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Jonathan Schoenfeld
University of Maryland

Jan Muehlbauer
University of Maryland

Yunho Hwang
University of Maryland

Reinhard Radermacher
University of Maryland

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Integration of a Thermoelectric Subcooler into a Carbon Dioxide Transcritical Vapor Compression Cycle Refrigeration System

Jonathan SCHOENFELD, Jan MUEHLBAUER, Yunho HWANG, Reinhard RADERMACHER

Center for Environmental Energy Engineering
Department of Mechanical Engineering, University of Maryland
3163 Glenn Martin Hall Bldg., College Park, MD 20742, USA

* Corresponding Author: Tel. (301) 405-5247, E-mail: yhhwang@umd.edu

ABSTRACT

Thermoelectric (TE) modules have been integrated into a subcooler of an experimental CO₂ transcritical vapor compression cycle test system. The TE Subcooler was designed and fabricated to subcool CO₂ exiting the gas cooler to a temperature below ambient. The heat from the TE Subcooler was rejected to the ambient utilizing a separate thermosyphon refrigerant loop with a separate condenser. The thermoelectric modules operate at efficiencies greater than the baseline system, increasing capacity and the overall coefficient of performance (COP) of the entire system. Additionally, subcooling of the CO₂ before the expansion device leads to a reduced optimum gas cooling pressure, resulting in greater COP improvement that cannot be achieved in conventional refrigerant vapor compression systems utilizing a TE Subcooler. The use of a simple aluminum microchannel TE Subcooler was shown to increase the COP of the system by 5% and increase the capacity by 15%.

1. INTRODUCTION

Thermoelectric cooling is a result of the Peltier effect. A thermoelectric element consists of an electrical junction between two semiconductors with different Peltier coefficients in which a DC current is applied. The semiconductors are usually n-type and p-type Bismuth Telluride. In an n-type semiconductor a thermal drift flux (Gurevich and Logvinov, 2005) proportional and opposite in direction to the applied current will develop while in a p-type semiconductor the thermal drift flux develops in the same direction. Because of this the junction of the two semiconductors increases or decreases in temperature depending on the direction of the current.

A thermoelectric module consists of many thermoelectric elements connected electrically in series and thermally in parallel. A thin electrically insulated and thermally conductive plate is placed on both the cold and hot side electric junctions. Through the Peltier effect a thermoelectric module is able to pump heat from the cold surface to the hot surface. The thermal conductivity of the semiconductors, as well as the radiation and convection within pockets between them, increasingly compete with the thermoelectric effect as the temperature difference (ΔT) across the module increases (Nabi and Asias, 2005). High coefficient of performance (COP) has been reported, but only at a relatively low ΔT . Due to this limitation thermoelectric coolers have been used almost exclusively in small cooling applications including small refrigeration systems and vehicle seat cooling.

Conventional vapor compression air conditioning systems operate at a high COP over a fairly large temperature difference. Winkler et al. (2006) recognized that thermoelectric modules could be employed at the outlet of a condenser to reduce the refrigerant temperature to below the ambient temperature. In this way the thermoelectric modules would operate with a small ΔT and high efficiency. The heat removed from the refrigerant results in a direct increase in the capacity of the vapor compression system. In a transcritical cycle the high side pressure is optimized for a particular working fluid temperature entering the expansion device. By subcooling the working fluid the high side pressure can be reduced, increasing compressor efficiency and decreasing compressor power. The

thermoelectric (TE) Subcooler is well suited to increase both the COP and capacity of carbon dioxide (CO₂) transcritical cycle for the reasons previously discussed.

The objective of this project is to design and fabricate a subcooler which utilizes TE modules and demonstrate an improvement in the capacity and COP of a transcritical CO₂ refrigeration system utilizing a TE Subcooler.

2. APPROACH

2.1 CO₂ System

A small CO₂ transcritical vapor compression cycle was built to test the potential performance improvement of a TE Subcooler utilizing a prototype compressor with a cooling capacity of roughly 1 kW. The gas cooler was constructed from two banks of two aluminum microchannel, louver-fin heat exchangers connected in series. A manual expansion valve was used as an expansion device. The evaporator was constructed out of thick-walled copper pipe wrapped in heating tape with a total heating capacity of about 1.6 kW. The output of one of the heating tapes was controlled by a variable-voltage AC power supply.

Initial testing was performed on the first generation TE Subcooler, which was designed for use in a large conventional refrigerant air conditioning system. The subcooler consists of a single aluminum microchannel CO₂ heat exchanger sandwiched in between two aluminum microchannel heat rejection heat exchangers. The TE modules were placed in between the CO₂ heat exchanger and the heat rejection heat exchangers, with five on each side for a total of ten modules. A thermosyphon loop was employed to reject the heat from the CO₂. The flow of refrigerant within this separate loop is driven by buoyancy. The TE Subcooler was aligned vertically, with CO₂ entering at the top from the gas cooler and exiting at the bottom to the expansion valve. The thermosyphon refrigerant, in this case R22, boils within the heat rejection microchannels. The vapor refrigerant travels upward to the thermosyphon condenser, where it condenses and travels back to the inlet of the heat rejection microchannels at the bottom of the TE Subcooler. In the original system setup the thermosyphon condenser was integrated into the gas cooler utilizing the same set of fans. Initial testing showed that this significantly reduced the capacity of the gas cooler and resulted in a negligible system performance improvement. In subsequent testing the thermosyphon condenser was separated from the gas cooler employing a second fan. Figure 1 shows a schematic diagram of the experimental setup.

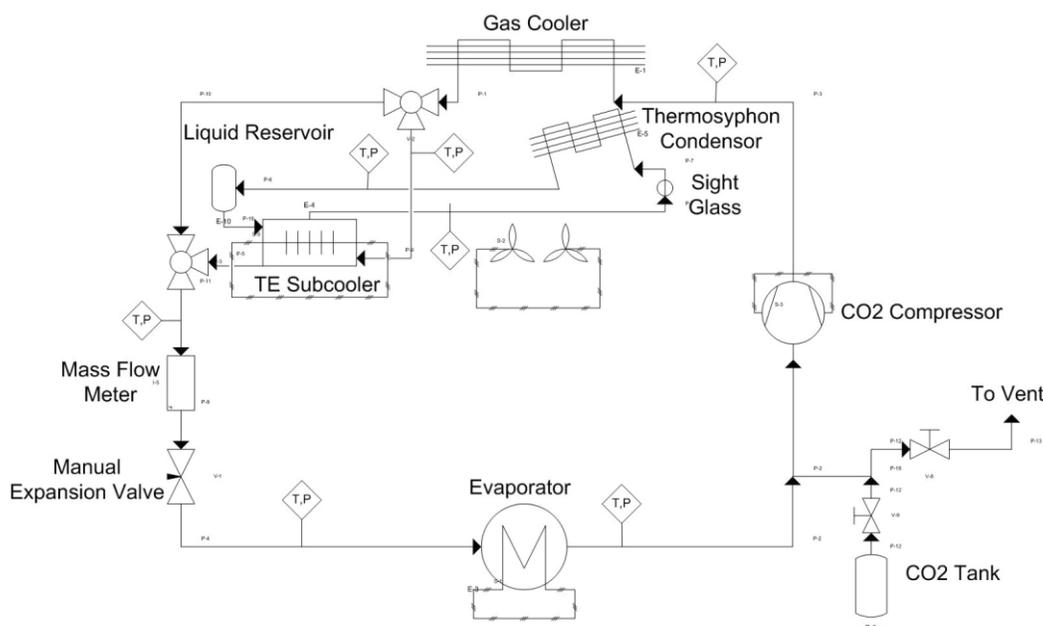


Figure 1: Schematic of the experimental setup

In order to characterize the system performance, in stream T-type thermocouples and pressure transducers were installed at the compressor suction and discharge, gas cooler outlet, TE Subcooler outlet, and evaporator inlet. The

mass flow rate (MFR) was measured by a Coriolis mass flow meter. Watt meters were installed to measure the power input to the compressor and the evaporator heating tapes. Additional thermocouples were installed on the insulation of the heating tapes to estimate heat loss and at the inlet fans of the gas cooler to ensure constant inlet air temperature between tests.

Testing was performed within an environmental chamber at an ambient temperature of 35°C. Steady state performance was measured over 20 minute intervals at a frequency of 1 Hz. All tests were run at a constant superheating of 11.1 K according to ANSI/ARI Standard 540 (1999). The suction pressure was also kept constant at 4,198 kPa, corresponding to a 7.2°C saturation temperature. The high side pressure was varied by adjusting the charge in order to determine the optimum value for a maximum COP.

The system capacity was calculated from the MFR, compressor suction enthalpy and evaporator inlet enthalpy as shown in Equation (1). It was assumed that the expansion device was isenthalpic, so the enthalpy entering and exiting the expansion valve are equivalent. The COP was calculated from the compressor power input and the system capacity as shown in Equation (2). The power input to the fan was not considered in this analysis. The incorporation of the thermosyphon condenser into the gas cooler is required to eliminate additional power required by the condenser fan.

$$Q = \dot{m}(h_{cmp,suct} - h_{evap,in}) \quad (1)$$

$$COP = \frac{Q}{P_{cmp}} \quad (2)$$

3. RESULTS and DISCUSSION

3.1 System Performance

The first generation TE Subcooler system was tested at a suction pressure of 4,198 kPa. The maximum COP for the baseline system was 2.35 at a discharge pressure of 9,553 kPa with a corresponding cooling capacity of 1.44 kW. When employing the TE Subcooler a maximum COP of 2.48 was reached at a reduced discharge pressure of 9,042 kPa with a corresponding cooling capacity of 1.57 kW. The maximum system capacity achieved utilizing the TE Subcooler was 1.66 kW at a COP of 2.38. The system performance for each of the three cases described above is listed in Table 1. For each of the TE Subcooler systems the CO₂ temperature entering the expansion device was below the ambient temperature of the chamber, resulting in capacity increases beyond what could be accomplished by increasing the size of the gas cooler. Due to heat generated by the compressor and heating tape evaporator the gas cooler inlet air temperature was about ~1 K above the ambient temperature. With an approach temperature of less than a degree it is unlikely that increasing the gas cooler size would result in any significant system performance improvement.

Table 1: Baseline and TE Subcooler system performance at a suction pressure of 4,198 kPa

System	Baseline System (Maximum COP)	TE System (Maximum COP)	TE System (Maximum Capacity)
System COP	2.354	2.476 (+5.2%)	2.379 (+1.1%)
System Capacity (kW)	1.440	1.569 (+9.2%)	1.657 (+15.3%)
Discharge Pressure (kPa)	9,553	9,042	9,206
Compressor Power (kW)	0.610	0.591	0.601
Mass Flow Rate (g/s)	9.9	10.2	10.1
TE Supply Current (Amps)	-	4	6
Number of Modules	-	10	10
TE Capacity (kW)	-	0.204	0.256
TE COP	-	4.839	2.676
GC/TE Outlet Temperature (°C)	36.6	33.9	31.9

Figure 2 shows the system COP and capacity in response to discharge pressure. The COP of the TE Subcooler system is fairly flat over a large discharge pressure range particularly at lower pressures. The TE Subcooler system may require less control over the high side pressure in order to operate efficiently, which could be considered an additional benefit.

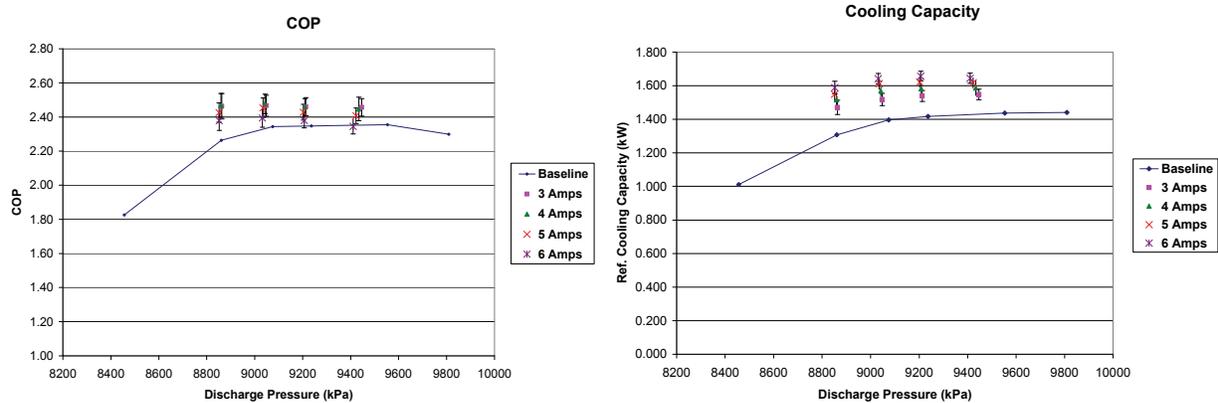


Figure 2: System performance response to discharge pressure

The P-h diagram for the baseline and TE Subcooler system with a maximum COP and the TE Subcooler system with a maximum capacity are shown in Figure 3. The increased slope during the compression process for the TE Subcooler systems reflects the increase in compressor efficiency at a reduced pressure ratio. The additional capacity results primarily from the reduced quality entering the evaporator. There is also an increase in flow rate resulting from the increase in volumetric efficiency of the compressor. For all cases the suction conditions were identical, which can be seen from the lower right point on the graph.

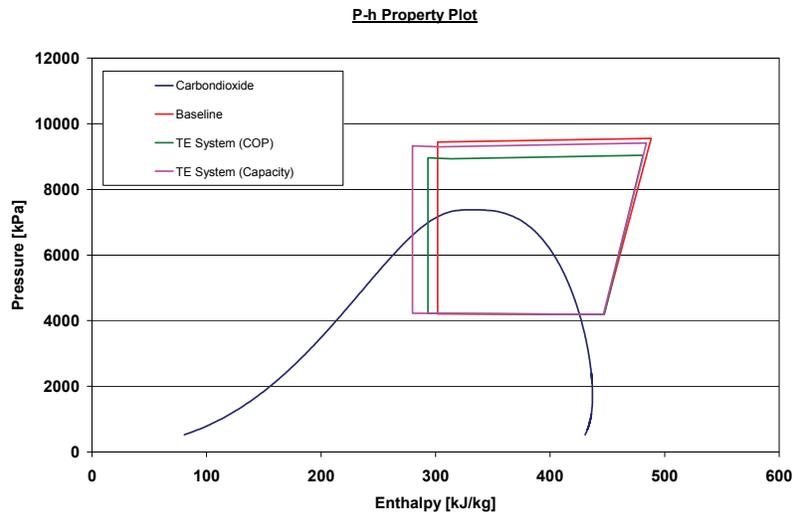


Figure 3: Pressure - enthalpy plot for the baseline system, TE System with maximum COP, and TE System with maximum capacity

3.2 TE Subcooler Performance

As discussed above, the COP of the TE modules decreases with an increasing temperature difference across the module. As the supply current to the TE Subcooler is increased the temperature difference across the modules increases which results in both an increase in cooling capacity and a decrease in TE COP. This trend can be seen in Figure 4. Additionally, as the TE Subcooler cooling capacity increases the temperature of the CO₂ at the outlet decreases resulting in greater temperature differences across the modules at the end of the subcooler.

The thermosyphon refrigerant temperature increases with increasing TE Subcooler capacity. This is due to the fixed thermosyphon condenser size. The thermosyphon pressure will increase if there is a greater rate of refrigerant boiling in the thermosyphon evaporator than condensing in the thermosyphon condenser. As the pressure rises, so does the saturation temperature until the condenser heat load is great enough to match the evaporation rate in the evaporator.

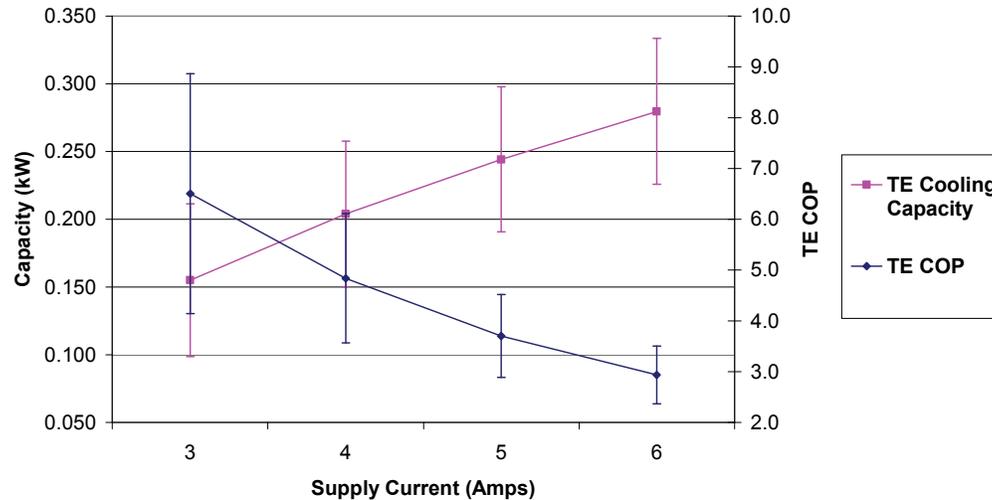


Figure 4: TE Subcooler performance at compressor discharge pressure of ~9,040 kPa

The performance of the TE Subcooler decreased as the discharge pressure was increased. As discharge pressure increases the volumetric efficiency of the compressor is reduced causing a reduction in mass flow rate. Even though the compressor discharge temperature increases, the reduced mass flow rate results in a decreased gas cooler outlet temperature. This has a negative effect on the TE Subcooler, as the modules must pump heat from a lower temperature. Additionally, the reduced MFR reduces the convective heat transfer of the microchannel CO₂ heat exchanger within the TE Subcooler. Figure 5 illustrates the two effects resulting from the increase in discharge pressure. The optimal discharge pressure is a balance of the effects on the baseline system performance and TE Subcooler performance.

There is an additional barrier to subcooling to lower temperatures besides the increased temperature difference between the CO₂ and heat rejection refrigerant. The properties of supercritical CO₂ are sensitive to temperature fluctuations around 35°C. Particularly, viscosity increases sharply as temperature decreases. Along the length of the subcooler, the CO₂ viscosity increases resulting in a decrease in the Reynolds number which adversely affects the convective heat transfer.

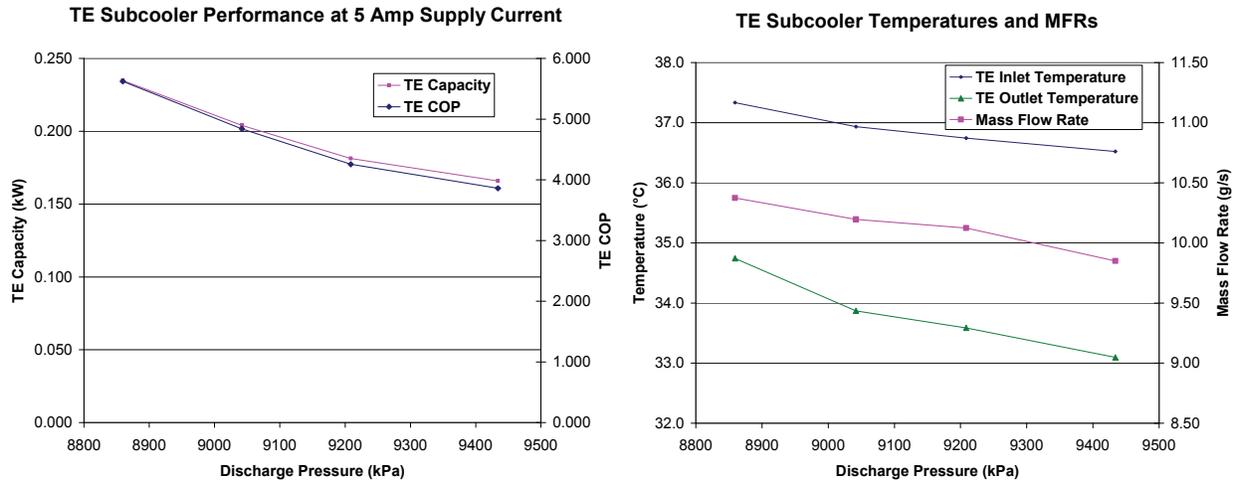


Figure 5: TE Subcooler performance dependence on discharge pressure

3.3 Expander-TE Subcooler System

The use of an expander in place of an expansion device has been shown to successfully improve the performance of transcritical CO₂ systems (Huff, 2003). The expander can be coupled directly to the compressor to provide a portion of the compression power or to a generator to provide electrical power to the compressor. The electrical power produced by an expander-generator could also be used to power the TE modules of a TE Subcooler, without the need for direct coupling to the compressor or power conditioning. A schematic of such a system is shown in Figure 6 (a). In this system the TE Subcooler would require no additional power besides that which is provided by the expander. Therefore, capacity gains would require no additional power and would result in greater improvements in COP. Figure 6 (b) shows a graph of the power input required by the TE Subcooler for tests performed at a discharge pressure of ~9,100 kPa. On the same axis is the theoretical electric power generated from an expander with an indicated and volumetric efficiency of 70% and 98%, respectively, and a generator efficiency of 90%. The point at which the two lines intersect represents the state at which such a system could operate. The resulting system COP and capacity would be 2.67 and 1.60 kW, respectively, a 13% and 11% improvement over the baseline system.

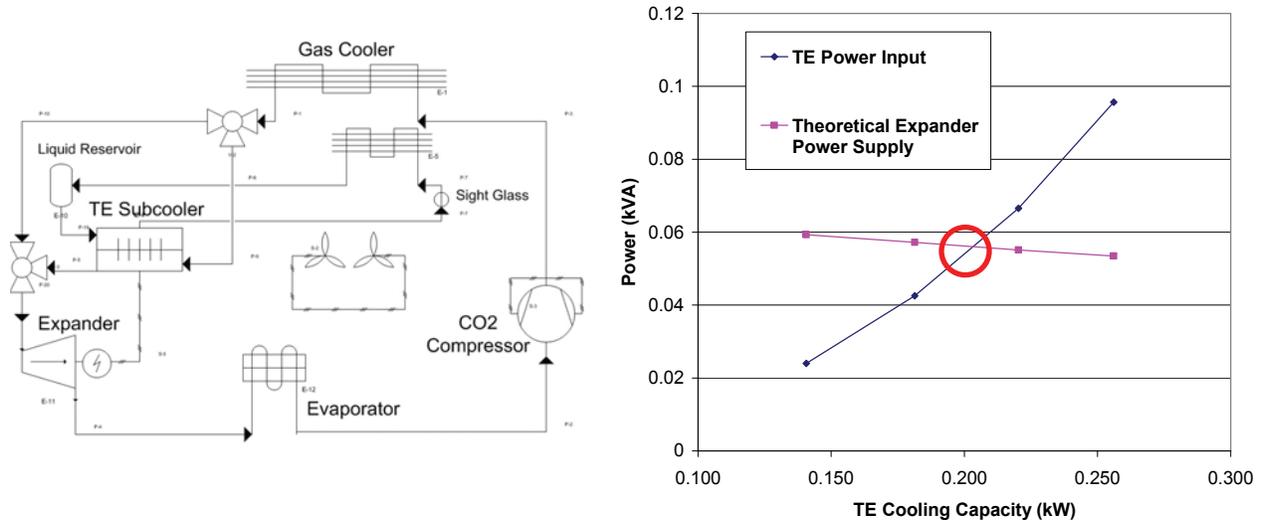


Figure 6: a) Schematic of Expander-TE system, b) TE Subcooler power input requirement and theoretical expander power supply

4. CONCLUSION

Performance improvement of a CO₂ transcritical cycle utilizing a TE Subcooler was experimentally demonstrated. An increase in COP of 5% was achieved with a corresponding 9% increase in capacity. A capacity improvement of 15% was achieved at comparable COP as the baseline system. Significant performance improvements in the TE Subcooler are possible and will result in greater increases in system COP and capacity. Second and third generation TE Subcoolers are currently in the testing phase.

The effects of discharge pressure on a TE Subcooler integrated into a transcritical cycle were investigated. At lower discharge pressures the reduced performance of the baseline system is compensated, at least in part, by the increased performance of the TE Subcooler. This balance allows the system to achieve a close to optimum COP at a wide range of discharge pressures lower than the optimum baseline pressure.

The greatest benefits of the TE Subcooler are found in the system capacity improvements. The utilization of alternative power sources, such as an expander can result in greater COP improvements. In drier climates in which cooling loads are dependent primarily on solar radiation, the possibility of powering the TE Subcooler using a Photovoltaic panel may result in greater performance improvements.

NOMENCLATURE

h	Specific enthalpy	(kJ/kg)	Subscripts	
\dot{m}	Mass flow rate	(kg/s)	cmp,suct	Compressor suction
Q	Cooling capacity	(kW)	evap,in	Evaporator inlet
P	Compressor power	(kW)		
COP	Coefficient of Performance			
MFR	Mass Flow Rate			
TE	Thermoelectric			
ΔT	Temperature difference			

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