relative to the baseline 2-speed heat pump in the pre1990 and IECC2021 buildings. Because the heat pumps are autosized to meet building design cooling loads of individual buildings, the energy saving ratios are identical between the pre-1990 and IECC 2021 buildings. In the less cold regions, e.g. 5A and 5B, the TE integrated heat pumps can save the energy up to 10%. In colder climate zones, the relative savings are less. Significant resistance heating percentages >15% are required by all these climate zones. And thus, increasing the heat pump efficiency with adding the subcooler can only save total energy from 2% to 4%.

The TE integrations, 1st-stage and 2[nd]-stage enhance the total heating capacity without noticeably impacting the total COP. It will lead to more energy reductions via reducing the supplemental resistance heat uses.

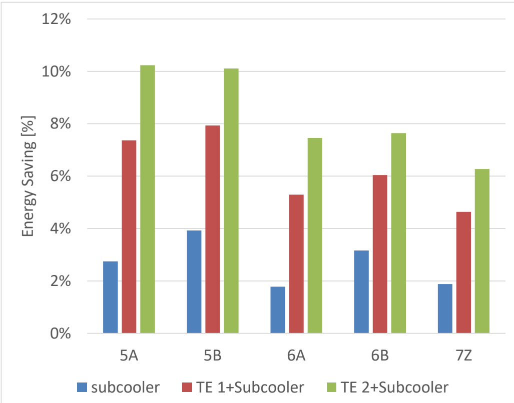

Figure 15: Electricity energy reduction ratios to the baseline heat pump in pre1990 buildings

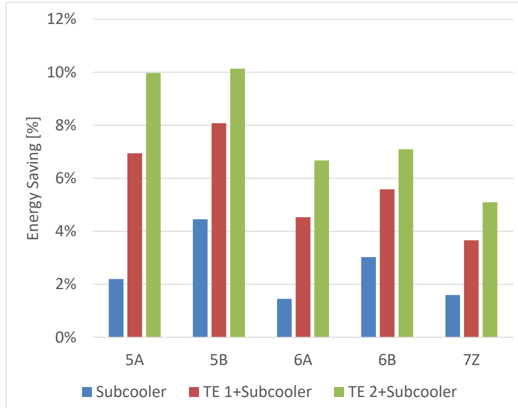

Figure 16: Electricity energy reduction ratios to the baseline heat pump in IECC2021 buildings

5. ECONOMICS ASSESSMENT

The figure below reports the TE DC power input to reduce the liquid refrigerant temperature by 20R and 40R, respectively, as a function of the ambient temperature. The required power decreases with the ambient temperature, i.e., the refrigerant mass flow rate. The 1[st] TE heating only requires 20% power input, i.e., 50 Watts per rated tonnage of the heat pump, compared to the 2[nd] stage TE heating.

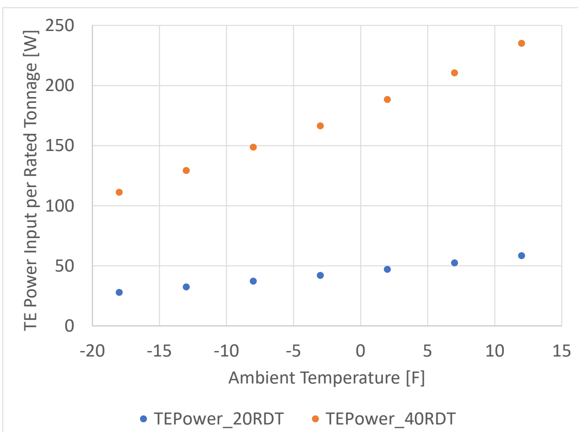

Figure 17: TE Inputs Powers Changing with Ambient Temperature.

Referring to Figure 2, to achieve a TE cooling COP of 3.0 at 20R temperature difference, it requires a DC voltage of 5 Volts. The figure below depicts the input current at various voltages and temperature differentials. At 20R and 5.4V, the input current is 2.0. It indicates that each TE module will consumes around 10 Watts. Therefore, 50 Watts DC power input per tonnage requires 5 pieces, (scale them up by compressor, 150 watts for A 3-ton compressor) to achieve the efficiency and capacity. The cost of TE modules per rated tonnage is around $35 to $40. To achieve a COP of 1.5 around 40R, it requires an input voltage around 6 V, and the related input current is 1.5 A, 250 watts input power would require 250/ (6 * 1.5) = 28 pieces, around $200 per tonnage. it is most cost-effective to pursue an optimum COP for the 1[st] stage heating and size the TE modules, which will result 70% of the total saving with 5 pieces/28 pieces = 18% of total cost increment.

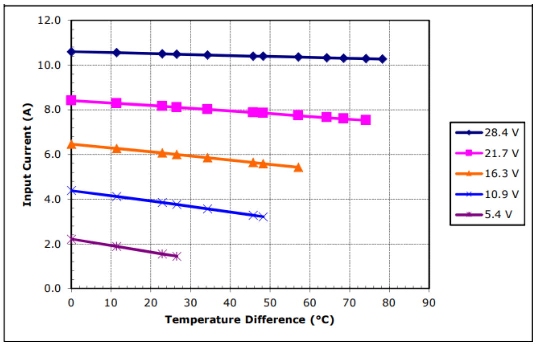

Figure 18. Input Current of a TE element changing with temperature difference and voltage (Performance data from TE Technology Inc.)

Figure 19. Electricity cost reductions per tonnage in IECC2021 buildings

Figure 19 gives the annual utility cost savings running the first stage and second-stage TE heating. If only size for 1st stage heating at 20R temperature difference, it will save $30 each year. If the total addition of the TE and subcooler per tonnage costs $150 ($40 of TE modules + $18 DC power supply + $92 supplemental heat exchanger), it will pay back within 5 years.

6. CONCLUSIONS

The TE integrated heat pump harvest energy saving from two perspectives, one is that the supplemental coil increases the condensing capacity and operation efficiency. It improves the HSPF of baseline two-stage heat pump from 10.5 to 11.4, i.e. 8.6%; the other is that the TE heating increases the total heating capacity by 10% to 20% without impacting the heating COP noticeably, which reduces the electric resistance heat use. In the cold climate zones where the annual resistance heating use larger than 15%, increase in the heat pump COP is not adequate, which only results in energy saving around 3%. The TE integrations will augment the heating capacity and result in more energy reductions. In the building energy simulations from pre1990 and IECC2021 buildings in five cold climate zones, the TE integrated heat pumps can save energy up to 10% in region 5A and 5B, which is consistent between the poor insulated and well-insulated buildings.

The TE heat pump can be staged at two levels, i.e. 1st stage to drive 20R liquid temperature differential; and 2nd stage to drive 40R temperature differential. The 1st stage TE heating is most cost-effective, which can result in 70% of the total saving with requiring only 18% of the TE modules of the 2nd stage TE heating. The annual cost savings via using the 1st stage is $30 per rated tonnage; and $50 using the 2nd stage TE heating. It is practical to achieve a payback period within 5 years.

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