

PERFORMANCE IMPROVEMENT OF DOUBLE-TUBE GAS COOLER IN CO₂ REFRIGERATION SYSTEM USING NANOFLUIDS

by

Jahar SARKAR

Department of Mechanical Engineering, Indian Institute of Technology, BHU, Varanasi, India

Original scientific paper

DOI: 10.2298/TSCI120702121S

The theoretical analyses of the double-tube gas cooler in transcritical carbon dioxide refrigeration cycle have been performed to study the performance improvement of gas cooler as well as CO₂ cycle using Al₂O₃, TiO₂, CuO, and Cu nanofluids as coolants. Effects of various operating parameters (nanofluid inlet temperature and mass flow rate, CO₂ pressure and particle volume fraction) are studied as well. Use of nanofluid as coolant in double-tube gas cooler of CO₂ cycle improves the gas cooler effectiveness, cooling capacity, and COP without penalty of pumping power. The CO₂ cycle yields best performance using Al₂O₃-H₂O as a coolant in double-tube gas cooler followed by TiO₂-H₂O, CuO-H₂O, and Cu-H₂O. The maximum cooling COP improvement of transcritical CO₂ cycle for Al₂O₃-H₂O is 25.4%, whereas that for TiO₂-H₂O is 23.8%, for CuO-H₂O – 20.2%, and for Cu-H₂O – 16.2% for the given ranges of study. Study shows that the nanofluid may effectively use as coolant in double-tube gas cooler to improve the performance of transcritical CO₂ refrigeration cycle.

Key words: transcritical CO₂ cycle, double tube gas cooler, nanofluids, heat transfer, modeling, performance

Introduction

To attain a higher degree of sustainability, natural refrigerants such as ammonia, carbon dioxide, hydrocarbons, and nitrous oxide do appear more attractive than the other synthetic refrigerants. CO₂ is increasingly becoming the refrigerant of choice and have been widely accepted due to its various advantages including zero ozone depletion potential and negligible global warming potential. Within the last decade, CO₂ has drawn ample interest as a natural refrigerant in refrigeration, air-conditioning, and heat pump applications [1, 2]. The double tube heat exchanger have been extensively used as a gas cooler in transcritical CO₂ systems for smaller and medium capacity refrigeration and heat pump applications [3-6]. Nanofluid can be used as a coolant in gas cooler to improve the performance of CO₂ refrigeration system.

Nanofluids are a new class of nanotechnology-based heat transfer fluids that are engineered by stably suspending a small amount of nanoparticles. Nanofluids consisting of such particles suspended in liquids (typically conventional heat transfer liquids) have been shown to enhance the thermal conductivity and convective heat transfer performance of the base liquids. Nanofluids have the potential to reduce such thermal resistances, and various industrial groups would benefit from such improved heat transfer fluids [7]. Use of nanofluids in double tube heat exchanger is comparatively more effective [8]. Chun *et al.* [9] measured heat trans-

fer characteristics of γ -Al₂O₃/transformer oil nanofluid in a double tube heat exchanger system under laminar flow condition and showed significant heat transfer enhancement possibly due to high concentration of nanoparticles in the thermal boundary layer at the wall side through the migration of nanoparticles. Experimental studies of pressure drop and convective heat transfer of TiO₂/water nanofluid in a double pipe heat exchanger are reported by Duangthongsuk *et al.* [10]. They showed that the convective heat transfer coefficient of nanofluid is slightly higher than that of the base liquid. Fard *et al.* [11] numerically and experimentally investigated the heat transfer characteristics of ZnO/water nanofluid in a concentric tube heat exchanger under laminar flow condition and showed that the heat transfer coefficient of nanofluid was 14% higher than base fluid. Zamzamian *et al.* [12] experimentally investigated forced convective heat transfer coefficient of Al₂O₃/EG and CuO/EG nanofluids in a double pipe heat exchanger under turbulent flow. However, any theoretical or experimental investigation on the use of nanofluid as secondary fluid in refrigeration or heat pump system is scarce in the open literatures.

In the present work, a nanofluid cooled double tube heat exchanger (gas cooler) has been modeled and simulated for transcritical CO₂ refrigeration system. To take care of highly variable heat transfer characteristics (due to the sharp variation of CO₂ properties near pseudocritical region), the lengthwise discretization of gas cooler has been incorporated in the model. Effects of operating pressure and temperature, mass flow rate, nanofluid variety, and particle volume concentration on the performance have been studied as well.

Theoretical modeling and simulation

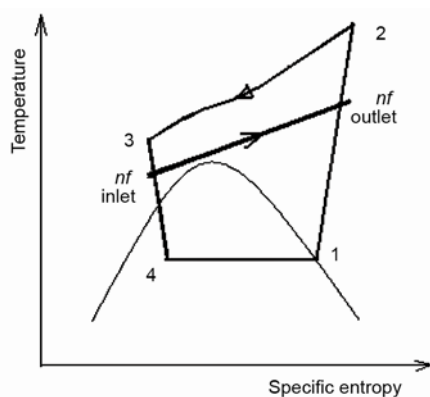


Figure 1. *T-s* diagram of transcritical CO₂ refrigeration cycle

and evaporation/heat addition (4-1) to get the useful refrigeration effect.

It may be noted that the CO₂ properties variations in gas cooler is very significant and hence to take care of this, the lengthwise discretization technique has been successfully applied for gas cooler of double pipe, fin-tube compact heat exchanger as well as shell and tube types [5, 13, 14] in transcritical CO₂ system. In the present study also, to consider the lengthwise highly property variation, the double tube heat exchanger (gas cooler) has been discretized and mass, momentum, and energy conservation equations have been applied to each segment. The following assumptions have been made in the analysis:

- compressor inlet condition is dry-saturated,

The nanofluid (nf) cooled concentric double tube gas cooler considered in this study has following dimensions: inner and outer diameters of inner tube are 4.75 mm and 6.35 mm, respectively, inner diameter of outer tube is 10 mm, and tube length is 14 m [5]. The heat exchanger is counter-flow type, where the nanofluid flows through the inner tube and the refrigerant flows through the annulus. As the effect of using nanofluid as a cooling medium in gas cooler on the cycle performance improvement is the main aim of the present study, only gas cooler has been modeled. The temperature entropy diagram of transcritical CO₂ refrigeration cycle is shown in fig. 1 with non-isentropic compression (1-2), gas cooling/heat rejection through double-tube gas cooler (2-3), isenthalpic expansion (3-4),

- heat transfer with the ambient is negligible, and
- only single-phase heat transfer occurs for nanofluid.

The double tube gas cooler has been equally divided into number of segments of length ΔL as shown in fig. 2. Employing log mean temperature difference expression, heat transfer in i^{th} segment of the gas cooler is given by [5]:

$$Q^i = (UA)^i \frac{(T_r^i - T_{nf}^i) - (T_r^{i+1} - T_{nf}^{i+1})}{\ln[(T_r^i - T_{nf}^i)/(T_r^{i+1} - T_{nf}^{i+1})]} \quad (1)$$

The overall heat transfer coefficient and heat transfer area product is determined by the equation [5]:

$$\frac{1}{UA} = \frac{1}{A_{nf}\alpha_{nf}} + \frac{\ln(d_o/d_i)}{2\pi k_i \Delta L} + \frac{1}{A_r\alpha_r} \quad (2)$$

Additionally, energy balance in i^{th} segment of gas cooler for both the fluids yield [5]:

$$Q^i = \dot{m}_r (h_r^i - h_r^{i+1}) = \dot{m}_{nf} c_{p,nf} (T_{nf}^i - T_{nf}^{i+1}) \quad (3)$$

For supercritical CO₂ cooling in annulus, to take care of large variation of CO₂ properties in the radial direction, Pitla *et al.* [15] proposed a modified correlation incorporating both bulk and wall properties. This correlation, used for gas cooler model, is given by:

$$Nu_r = \left(\frac{Nu_{rb} + Nu_{rt}}{2} \right) \frac{k_{rt}}{k_{rb}} \quad (4)$$

Here, Nu_{rb} and Nu_{rt} are the Nusselt numbers at bulk and wall temperature, respectively, predicted by Gnielinski equation within the range $2300 < Re < 10^6$ and $0.6 < Pr < 10^5$.

$$Nu = \frac{(f/8)(Re - 1000)Pr}{1.07 + 12.7(f/8)^{0.5}(Pr^{2/3} - 1)} \quad (5)$$

where f is the friction factor given by:

$$f = \frac{1}{[0.79 \ln(Re) - 1.64]^2}, \quad Re = \frac{4\dot{m}_r}{\pi(D_i - d_o)\mu_{rb}} \quad (6)$$

Hence, heat transfer coefficient is given by:

$$\alpha_r = \frac{k_{rb}}{d_e} Nu_r \quad (7)$$

where the equivalent diameter of annulus side d_e is given by:

$$d_e = \frac{D_i^2 - d_o^2}{d_o} \quad (8)$$

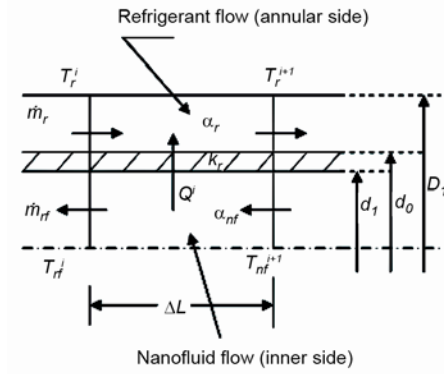


Figure 2. A computational segment of gas cooler

To evaluate the heat transfer coefficient of nanofluid for turbulent flow, Xuan and Li correlation [16] has been used, which is given by ($10^4 \leq \text{Re}_{\text{nf}} \leq 2.5 \cdot 10^4$, $\phi \leq 2\%$):

$$\frac{\alpha_{\text{nf}} d_i}{k_{\text{nf}}} = 0.0059(1 + 7.6286\phi^{0.6886} \text{Pe}_p^{0.001}) \text{Re}_{\text{nf}}^{0.9238} \text{Pr}_{\text{nf}}^{0.4} \quad (9)$$

The Reynolds number, the Prandtl number, and the particle Peclet number for nanofluid are defined, respectively, as [14]:

$$\text{Re}_{\text{nf}} = \frac{\rho_{\text{nf}} u_m d_i}{\mu_{\text{nf}}} = \frac{4\dot{m}_{\text{nf}}}{\pi d_i \mu_{\text{nf}}} \quad (10)$$

$$\text{Pr}_{\text{nf}} = \frac{c_{p,\text{nf}} \mu_{\text{nf}}}{k_{\text{nf}}} \quad (11)$$

$$\text{Pe}_d = \frac{\rho_{\text{nf}} c_{p,\text{nf}} u_m d_p}{k_{\text{nf}}} = \frac{4\dot{m}_{\text{nf}} c_{p,\text{nf}} d_p}{\pi d_i^2 k_{\text{nf}}} \quad (12)$$

The effective density and the effective specific heat of the nanofluid can be calculated from relations [14]:

$$\rho_{\text{nf}} = (1 - \phi)\rho_w + \phi\rho_p \quad (13)$$

$$(\rho c_p)_{\text{nf}} = (1 - \phi)(\rho c_p)_w + \phi(\rho c_p)_p \quad (14)$$

The viscosity of nanofluid has been calculated by Einstein's equation [14], given by:

$$\mu_{\text{nf}} = (1 + 2.5\phi)\mu_w \quad (15)$$

The effective thermal conductivity of the nanofluid has been calculated by Yu and Choi equation [17], given by ($\phi \leq 5\%$):

$$k_{\text{nf}} = \frac{k_p + 2k_w + 2(k_p - k_w)(1 + \beta)^3 \phi}{k_p + 2k_w - (k_p - k_w)(1 + \beta)^3 \phi} k_w \quad (16)$$

where $\beta = 0.1$ and the temperature dependent transport properties: dynamic viscosity and thermal conductivity of water are given by [14], respectively ($0 \leq t \leq 90$ °C):

$$\mu_w = 0.00157 - 4.68 \cdot 10^{-5} t + 5.71 \cdot 10^{-7} t^2 - 2.52 \cdot 10^{-9} t^3 \quad (17)$$

$$k_w = 0.5473 + 2.14 \cdot 10^{-3} t - 9.6737 \cdot 10^{-6} t^2 \quad (18)$$

The annulus (CO₂)-side pressure drop in each segment is predicted by [13]:

$$p^i - p^{i+1} = \frac{8\dot{m}_r^2}{\pi^2 (D_i^2 - d_o^2)^2 \rho_r^i} \left(f_r^i \frac{\Delta L}{D_i - d_o} + 1.2 \right) \quad (19)$$

where the friction factor is given in [13].

The tube side pressure drop is given by [18]:

$$\Delta p_{nf}^i = f_{nf} \frac{\Delta L}{d_i} \frac{8 \dot{m}_{nf}^2}{\pi^2 d_i^4 \rho_{nf}^i} \quad (20)$$

The friction factor correlation of nanofluid, which has been established for Al₂O₃, SiO₂, and CuO nanofluids [18], is given by (4000 ≤ Re ≤ 16000, φ ≤ 6%):

$$f_{nf} = 0.3164 \text{Re}_{nf}^{-0.25} \left(\frac{\rho_{nf}}{\rho_w} \right)^{0.797} \left(\frac{\mu_{nf}}{\mu_w} \right)^{0.108} \quad (21)$$

Assuming 85% pump efficiency [14], the pump work is given by:

$$W_p = \frac{\dot{m}_{nf}}{0.85} \sum_n \frac{\Delta p_{nf}}{\rho_{nf}} \quad (22)$$

The cooling capacity and compressor power, respectively, are given by:

$$Q_e = \dot{m}_r (h_1 - h_3) \quad (23)$$

$$W_c = \dot{m}_r (h_2 - h_1) \quad (24)$$

Finally, the effective system performance using nanofluid is given by:

$$COP = \frac{Q_e}{W_c + W_p} \quad (25)$$

Based on previously developed simulation model for CO₂ heat pump (validation with experimental data showed good agreement [5]), a computer code has been developed to simulate the double tube heat exchanger (gas cooler) in transcritical CO₂ refrigeration system at various operating conditions. Author's own developed subroutine 'CO2PROP' [5] has been integrated with code to estimate the thermodynamic and transport properties of CO₂ in subcritical as well as supercritical zones. To consider the sharp property variation, the entire length of the gas cooler has been divided equally into several discrete segments (segment length ΔL = total length/n) and each segment has been treated as a counter flow heat exchanger. In each segment, heat transfer coefficients for both refrigerant and nanofluid are calculated based on mean values. This way, the gas cooler is made equivalent to a number of counter flow heat exchangers arranged in series and the combined heat transfer of all the segments is the total heat transfer of the gas cooler. Therefore, fast changing properties of CO₂ have been modeled accurately in the gas cooler. For given evaporator temperature and gas cooler pressure, gas cooler inlet enthalpy and temperature of CO₂ are calculated by assumed 75% compressor efficiency to accommodate non-isentropic compression. For given, gas cooler geometry, CO₂ and nanofluid mass flow rates and nanofluid properties [14], gas cooler model is used to calculate outlet temperatures and capacity. It may be noted that the mass flow rates are so selected that the both CO₂ and nanofluid flows will be turbulent with moderate Re. The heat transfer coefficient for pure water has been calculated by Gnieliniski equation for liquids [14]. The effective iteration technique has been used in the code to get good accuracy of results. Finally, performance parameters have been calculated for various input parameters.

Results and discussion

The performances of the double tube gas cooler as well as the CO₂ refrigeration cycle are presented for various compressor discharge pressures (90 to 110 bar), nanofluid mass

flow rates (0.015 to 0.05 kg/s) and nanoparticle volume fraction in nanofluid (0.1 to 2%). Unless otherwise stated the mean values are refrigerant pressure at gas cooler inlet of 100 bar, nanofluid mass flow rate of 0.03 kg/s, and nano-particle volume concentration of 1%. The CO₂ temperature in evaporator, CO₂ mass flow rate and nanofluid inlet temperature to gas cooler have been taken as 5 °C, 0.02 kg/s, and 30 °C, respectively. The nanoparticle diameter is taken as 50 nm. The heat exchanger (gas cooler) effectiveness and cooling *COP* are suitably plotted to illustrate the various performance trends.

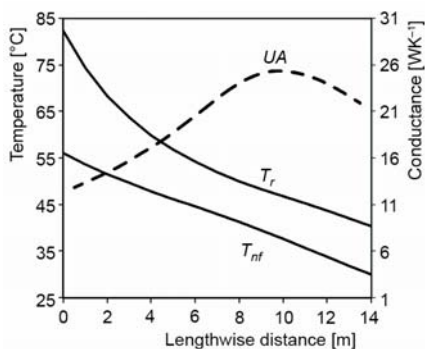


Figure 3. Lengthwise variations of temperatures and conductance

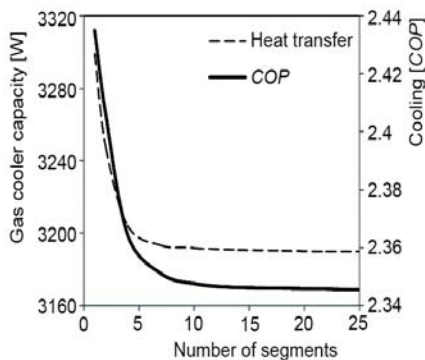


Figure 4. Grid dependent test

The grid dependent test results show that with the increase in number of segments, the result initially changes rapidly and then slowly merges to same value and that has been happened approximately after 15 number of segments. Hence the number of segment has been taken as 20 for simulation, where result becomes nearly independent on number of segments (*e. g.* error in *COP* is less than 0.1% per segment). For example, the lengthwise variations of CO₂ and nanofluid temperatures, and overall heat transfer coefficient are shown in fig. 3 for mean operating conditions with alumina-water nanofluid, which shows the significant variation of overall heat transfer coefficient and the maximum value is at the pseudocritical temperature and confirms to need of discretization, and also the grid dependent variation is shown in fig. 4. Hence, similar to previous studies [5, 13, 14], the discretization has been effectively used in gas cooler to get the better accuracy of results.

The performance comparison of double-tube gas cooler with four nanofluids: alumina-water (Al₂O₃-H₂O), titanium dioxide-water (TiO₂-H₂O), copper oxide-water (CuO-H₂O), and copper-water (Cu-H₂O) as coolant is shown in tab. 1 for mean operating conditions. Results show that the heat transfer in double-tube gas cooler increases by using nanofluid due to increase in heat transfer properties (overall heat transfer coefficient) and hence the effectiveness of gas cooler also increases. As a result, the cooling capacity increases due to decrease in CO₂ exit temperature in gas cooler. It may be noted that the density, viscosity, and thermal conductivity increases and specific heat capacity decreases by using nanoparticle, and hence both *Re* and *Pr* decreases for same mass flow rate. However, the heat transfer coefficient increases compared to base fluid due to increase in thermal conductivity and probably improvement of heat transport properties due to several slip mechanisms of nanofluids. Pressure drops by using nanofluids are similar to that of pure water as a coolant in gas cooler, which agreed with the experimental results with nanofluid [16]. However, the pump work decreases slightly with the use of nanoparticle in water due to increase in fluid density. As a result, the cooling *COP* of CO₂ cycle improves by using nanofluid as coolant in double-tube gas cooler. It may be noted that the specific heat for alumina nanofluid is maximum and the thermal conductivity for Cu nanofluid is maximum and as a result, *Pr* and *Pe* are maximum for alumina nanofluid

for same particle volume concentration, and ultimately, Nusselt number and heat transfer coefficient are maximum for alumina nanofluid. Hence, as shown in tab. 1, the increase in heat rejection and effectiveness are maximum for alumina followed by TiO₂, CuO, and Cu nanofluids. On the other hand, the pressure

Table 1. Performance comparison ($p_2 = 100$ bar, $\dot{m}_{nf} = 0.03$ kg/s, $\phi = 1\%$)

Parameters	Al ₂ O ₃	TiO ₂	CuO	Cu
Discharge temperature, [°C]	82.3	82.3	82.3	82.3
Gas cooler CO ₂ exit temperature, [°C]	40.38	40.45	40.61	40.82
Nanofluid outlet temperature, [°C]	56.20	56.28	56.65	56.98
Nanofluid pressure drop, [bar]	1.2720	1.2705	1.2654	1.2562
Cooling capacity, [W]	2245.7	2239.9	2217.7	2192.0
Compressor work, [W]	950.4	950.4	950.4	950.4
Pump work, [W]	4.371	4.347	4.227	4.108
Increase in pump work, [%]	-1.5	-2.0	-4.0	-6.5
Heat rejection in gas cooler, [W]	3196.2	3190.3	3168.1	3142.5
Increase in heat rejection, [%]	3.3	3.2	2.9	2.5
Gas cooler effectiveness, [%]	80.16	80.07	79.71	79.30
Increase in effectiveness	0.67	0.64	0.55	0.45
Cooling COP	2.3521	2.3460	2.3230	2.2965
COP improvement, [%]	4.36	4.16	3.66	3.03

drop and pumping power are maximal for alumina nanofluid. The cooling COP is maximal for alumina followed by TiO₂, CuO, and Cu nanofluids for same operating conditions.

The variations of double tube gas cooler effectiveness and cooling COP with compressor discharge pressure are shown in figs. 5 and 6, respectively, using studied nanofluids. The CO₂ inlet temperature increases with increase in compressor discharge pressure and hence, the heat rejection in gas cooler increases due to increase in effective heat transfer temperature difference. Whereas, the effectiveness of gas cooler decreases with increase in CO₂ pressure due to degradation of heat transfer properties as go away from critical pressure. However, due to distinct cycle behavior of transcritical CO₂ cycle [5], the cooling COP of the cycle increases with increase in CO₂ inlet pressure to gas cooler. As the compressor discharge pressure has no effect on nanofluid heat transfer or flow properties, the performance deviations with all four nanofluids are independent on the CO₂ inlet pressure to gas cooler as shown in figures.

The variations of double tube gas cooler effectiveness and cooling COP with nanofluid mass flow rate are shown in figs. 7 and 8, respectively, using studied nanofluids.

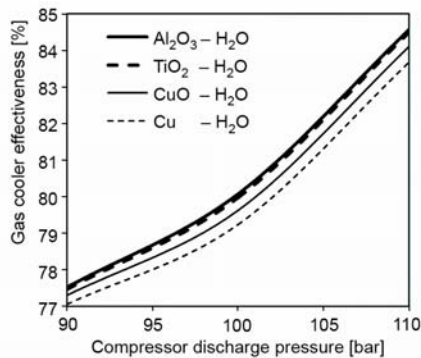


Figure 5. Variation of gas cooler effectiveness with compressor discharge pressure

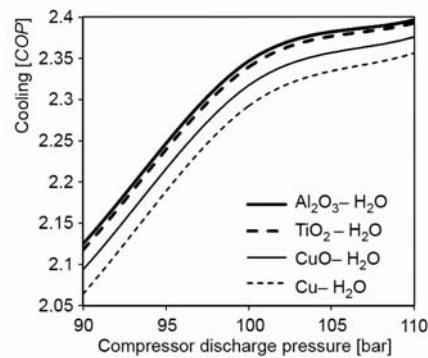


Figure 6. Variation of cooling COP with compressor discharge pressure

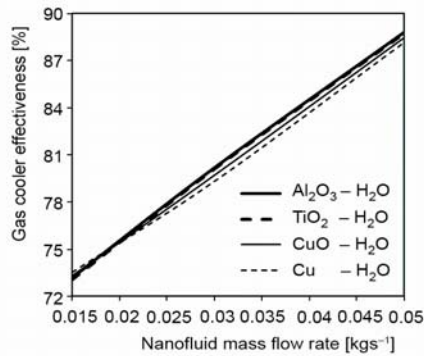


Figure 7. Variation of gas cooler effectiveness with nanofluid mass flow rate

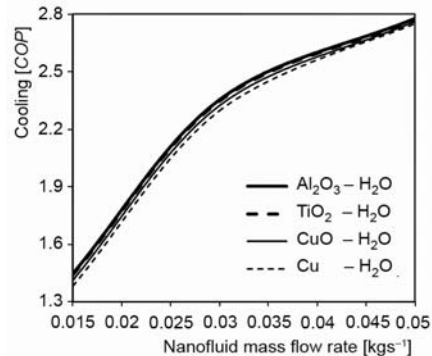


Figure 8. Variation of cooling *COP* with nanofluid mass flow rate

The nanofluid heat transfer coefficient increases with increase in nanofluid mass flow rate due to increase in Reynolds number, and hence overall heat transfer coefficient increases resulting in increase in gas cooler effectiveness. The cooling capacity increases due to increase in gas cooler effectiveness and mass flow rate. The compressor work remain unchanged and the pumping power increases significantly with increase in mass flow rate, however, negligibly small compared to compressor work. Hence, the cooling *COP* of the cycle increases with increase in nanofluid mass flow rate. As the nanofluid properties are independent on nanofluid mass flow rate, the effect on performance deviations with all four nanofluids is negligible.

The variations of double tube gas cooler effectiveness and cooling *COP* with particle volume fraction are shown in figs. 9 and 10, respectively, using studied nanofluids. The specific heat capacity decreases and viscosity increases, whereas the thermal conductivity increases initially rapidly and then slowly, and hence the heat transfer coefficient initially increases rapidly and then negligibly with increase in particle volume fraction. As a result, the effectiveness of gas cooler and cooling capacity increases with increase in volume fraction. The pressure drop increases but pumping power decreases negligibly with increase in volume fraction due to increase in density. Hence, the cooling *COP* increases rapidly and then slowly with increase in volume fraction. The deviation of nanofluid properties increases and hence the deviation of *COP* values increases with increase in volume fraction. Study shows that the particle volume fraction may be optimize based on cooling *COP*, however, the higher optimum volume fraction will cause more stability problem of nanofluid.

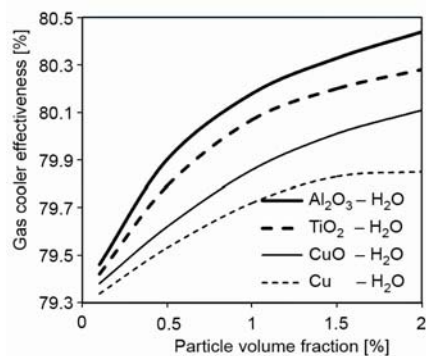


Figure 9. Variation of gas cooler effectiveness with particle volume fraction

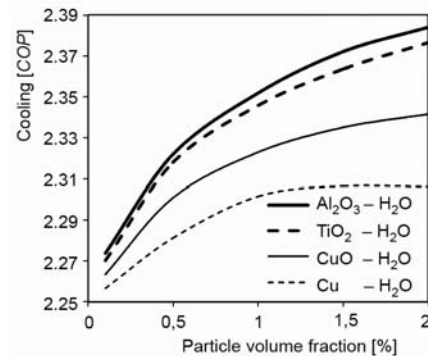


Figure 10. Variation of cooling *COP* with particle volume fraction

The improvement of performance has been observed for CO₂ cycle by using nanofluid as a coolant instead of water in double tube gas cooler and the improvement is dependent on nanofluids kind as well as operating conditions. Figure 11 predicts the percentage of cooling *COP* improvements using nanofluid compared to water for various operating conditions. The percentage of improvement is also dependent on absolute value. For the given ranges of study, the maximum cooling *COP* improvement of transcritical CO₂ cycle for Al₂O₃-H₂O is observed as 25.4%, whereas that for TiO₂-H₂O is 23.8%, for CuO-H₂O – 20.2%, and for Cu-H₂O – 16.2%. It may be noted that the large deviation between various correlations for nanofluid heat transfer coefficient has been observed [19], which leads to performance deviation. Interestingly, system performance will be obviously improved without extra pump power by using nanofluids.

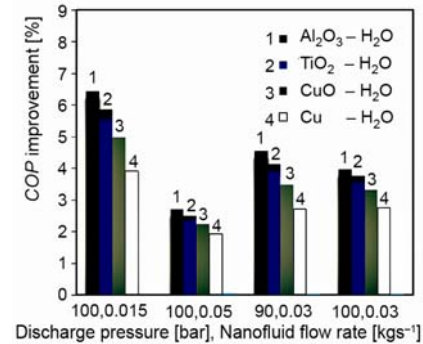


Figure 11. Comparison of *COP* improvements

Conclusions

The nanofluid cooled double tube gas cooler has been modeled and simulated for transcritical CO₂ refrigeration system to study the effects of various operating parameters on system performance improvement. From the results and discussion, the following conclusion can be made.

- Use of nanofluid as coolant in double-tube gas cooler improves the gas cooler effectiveness, cooling capacity, and *COP* without penalty of pumping.
- The CO₂ system yields best performance using Al₂O₃-H₂O as coolant in double-tube gas cooler followed by TiO₂-H₂O, CuO-H₂O, and Cu-H₂O.
- The performance variation shows the similar trends with respect to gas cooler inlet pressure and nanofluid mass flow rates for all nanofluids, however, the performance deviation increases with increase in particle volume fraction.
- The cooling *COP* increases with increase in compressor discharge pressure, nanofluid mass flow rate and particle volume fraction.
- The maximum cooling *COP* improvement of transcritical CO₂ cycle for Al₂O₃-H₂O is 25.4%, whereas that for TiO₂-H₂O is 23.8%, for CuO-H₂O – 20.2%, and for Cu-H₂O – 16.2% for the given ranges of study.

Nomenclature

A – heat transfer area, [m²]
c_p – specific heat capacity, [kJkg⁻¹K⁻¹]
d, D – inner and outer tube diameter, [m]
f – friction factor, [–]
h – specific enthalpy, [kJkg⁻¹]
k – thermal conductivity, [Wm⁻¹K⁻¹]
 ΔL – segmented length, [m]
 \dot{m} – mass flow rate, [kgs⁻¹]
n – number of segments, [–]
Nu – Nusselt number, [–]
p – pressure, [bar]
Pe – Peclet number, [–]
Pr – Prandtl number, [–]

Q – heat transfer rate, [W]
Re – Reynolds number, [–]
t, T – temperature, [°C, K]
U – overall heat transfer coefficient, [Wm⁻²K⁻¹]
u_m – mean velocity, [ms⁻¹]
W – work transfer, [W]

Greek symbols

α – heat transfer coefficient, [Wm⁻²K⁻¹]
 μ – dynamic viscosity, [Nsm⁻²]
 ρ – fluid density, [kgm⁻³]
 ϕ – particle volume fraction, [–]

Subscripts

b – bulk property
c – compressor
i – inner
nf – nanofluid

o – outer
p – nanoparticle, pump
r – refrigerant
t – tube, tube wall
w – water

References

- [1] Kim, M., *et al.*, Fundamental Process and System Design Issues in CO₂ Vapour Compression Systems, *Progress in Energy and Combustion Science*, 30 (2004), 2, pp. 119-174
- [2] Austin, B. T., Sumathy, K., Transcritical Carbon Dioxide Heat Pump Systems: A Review, *Renewable and Sustainable Energy Reviews*, 15 (2011), 8, pp. 4013-4029
- [3] Rigola, J., *et al.*, Numerical Simulation and Experimental Validation of Vapour Compression Refrigeration Systems. Special Emphasis on CO₂ Trans-Critical Cycles, *International Journal of Refrigeration*, 28 (2005), 8, pp. 1225-1237
- [4] Cabello R., *et al.*, Experimental Evaluation of the Energy Efficiency of a CO₂ Refrigerating Plant Working in Transcritical Conditions, *Applied Thermal Engineering*, 28 (2008), 13, pp. 1596-1604
- [5] Sarkar, J., *et al.*, A Transcritical CO₂ Heat Pump for Simultaneous Water Cooling and Heating: Test Results and Model Validation, *International Journal of Energy Research*, 33 (2009), 1, pp. 100-109
- [6] Sanchez, D., *et al.*, Influence of the Superheat Associated to a Semihermetic Compressor of a Transcritical CO₂ Refrigeration Plant, *Applied Thermal Engineering*, 30 (2010), 4, pp. 302-309
- [7] Wong, K. V., Leon, O. D., Applications of Nanofluids: Current and Future, *Advances in Mechanical Engineering*, (2010), Article ID 519659
- [8] Liu, L., *et al.*, The Potential Impact of Nano-Fluid Enhancements on the Performance of Heat Exchangers, *Heat Transfer Engineering*, 33 (2011), 1, pp. 31-41
- [9] Chun, B. H., *et al.*, Effect of Alumina Nanoparticles in the Fluid on Heat Transfer in Double-Pipe Heat Exchanger System, *Korean Journal of Chemical Engineering*, 25 (2008), 5, pp. 966-971
- [10] Duangthongsuk, W., Wongwises, S., Heat Transfer Enhancement and Pressure Drop Characteristics of TiO₂-Water Nanofluid in a Double-Tube Counter Flow Heat Exchanger, *International Journal of Heat and Mass Transfer*, 52 (2009), 7-8, pp. 2059-2067
- [11] Fard, M. H., *et al.*, Numerical and Experimental Investigation of Heat Transfer of ZnO/Water Nanofluid in the Concentric Tube and Plate Heat Exchangers, *Thermal Science*, 15 (2011), 1, pp. 183-194
- [12] Zamzamin, A., *et al.*, Experimental Investigation of Forced Convective Heat Transfer Coefficient in Nanofluids of Al₂O₃/EG and CuO/EG in a Double Pipe and Plate Heat Exchangers under Turbulent Flow, *Experimental Thermal and Fluid Science*, 35 (2011), 3, pp. 495-502
- [13] Sarkar, J., *et al.*, CO₂ Heat Pump Dryer: Part 1. Mathematical Model and Simulation, *Drying Technology*, 24 (2006), 12, pp. 1583-1591
- [14] Sarkar, J., Performance of Nanofluid-Cooled Shell and Tube Gas Cooler in Transcritical CO₂ Refrigeration Systems, *Applied Thermal Engineering*, 31 (2011), 14-15, pp. 2541-2548
- [15] Pitla, S. S., *et al.*, New Correlation to Predict the Heat Transfer Coefficient during in-Tube Cooling of Turbulent Supercritical CO₂, *International Journal of Refrigeration*, 25 (2002), pp. 887-895
- [16] Xuan, Y., Li, Q., Investigation of Convective Heat Transfer and Flow Features of Nanofluids, *Journal of Heat Transfer*, 125 (2003), 7, pp. 151-155
- [17] Yu, W., Choi, S. U. S., The Role of Interfacial Layers in the Enhanced Thermal Conductivity of Nanofluids: a Renovated Maxwell Model, *Journal of Nanoparticle Research*, 5 (2003), 1-2, pp. 167-171
- [18] Vajjha, R. S., *et al.*, Development of New Correlations for Convective Heat Transfer and Friction Factor in Turbulent Regime for Nanofluids, *International Journal of Heat and Mass Transfer*, 53 (2010), 21-22, pp. 4607-4618
- [19] Sarkar, J., A Critical Review on Convective Heat Transfer Correlations of Nanofluids, *Renewable & Sustainable Energy Reviews*, 15 (2011), 6, pp. 3271-3277