Performance Analysis of Tube in Tube CO_2 Gas Cooler

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Abstract- **As a kind of promising natural refrigerant, carbon dioxide has been attracting much attention from many researchers, especially in the fields of automobile air conditioners, commercial refrigeration, and water heating. When the CO_2 refrigerant is used in heat pump water heaters (HPWHs), it exhibits excellent performance in HPWH applications. The coefficient of performance (COP) of the system may achieve higher values than conventional refrigerants because of better temperature glide matching. Additionally, CO_2 HPWHs could provide higher temperature water up to 90 °C. In CO_2 air-conditioning and heat pump systems, CO_2 rejects heat at a pressure above the critical pressure (7.38 MPa) in the gas cooler without phase change. When the CO_2 is at supercritical pressures, some small fluid temperature and pressure variations may produce large changes in the thermophysical properties, and this is especially pronounced when the temperature is near the critical point. The gigantic change in the thermophysical properties may result in significant deviations in both heat transfer and fluid flow behaviors. This study will show the effect of nonlinear variation of supercritical CO_2 specific heat capacity with temperature on local heat transfer rate.**

Index Terms- **Gas cooler, CO_2 Heat pump, Transcritical, approach temperature difference.**

1. INTRODUCTION

Climate change is a major worldwide concern with potentially dramatic impacts on developing and industrialized countries alike. The effort to mitigate climate change centers on the reduction of greenhouse gas emissions. Carbon dioxide is the greenhouse gas which receives the most attention due to the sheer volume of emissions, much of it from power generation. However, other atmospheric pollutants such as methane and nitrous oxide have a far greater impact on climate change on a per mass basis. The global warming potential (GWP) is a relative measure of the heat-trapping effect of a gas in comparison to an equal mass of carbon dioxide over a given quantity of time in the atmosphere. Methane and nitrous oxide have GWP values of 23 and 296 respectively, both based on a 100-years' time span.

Many of the refrigerants used in HVAC&R systems are potent greenhouse gases. R134a, for instance, has a GWP of 1300 over a 100-year time span. When securely contained in properly operating system refrigerants do not impact climate change; however, system leaks and improper recovery of refrigerants during repairs or at end of life result in these harmful gases entering the atmosphere. Some climatologists have called for a complete worldwide phase-out of refrigerants with high GWP similar to the phase-out of ozone-depleting substances enacted under the Montreal Protocol in 1987. Already, the European Union has approved the scheduled phase-out of mobile air conditioning systems using refrigerants with GWP greater than 150. This directive was ratified in 2007 and went into effect beginning in 2008.One potential replacement refrigerant is carbon dioxide, a natural refrigerant which has a negligible impact on climate change. Carbon dioxide used in HVAC&R systems has a zero net impact on climate change because it has been recovered from other industrial processes. Furthermore, carbon dioxide (CO_2) is not toxic, flammable or corrosive, and it has no impact on the ozone layer. It is inexpensive and readily available. CO_2's performance as a refrigerant in heat pump systems is also competitive with refrigerants currently in use. Gas cooler, in which CO_2 is cooled with the persistent temperature drop, is different from those constant temperature condensation processes, and it is one of the most important devices in the CO_2 transcritical cycle since its flow arrangements and behaviors can greatly affect the optimal operating pressure and system efficiency.

Carbon dioxide becomes a supercritical fluid at 31.1 ◦C at 73.7 bar. In a conventional (subcritical) heat pump cycle, low critical temperature (Tcrit) is a disadvantage because it limits the operating temperature range; heat cannot be delivered at temperatures greater than the critical temperature. Further, at temperatures less than but near Tcrit, the enthalpy of vaporization is reduced. This leads to a reduction in heating capacity and poor performance of the system. Thus a conventional heat pump should avoid operating at a heat rejection temperature near Tcrit. In a transcritical heat pump, heat rejection pressures are greater than the supercritical pressure and heat delivery temperatures are no longer limited by Tcrit. CO_2's low critical temperature provides the opportunity to operate in a transcritical manner. High working pressure is the other notable distinction of CO_2 heat pumps. Both subcritical and transcritical heat pump systems using CO_2 operate at pressures greater than with most other refrigerants.

In a liquid-cooled gas cooler, the contribution of CO_2 thermophysical changes is more important than that in an air-cooled one due to air-side thermal resistance is dominant in the air-cooled heat exchangers. But, investigations concerning the water cooled gas cooler are comparatively fewer. Therefore, the objective of this study is to develop a simple counter-flow heat exchanger model capable to analyze the heat transfer behavior of CO_2 tube in- tube water-cooled gas cooler subject to the refrigerant flowing in an inner tube side and water flow in annulus side using CFD modeling.

1.1. Performance of transcritical CO_2 heat pump systems

A heat pump is generally used to heat either air or water. Air, for example, may be heated for the purpose of space heating, while water may be heated for domestic hot water or spacing heating. The use of air or water as the heat recovery fluid affects the heat flux at the gas cooler due to the differing thermos physical properties of the fluids. Likewise, the evaporator may be absorbing heat from different sources such as ambient air, water from a lake or the ground, which will impact the heat exchange. These factors, directly or indirectly, will influence the operating parameters and hence will impact the performance of a heat pump. This review includes theoretical and experimental works conducted with water as the heat source and heat sink fluids. An air conditioning or refrigeration system is essentially the same as a heat pump except that the desired output is different and the operating temperatures are different. Because the systems are similar, research on transcritical CO_2 air conditioning and refrigeration systems can be instructive for transcritical CO_2 heat pump development as well. For this reason, studies on systems designed for cooling purposes have been included in this paper. It must be taken into account that the COP and capacity are defined differently for cooling systems. Cooling capacity is the amount of heat absorbed by the evaporator, while COP cooling is the ratio of cooling capacity to work input.

1.2. Difference between refrigerant condensation and supercritical CO_2 cooling

The heat transfer process in the gas cooler is also very different from the condensing process of conventional refrigerants as shown in Figure 1. Because heat transfer occurs by sensible cooling, the difference between the CO_2 temperature at the gas cooler inlet and outlet is typically larger than during heat rejection by condensation. The warmth move process in the gas cooler is likewise altogether different from the consolidating procedure of customary refrigerants as appeared in Figure 1. Since warm exchange happens by sensible cooling, the distinction between the CO_2 temperature at the gas cooler gulf and outlet is regularly bigger than amid warm dismissal by buildup. This temperature contrast is known as the refrigerant temperature glide. Contrasted with a buildup procedure, the gliding temperature profile of CO_2 can be all the more firmly coordinated to the gliding temperature profile of the auxiliary liquid, which enhances warm exchanger adequacy.

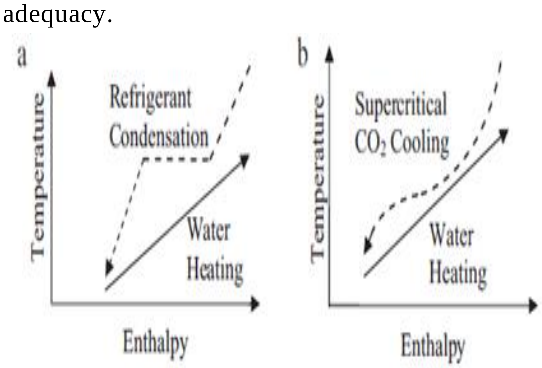


Figure 1. Temperature profile for heat rejection by: (a) condensation process and (b) supercritical gas cooling process.

The performance of a TCHP improves when the temperature glide increases. Results from a simulation by Laipradit et al. [20] showed that COP increased as water inlet temperature decreased. The reduced water inlet temperature corresponds with a reduced refrigerant temperature at the gas cooler outlet and hence a larger refrigerant temperature glide. Because CO_2 can provide a large temperature glide TCHP performance can actually benefit from multiple heating loads.

2. MODELING METHOD

2.1. Modeling of CO_2 tube-in-tube gas cooler

A CFD model was developed for CO_2tube-in-tube gas cooler as shown in figure 2 with detailed specification shown in table 1. The CO_2 refrigerant flows inside the inner tube while the heated water flows through the annular passage in reverse direction. For each segment, mass and energy conservation equations must be satisfied, so the heat transfer balance between the refrigerant and water is written as follows:

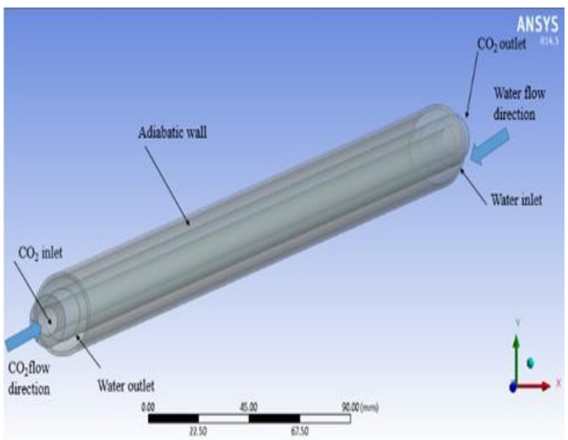


Figure 2. Design of Gascooler

To simplify numerical simulation, following assumption are made:

- The system is operating at steady state and exchangers operate adiabatically
- The fluid flow and heat transfer processes are turbulent and in steady state
- Changes in kinetic and potential energy are negligible
- The natural convection induced by the fluid density variation is neglected
- The tube wall temperature kept constant in all side
- The heat exchanger is well insulated hence the heat loss to the environment is totally neglected.

Inner pipe specification (mm)	
Inner diameter	8.88
Outer Diameter	15.88
Length	1400
Outer pipe specification (mm)	
Inner diameter	23.00
Outer Diameter	25.40
Length	1300

Table 1. Geometry specification

Boundary conditions are used according to the need of the model. The inlet mass flow rate and temperature are used similar to the experimental conditions in order to have a comparison. The walls are separately specified with respective boundary conditions. 'No slip' condition is considered for each wall. Except the inner tube walls, outer tube wall is set to zero heat flux condition. The tube walls are set to 'coupled' for transferring of heat between tube and side fluids. The details about all boundary conditions can be seen in the Table 2.

Table 2. Boundary conditions

Test Runs	Inlet temperature of water ($T_{w,i}$,°C)	Inlet temperature of CO2 ($T_{CO_{2,i}}$,°C)	Inlet mass flow rate of water ($\dot{m}_w\ k_g/sec$)	Inlet mass flow rate of CO2 ($\dot{m}_{CO_2}\ k_g/sec$)
A	20	120	0.04011	0.01963
B	25	125	0.040497	0.0274
C	35	70	0.12914	0.02043
D	16	90	0.09472	0.03657
E	27	105	0.0455	0.0198
F	18	100	0.09438	0.03866
G	23	123	0.084087	0.02862
H	19	115	0.084087	0.03436

The heat balance between the water and the carbon dioxide can be expressed by the following equations (1).

$$Q = m_c C_{p_c}\left(T_{c,i} - T_{c,o}\right) = m_w C_{P_w}\left(T_{w,o} - T_{w,i}\right) \quad (1)$$

In each case, the Gnielinski's correlation, as shown in equation (8), is used to calculate the respective Nusselt Number:

$$Nu = \frac{\frac{f}{8}(Re-1000)\,Pr}{12.7\sqrt{\frac{f}{8}}\left(Pr^{\frac{2}{3}}-1\right)+1.07} \quad (2)$$

Where $f = friction\ factor$

In addition, equation (2) requires the knowledge of the friction coefficient, ζ. appropriate results were obtained by using Filonenko's correlation as shown in equation (3):

$$f = (0.79 \ln(Re) - 1.64)^{-2} \quad (3)$$

Once the mean Nusselt Number has been obtained, the heat transfer coefficient can be computed as shown in equation (4):

$$h = \frac{Nu}{D} k_{bulk} \quad (4)$$

Dang and Hihara (2004) modified the Gnielinski correlation (2) become a new correlation as described in equation (5). Effects of parameters such as mass flux, pressure, heat flux, and tube diameter on the heat transfer coefficient and pressure drop were analysed. The correlation predicted experimental data with an accuracy of 20%.

$$Nu = \frac{\frac{f}{8}(Re-1000)\,Pr}{1.07+12.7\sqrt{\frac{f}{8}}\left(Pr^{\frac{2}{3}}-1\right)+1.07} \quad (5)$$

3. RESULT AND DISCUSSIONS

3.1. Effect of Inlet cold fluid temperature on heat transfer rate

For the constant value of hot fluid inlet temperature 393 k and mass flow rate of cold fluid and hot fluid 0.04011 kg/sec and 0.01963 kg/sec respectvely, the heat transfer rate is more for low inlet temperature of cold fluid (water) means if for increase the cold fluid inlet temperature for constant hot fluid(CO_2) temperature the heat transfer rate will decrease is shown in following figure 3.

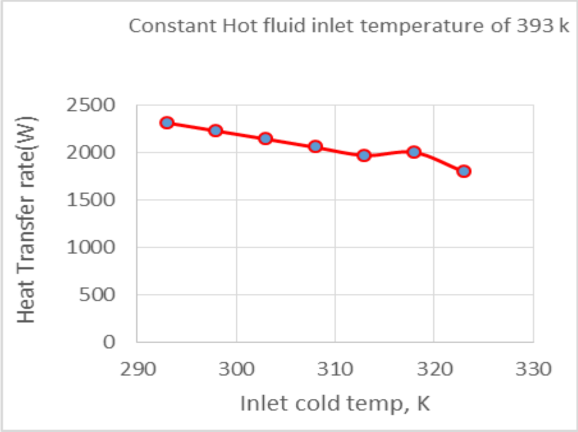
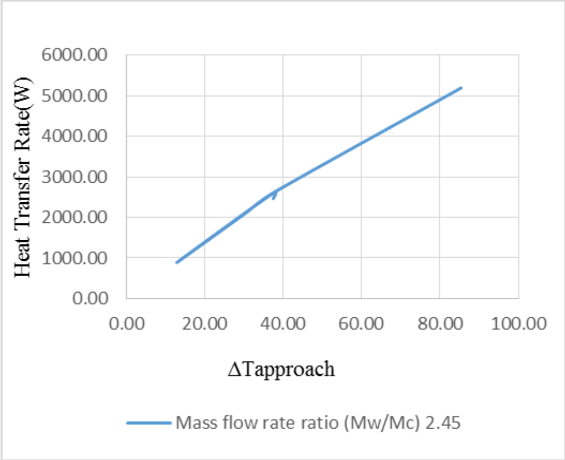


Figure 3. Heat transfer rate Vs inlet cold fluid temperature

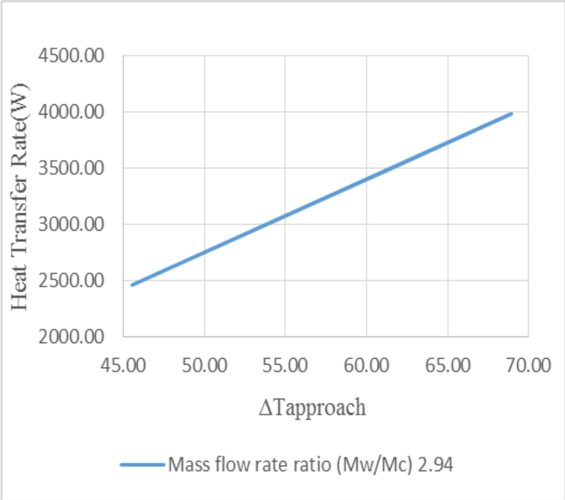
3.2. Effect of approach temperature on heat transfer rate

Heat transfer rate increase with the increase of approach temperature differences at cold end. Figure 4. (a,b,c,d) Shows the increment of heat transfer rate for constant mass flow rate at different set of inlet condition of temperature of hot and cold fluid.

$$\Delta T_{approach} = T_{C,o} - T_{w,i} \quad (6)$$



(a)



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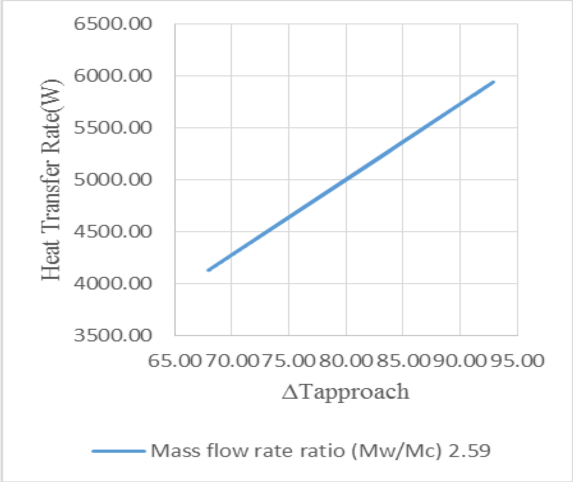
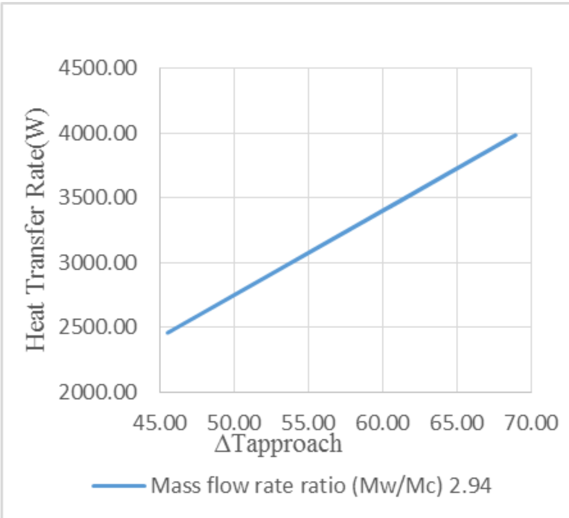


(c)



(d)

Figure 4. Heat transfer rate Vs ΔTapproach

Heat transfer rate calculated at constant mass flow of inlet hot and cold fluid for different inlet condition of temperature shown in table 2. From figure 5, initially approach temperature difference more for mass flow rate ratio of 2.59 and that mass flow rate ratio, heat transfer rate also maximum. This maximum heat transfer rate occurs because of high specific of CO_2 at that temperature.

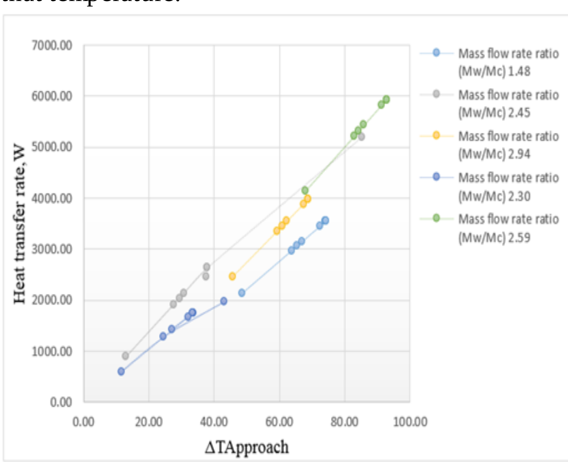


Figure 5. Comparisons of heat transfer rate Vs $\Delta T_{approach}$ at different mass flow rate ratio.

3.3 Specific heat capacity Vs absolute pressure variation at constant mass flow rate

At constant mass flow rate with different test runs from table 2, specific heat increase with the increase of absolute pressure till the pressure of 204000 pa beyond this pressure specific heat constant and may also increase with the increase of pressure for all test runs. Due to the drastic rise of CO_2 specific heat capacity, the local transfer rate has a maximum value within gas cooer.

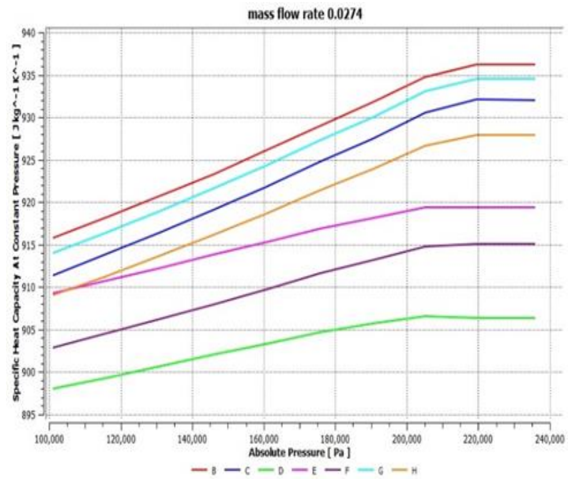


Figure 6. Specific heat capacity Vs absolute pressure variation

4. CONCLUSIONS

Carbon dioxide was used as a refrigerant in some of the earliest refrigeration systems. CO_2 fell out of favor when other refrigerants delivered superior performance over a wider range of conditions. In light of growing concerns over climate change, CO_2 has been revived as a potential refrigerant due to its benign environmental qualities. To overcome limitations imposed by CO_2's low critical point, most of the research has focused on the use of CO_2 in a transcritical cycle.

- It is found that the inlet water temperature at the gas cooler casts significant impact on this system performance. Despite the CO_2 mass flow rate may be increased with the inlet water temperature, the system heat transfer rate considerably change with the inlet water temperature.
- In order to increasing COP of system and the water outlet temperature of gas cooler, the heat transfer coefficient of the water side need to be improved.
- In order to enhance the efficiency of a milk processing and refrigeration plant by reducing coal consumption and ground water usage, a trans-critical CO_2 heat pump system with IHX is proposed that utilizes the rejected low grade heat from the ammonia based refrigeration system to pre-heat the boiler feed water.

- The natural fluid Carbon Dioxide displays some excellent properties in the use as a refrigerant in compression-type refrigerating or heat pump systems: it offers unequalled local and ecological safety, widespread availability at low cost, with no need for recycling and containment.

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