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International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2016

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Cost Optimization of Thermoelectric Sub-Cooling in Air-cooled CO_2 Air Conditioners

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ABSTRACT

This paper presents a cost-effective enhancement of a trans-critical carbon dioxide (CO_2) cycle in air-conditioning mode by utilizing a thermoelectric (TE) sub-cooler. It is well-documented that the cooling COP of the transcritical CO_2 cycle decreases as the ambient air temperature significantly increases above the critical temperature of the refrigerant. A high gas cooler outlet temperature limits the enthalpy of evaporation so that the air-conditioning cooling performance is reduced. Sub-cooling is known as a mitigation method to this problem. However, adding a small-scale heat pump to a residential or light commercial air conditioner can be quite costly. Therefore, a TE solid-state sub-cooler is proposed. The TE cooling devices utilized in small temperature differences ranging from 5 to 15 °C can be quite efficient since the intrinsic heat loss of the TE by heat conduction in reverse direction of pumping heat is minimal. Based on the prior work, the optimum design for cost-per-performance shows that the cost for sub-cooling is dominated by the heat exchangers and it is not by the TE material itself. The TE modules are compact and have a low thickness, which is in the range of a few mm. Hence the TE modules can be integrated into the form factor of a plate heat exchanger. In this study, the cooling COP of the CO_2 air conditioner is enhanced by approximately 12% using an optimally designed TE sub-cooler at an ambient temperature of 30 °C. This potential improvement is based on a figure-of-merit (ZT) of currently available TE materials (ZT~1). The seasonal COP is evaluated and shows a significant climate dependency of enhancement with TE subcooling assist. The analysis of the material mass based cost against performance shows a contradictive trend between the cost and the COP, hence the optimum cost effective design must exist depending on the practical conditions.

1. INTRODUCTION

A heat pump refrigeration using transcritical carbon dioxide (R744, CO_2) is known to have a big advantage in gas cooler heat exchanger and a large enthalpy margin, hence good coefficient-of-performance (COP) is expected (Austin et al., 2012). It is also well known, however, the cooling COP of an air-source transcritical CO_2 cycle decreases as the ambient air temperature significantly increases above the critical temperature of refrigerant. A high temperature at the gas cooler outlet limits the enthalpy of evaporation. Sub-cooling with another smaller scale heat pump (Khan et al., 2000) or other methods are considered to mitigate this problem by multistage (Zubair at al., 1996), flash inter cooling (Agrawal et al., 2007), and expander (Yang at al., 2007). A mechanical heat pump can become costly as it shrinks to smaller. In previously reported studies, thermoelectric (TE) solid-state sub-coolers have been investigated for particular cases but not in genetic or even optimized (Schoenfeld et al., 2008, Radermacher et al. 2005, and Sarkar, 2013). In this study, we optimize the TE design for performance and then maximize the system performance COP with minimizing the mass (cost) of additional TE materials. Seasonal COP of the system is also investigated depending on the annual climate by location. The TE cooling devices works efficiency when they are utilized as a heat pump for a small temperature difference ranging from 5 to 15 °C. It is because the intrinsic heat loss is minimal, which occurs by the heat conduction across the TE legs in reverse direction of pumping heat.

Based on the prior work, the optimum design for cost-per-performance shows that the cost for sub-cooling is dominated by heat exchangers and the major part of cost is not by TE material itself. TE coolers are compact and have thin form factor with the thickness in range of a few mm to a centimeter. Hence the TE modules could be integrated into a plate heat exchanger. This study begins with a condition at the external air temperature of 30 ºC and the gas cooler exit temperature of 40 ºC. By taking advantage of the properties of CO_2 as refrigerant, not only the air-conditioning of residential or commercial buildings but also the datacenter cooling (Yazawa, 2015) cold be another application of this technology. We investigate a seasonal COP of air source refrigeration as the function of ambient temperature according to the local annual climate information.

The improvement of system COP by TE subcooling is evaluated based on a non-dimensional figure-of-merit (ZT value) of currently available TE materials (ZT = 1.5). We also investigate the impact of a near term potential improvement of the TE materials up to ZT = 2.2-2.5 (Biswas et al., 2012) to show what would be expected to change by the material R&D and improvement. A mass based cost of the TE module is evaluated and optimized based on the same method for TE power generators (Yazawa et al., 2011).

2. PERFORMANCE ANALYSIS

2.1 Performance with subcooling

Fig. 1 shows a study case of a CO_2 transcritical cycle with a TE subcooler. The ambient air temperature T_a is 30 ºC. The vapor compression cycle is operated under a high discharged temperature as well as high discharged pressure taking advantage of supercritical phase CO_2. The temperature difference between the air inlet and the refrigerant after gas cooler depends on the effectiveness of the condenser heat exchanger, but let us take an example of a reference (Chen et al., 2006) and consider it as a 10 ºC difference. The T-s diagram of the system is shown in Fig. 2.

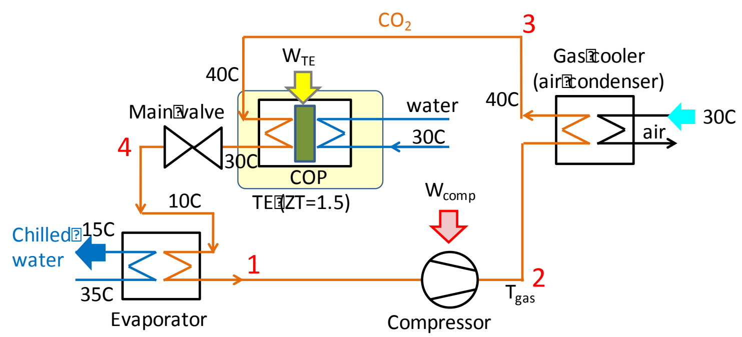

Figure 1: The cycle diagram of CO_2 cycle with TE assisted cooling.

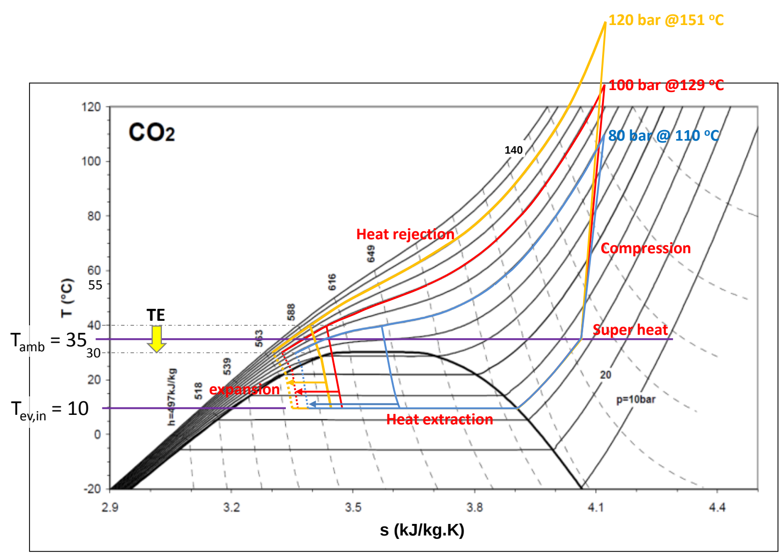

Figure 2: T-s diagram example of CO_2 cycle with and without TE assist for R744. Blue, red and yellow curves indicate the cycles of 8 MPa, 10 MPa, and 12 MPa, respectively.

2.2 Thermoelectrics modeling

TE coolers are well known for electronics cooling applications. The following model considers a single leg with assuming that the p-type and n-type material properties are symmetric. As reported by Koh et al. (2015), the design of TE cooler has to match to the thermal resistance of the external heat exchangers to obtain the maximum cooling COP. In particular, the thickness *d* of TE leg and the injecting current *I* are the key parameters to optimize. The COP is calculated as the following formula.

$$\mathrm{COP} = \frac{\psi_c SdI^3 - \left(d + 2\beta A\psi_c\left(1 + \frac{\sigma S^2}{\beta}T_h\right)\right)I^2 + 2\sigma AST_h I + 2\frac{\sigma\beta A^2}{d}(T_h - T_m)}{2((\beta A - SdI)\psi_c + d)I^2} \tag{1}$$

where, ψ_c is thermal resistance of the cold side heat sink, A is area of heat flow, T_h is the heat source side and T_m is the dissipation side reservoir temperatures, respectively. S, σ and β are the TE material properties representing Seebeck coefficient, electrical conductivity, and thermal conductivities, respectively. The dimensionless figure-of-merit of TE material is found as ZT = ($\sigma^2 S / \beta$) x T_{mean}.

2.3 Design Simulation

The main heat pump loop is modeled with Engineering Equation Solver (EES) to simulate the impact of introducing a TE module as a subcooler. The model is simply based on gas state equations according to the ideal gas law in each point of the cycle along with considering the vapor quality of the gas-liquid mixture phase. The refrigerant properties are provided in a material data library with EES including the liquid-vapor mixture as a function of the vapor quality and supercritical phase of CO_2. The following equations represent a closed vapor compression cycle. The suffix indicates the points shown in Fig.1. A constant 70% of isentropic efficiency of the compressor is used for the calculation independent of the operating condition (discharge pressure). This range of isentropic efficiency is reasonably considered for 100 kW- 10s MW scale compressors.

The enthalpy gain assuming the adiabatic process is given by real compression work W_c. The temperature T_2 at the compressor discharge is determined by the ideal gas law considering the gas property. The pressure after the condenser is maintained through the TE cooler embedded heat exchanger. The subcooled refrigerant temperature T_{gc} is given here. The energy balance of the TE subcooling is externally calculated later.

$$h_2 - h_1 = W_c \tag{2}$$

$$T_3 = T_{gc},\ P_3 = P_2 \tag{3}$$

The heat rejected Q_{gc} by condenser is found as,

$$Q_{gc} = h_3(T_3, P_3) - h_2 \tag{4}$$

If the valve is ideally functioning, the pressure after the valve consists of the pressure of the compressor inlet along with neglecting the pressure loss of the evaporator. Also, the enthalpy *h* remains through the valve.

$$P_4 = P_1,\ h_4 = h_3 \tag{5}$$

To determine the remaining enthalpy after going through the valve, the vapor quality *x* is found based on the pressure and enthalpy. The vapor quality at the evaporator inlet is found by a property of the refrigerant from the phase diagram. Then, depending on the vapor quality. Finally, the cooling capacity Q_{ev} is determined as,

$$Q_{ev} = h_1 - h_4 \quad (6)$$

The model considers an ideal cycle with the evaporator temperature 10 °C and the gas cooler outlet temperature 40 °C at a given discharge pressure. The TE cooler performance is separately calculated by an analytic model implemented in spread sheet. The TE embedded heat exchangers are assumed properly designed. TE leg thickness is a key parameter and designed to maximize the COP of the TE subcooler alone. The figure-of-merit of TE material is ZT = 1.5 (available in market). A state-of-the-art material with ZT = 2.5 (currently lab champion) is also investigated for a future potential.

The simulation results are shown in following figures as functions of discharge pressure, which is directly related to power consumption of compressor as well as the system cost. Two factors are evaluated. One is Q; the cooling capacity [kJ/kg] and another is COP = Q / W [-], while W is electrical power required for the system including the compressor and the TE. Constant isentropic efficiency is η = 70%.

Figs. 3 and 4 shows the enhancement ratio of the TE assisted system compared with the maximum performance of the baseline (standard) system in Q' / Q_{max} (Fig. 3) and COP'/COP_{max} (Fig. 4) as functions of discharge pressure, respectively. The light green curves give the normalized performance of the baseline system. Cooling performance is still the better for lower pressure. By remaining the same COP, TE assisted system can improve both the cooling performance and cost simultaneously.

If a TE material with ZT=2.5 is commercialized, the COP further increases >+20% of current maximum COP of the system compared to a currently available material (ZT=1.5). The peak is observed at around 8.5 to 9.0 MPa of discharge pressure. If the regeneration is combined with pumped heat from TE, the required pump power is reduced by ~10% so that the Q' / Q_{max} can increase >+20%. The TE waste heat recovery could be also utilized to use the high temperature before entering gas cooler heat exchanger. If this TE generator is installed, 3-4% of exergy is further recovered. If there is upper limit for discharge pressure less than 10 MPa, the improvement is significant.

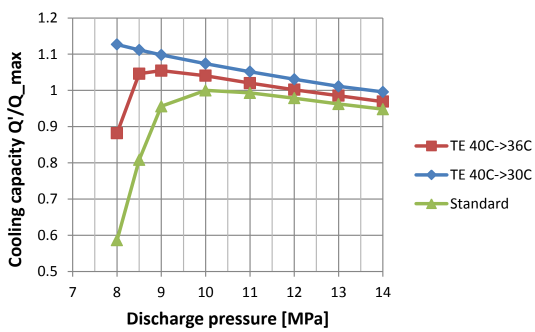

Figure 3: Q' / Q_{max}

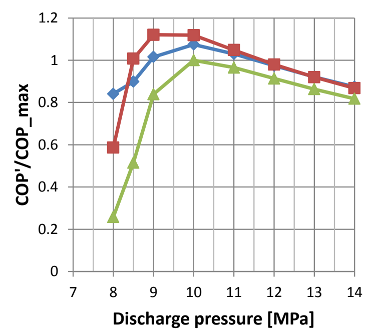

Figure 4: COP' / COPmax

Fig. 5 shows another view of performance improvement as functions of subcooling temperature ΔT. The changes in the COP of the TE heat pump alone, the system cooling capacity, and the overall cooling COP are in plot as well. This is based on a condition of 10 MPa discharge pressure. The optimum subcooling is found to be about 12 °C and the system COP reaches 2.39 while the base line COP is 1.95.

The enhanced COP is found as the following formula.

$$COP_{TE} = \frac{Q_3}{W_{TE}} \quad (7)$$

$$COP = \frac{Q_{ev}}{W_c} \quad (8)$$

The modified COP' is found as Eq. (9) by finding the improved enthalpy change at the evaporator Q'_{evap} and the Q_3 by the thermodynamic cycle simulation on EES. The W_{TE} is found by subtracting Eq. (7), hence,

$$COP' = \frac{Q'_{ev}}{W_c + W_{TE}} = \frac{Q'_{ev}}{W_c + Q_3/COP_{TE}} \tag{9}$$

where, COP_{TE} is found by the 1-D analytic model.

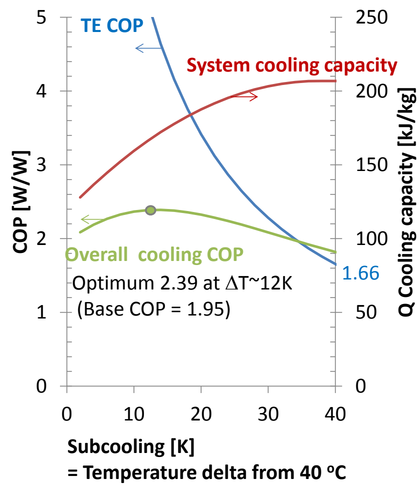

Figure 5: TE COP, the overall system COP, and the cooling capacity change as functions of the subcooling temperature.

3. SEASONAL COP

3.1 Seasonal COP

The ambient air temperature is a significant contributor of system COP in CO_2 refrigerant cycles. It is shown in Fig. 6, where ZT=1.5 and 2.5 are considered in TE materials. Seasonal COP is investigated according to the annual climate cycle information at potential locations for datacenters. Since the datacenters are intentionally located in a relatively abundant area and potentially adjacent to a power plant, hence the fresh-water-rich environment has been desired in addition to the shorter distance from a busy traffic network for the information access. The places compared are Mumbai, Singapore, Tokyo, as well as Lakeland, Austin, and Seattle in the United States. Among the condition of discharge pressures, Figs. 7 and 8 show the cases of 8.5 MPa and 10MPa, respectively. Due to the significant reduction of COP in baseline at high temperature over upper 20s °C of baseline, the dramatically improvement in tropical climate region such as Mumbai and Singapore. In contrast, there is almost no change in Seattle and very few change in Tokyo. Changes in the TE material performance from ZT=1.5 to 2.5 are observed relatively smaller.

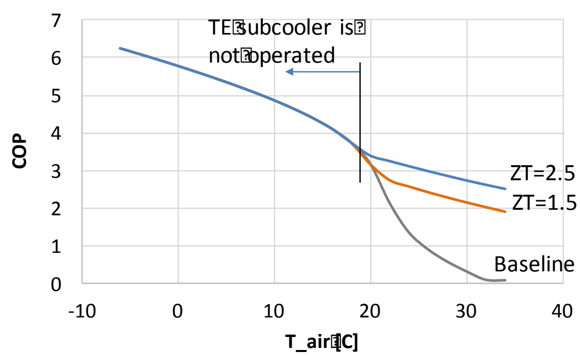

Figure 6: COP vs ambient air temperature with 8.5 MPa discharge pressure from the compressor and the evaporator temperature is 10 ºC. TE subcooling is effective compared to the baseline system in high temperature ambient approximately over 20 ºC depending on the ZT value of TE material.

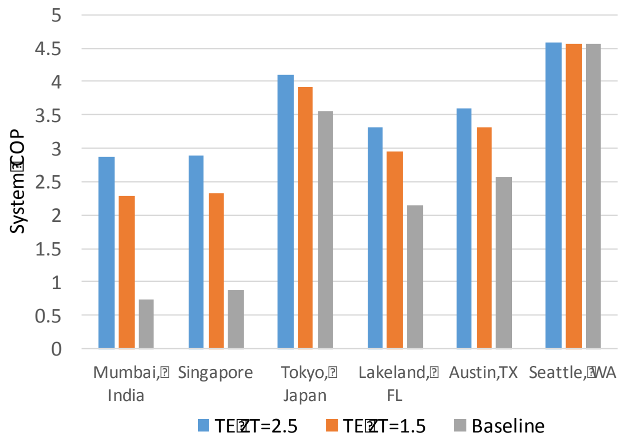

Figure 7: System seasonal COP comparison of six selected locations at 8.5 MPa discharge pressure condition in variation of with and without TE subcooling and ZT value of 1.5 and 2.5.

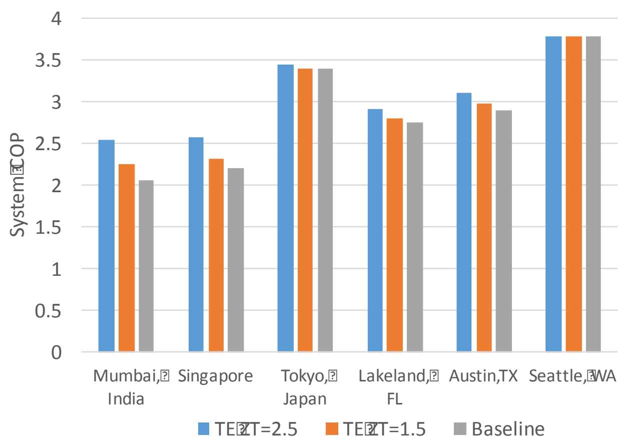

Figure 8: System seasonal COP comparison of six selected locations at 10 MPa discharge pressure condition in variation of with and without TE subcooling and ZT value of 1.5 and 2.5.

4. COST ANALYSIS AND OPTIMIZATION

The mass use of TE material is a major cost driver since the material is expensive. In this particular study, we investigate the impact of the subcooling temperature to the additional cost for the TE subcooling. Fig. 10 shows the relationship between the subcooling temperature and the TE cost. As increasing the subcooling temperature, system cooling increases and the TE module cost also increases. This is a contradict trend and should have a best balance for every particular design. This cost is based on the mass of the TE legs, which is required to obtain the maximum COP for the TE subcooler at the temperature condition. The cooling capacity is fix to 1.5 kW/m^2 for this calculation. The optimum design may differ depending on the heart flux condition.

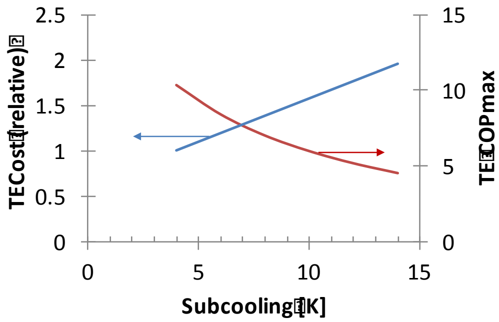

Figure 9: TE cost and TE COPmax as functions of subcooling temperature.

More details of the relationship between the cost and COP for the TE subcooler is shown in Fig. 10 in variation of subcooling target temperature 26 - 36 ºC, while the gas cooler exit temperature is 40 ºC. The design conditions are ZT = 1.5, thermal conductivity β = 1.5 W/m.K, electrical conductivity 8.65x10^5 1/Sm, and Seebeck coefficient 290 µV/K, number of legs N = 10^4 1/m^2, fill factor F = 50%. The reference design of the leg is a cubic shape defined by the thickness to cross section area ratio d/a = 1. The reference leg thickness is 5 mm. Heat load is considered as 1.5 kW/m^2 and the heat transfer coefficient is 500 W/m^2.K (with ideal heat exchanger effectiveness =1) each for the hot and cold side. Since the cost is linear to the leg thickness, thinner leg costs lesser. As seen in Fig. 10, the COP is initially increases as increasing leg thickness starting from zero but saturates at some point and decreases thereafter. The maximum COP is observed and the peak is dependent to the subcooling temperature. According to the Fig. 5, the best system COP is found at near 12K of subcooling, which equivalent to the curve 28 ºC in this figure. The optimum design of the TE leg is d/a = 1.77 (thickness is 8.86 mm) for 5 mm x 5 mm cross section. The internal electrical resistance of legs is 20.5 Ω per unit area (m^2) of the module. Required current is then 3.76 A.

The minimum cost at the maximum COP of TE subcooling except the heat exchanger is approximately $ 1.5 – 3 /W (per cooling capacity), which is almost comparable to the cost required for a heat exchanger.

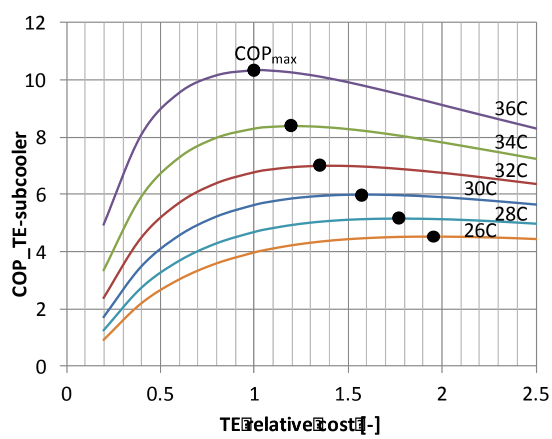

Figure 10: TE subcooler COP as function of TE relative cost in variation of subcooling temperature based on the change in thickness of the TE module. ZT value is 1.5 and the thermal conductivity of the TE element is 1.5 W/m.K.

5. CONCLUSIONS

A CO_2 air source refrigeration cycle with an assist of thermoelectric subcooler was investigated. Cooling COP of the baseline (standard) transcritical CO_2 system shows well under a colder climate but becomes significantly lower under a warm or hot climate approximately at 20 ºC or higher. TE cooler works significantly effectively while the temperature delta of heat pumping is small around 10 -15 ºC or less. Using an ordinal TE material (ZT = 1.5), the optimum design yields a 20% improvement of the system COP with TE subcooling temperature of 12 ^{o}C at the gas cooler exit temperature of 40 ^{o}C condition.

Seasonal COP of the TE subcooled system was evaluated with six different climate conditions. Under a hot climate, the TE subcooling improves the system COP significantly, while there is a very small change observed under a cold climate. The difference is quite visible. In design of the TE subcooler, the COP and the cost of TE module are in controversial trend, hence there should be an optimum cost effective design. In future study, the most cost effective design will be investigated.

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

A	cross section area	(m^2)
COP	coefficient of performance	(-)
d	thickness of TE element	(m)
G	cost per heat removed	($/W)
H	enthalpy	(J/kg)
I	current	(A)
Q	heat	(W)
S	Seebeck coefficient	(V/K)
T	temperature	(C)
W	input power	(Watt)

Greeks

β	thermal conductivity	(W/m-K)
σ	electrical conductivity	(1/S-m)
ψ	thermal resistance	(K/W)

Subscript

C	compressor
max	maximum
TE	thermoelectric

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