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Assessment of alumina nanofluid as a coolant in double pipe gas cooler for trans-critical CO_2 refrigeration cycle

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Abstract

In this study, the performance of an alumina nanofluid cooled double pipe gas cooler for trans-critical CO_2 refrigeration cycle is theoretically compared to that of water cooled gas cooler. Equal pumping power comparison criterion is adopted besides conventional equal Reynolds number comparison base. Nanofluid is loaded with 0.5%, 1.5% and 2.5% of particle volume fraction under turbulent flow conditions. Drastic variation of thermal and transport properties of CO_2 in the vicinity of pseudo critical temperature is taken care of by employing an appropriate discretization technique. Effect of gas cooler pressure, Reynolds number, pumping power and nanoparticle volume fraction on COP of refrigeration system, gas cooler overall conductance, effectiveness and its capacity has been studied. Results indicate that at equal Reynolds number comparison, performance for alumina nanofluid cooled system is better than that of water cooled system. On the other hand, at equal pumping power comparison basis, the performance of water cooled system is superior. Even at equal mass flow rate comparison criterion, the performance of nanofluid cooled system degrades with increase in particle volume fraction. This study is expected to help to assess the nanofluid as a coolant before expensive experimentation.

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Keywords: Trans-critical CO_2 refrigeration cycle; Gas cooler; Double pipe; Nanofluid; Comparison criteria; Alumina

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1. Introduction

With growing awareness about threat of global warming and ozone depletion, refrigeration and air-conditioning industries are forced to replace otherwise high performing synthetic refrigerants like HFC/CFC by natural ones like water, air, noble gases, ammonia, carbon dioxide (CO_2) etc. [1]. Among natural refrigerants, CO_2 has established itself as a promising long term alternative for both refrigeration & air conditioning and heat pump systems.

Nomenclature

A	Area of cross section (m^2)	Pr	Prandtl number
a	Circumferential area (m^2)	PP	Pumping power (W)
C	Compressor	Rc	Convective resistance ($K\cdot W^{-1}$)
CC	Cooling capacity (W)	Re	Reynolds number
COP	Coefficient of Performance	Rw	Conductive resistance of tube wall ($K\cdot W^{-1}$)
c	Specific heat capacity ($J\cdot kg^{-1}\cdot K^{-1}$)	T	Temperature (^oC)
d	Inner diameter of inside tube (m)	G	Mass flux ($kg\cdot s^{-1}\cdot m^{-2}$)
Exp	Expansion valve	UA	Overall conductance ($W\cdot K^{-1}$)
f	Friction factor	u	Velocity ($m\cdot s^{-1}$)
GWP	Global warming potential ($kg_{CO_{2,equ}} \cdot kg^{-1}_{refrigerant}$)	x	Grid size (m)
H	Heat transfer coefficient ($W\cdot m^{-2}\cdot K^{-1}$)	**Greek letters**	
h	Specific enthalpy ($J\cdot kg^{-1}$)	ρ	Density ($kg\cdot m^{-3}$)
k	Thermal conductivity ($W\cdot m^{-1}\cdot K^{-1}$)	ΔP	Pressure drop (Pa)
$\dot{m}$	Mass flow rate ($kg\cdot s^{-1}$)	$\emptyset$	Particle volume fraction
NF	Nanofluid	μ	Viscosity ($Pa\cdot s$)
NP	Nanoparticle	**Subscripts**	
Nu	Nusselt number	bf/nf	Basefluid/Nanofluid
ODP	Ozone depletion potential	c/r	Coolant/Refrigerant
$\dot{Q}$	Heat transfer rate (W)	wt/bt	Wall temperature/Bulk temperature

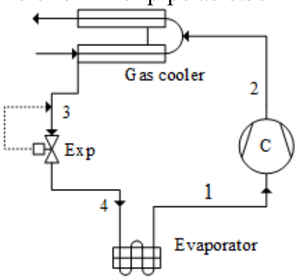
CO_2 as a refrigerant has zero ozone depletion potential (ODP), unit global warming potential (GWP), low price, easy availability, non-flammability, non-toxicity, compatibility with various common materials, equipment compactness owing to high operating pressure and excellent transport properties [2]. Performance of CO_2 system is comparable or even better than high GWP base vapor compression system in sub-critical region, whereas they need to be significantly enhanced when they work in warm climatic conditions [3,4]. This can be attributed to low critical temperature of CO_2, which triggers trans-critical modes for systems operating at high outdoor temperatures. A comparison of CO_2 cycle to conventional vapor compression cycle shows that heat rejection process in CO_2 cycle takes place in supercritical state at high cooling medium temperatures and features the single phase heat transfer in a heat exchanger called as gas cooler. Gas coolers are either air cooled or liquid cooled depending on the type of application. Gupta and Dasgupta [5] investigated air cooled finned tube gas cooler for trans-critical CO_2 refrigeration cycle working in Indian climatic conditions, while Sarkar [6] investigated water cooled gas cooler for simultaneous heating and cooling application. Irrespective of cooling medium, the design of gas cooler is a topic of concern as it need to take care of variable heat transfer characteristics of CO_2 near pseudo critical temperature [7].

Liquid cooled gas coolers or double pipe heat pipe heat exchangers for trans-critical CO_2 refrigeration and heat pump cycles usually employs water as a cooling medium which has inherent low thermal conductivity and this creates opportunity for addition of higher thermal conductivity nano sized solid particles for enhancing heat transfer. Nanofluids (NFs) are fluids having small fraction of nanoparticles (NPs) stably suspended in base fluids like water, ethylene glycol etc. as coined by Choi and Eastman [8]. Overall conductance of NFs is higher than base fluids, however, addition of NPs also increases the effective viscosity of NFs, therefore, to maintain same Reynolds number (Re) of NFs as that of only base fluids, the flow rate of NFs has to be increased. Owing to this, comparing performance of heat exchangers employing NFs to that of base fluids at equal Re might be misleading as reported by Haghighi et al. [9]. Studies reporting comparison among NF and corresponding base fluid heat transfer characteristics can be found from a recent review by Sidik et al. [10]. Purohit et al. [11] numerically evaluated thermal performance of three metal oxide NFs at equal Re, equal mass flow rate and equal discharge comparison criteria. They reported enhancement in thermal performance of NFs only at equal Re comparison criterion. Sarkar [6,12] reported enhancement in effectiveness for trans-critical CO_2 based NF cooled gas cooler and claimed increase in system COP and cooling capacity without penalty of pumping power. Ndoye et al. [13] numerically investigated NF used as cooling medium in secondary loop of refrigeration systems under equal Re comparison criterion. They reported increase in performance factor for NFs irrespective of flow regime.

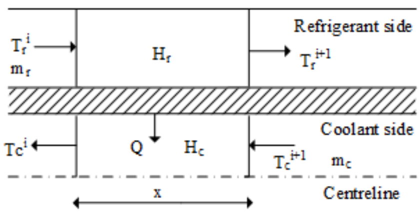
In this study Al_2O_3/water NF is theoretically investigated as a cooling medium for refrigerant flowing in a gas cooler of trans-critical CO_2 refrigeration cycle under equal Re and equal pumping power comparison criteria. Beside conventional equal Re comparison basis, same pumping power comparison criterion provides more practical basis directly attributed to the cost of operation of cooling systems [9]. Particle loading is varied from 0.5 % to 2.5 % under turbulent flow conditions. Single phase modeling approach is employed following strong justification, as discussed in section 2. Overall heat transfer coefficient of gas cooler and its effectiveness along with system COP and its cooling capacity are investigated for tested NF. To ensure the accuracy in modelling, appropriate models are utilized for calculating thermo-physical properties of NF, as discussed in section 2.

2. Mathematical modelling

The basic trans-critical CO_2 refrigeration cycle is shown in Fig. 1 (a). Process 1-2 is adiabatic compression in compressor, followed by heat rejection in gas cooler (process 2-3) then isenthalpic expansion i.e. process 3-4 in an expansion valve and finally evaporation of refrigerant in evaporator as given by process 4-1. The counter flow type double pipe gas cooler dimensions assumed follows: tube length of 14 m, inner diameter of outer pipe as 10 mm and outer and inner diameter of inner pipe as 6.35 mm and 4.72 mm respectively [6].



1 (a) Basic trans-critical CO_2 refrigeration cycle



1 (b) Discretization scheme for double pipe gas cooler

Fig. 1 Basic trans-critical CO_2 refrigeration cycle and discretization scheme for gas cooler

Coolant (water or NF) is fed into the tube side while supercritical refrigerant flows in the annulus side. There are two main difficulties in modelling: first is resolving large thermo-physical property variations of CO_2 in both axial and radial directions in pseudocritical region and second simulation of the stable suspension of base fluid and NPs. The former is taken care by employing correlation developed by Pitla et al. [7], while for the latter single phase approach is adopted [11]. One dimensional heat transfer and fluid flow for both coolant (water and NF) and refrigerant is modelled separately. The fundamental conservation equations are discretised using a finite difference scheme as shown in Fig. 1 (b). The model is first resolved for one element then developed for complete geometry element by element.

2.1 Refrigerant side modelling

The mass, momentum and energy conservation equations for double pipe gas cooler in discretized form for a single element as shown in Fig. 1 (b) is given in equations (1) to (3). For each element, there is a heat balance between coolant and the refrigerant as shown in equation (4).

$$(m_r)^{i+1}-(m_r)^i = 0 \tag{1}$$

$$(m_r \cdot u_r)^{i+1} - (m_r \cdot u_r)^i + A_r \cdot \Delta P_r = 0 \quad (2)$$

$$(m_r \cdot h_r)^{i+1} - (m_r \cdot h_r)^i + x \cdot a_r = 0 \quad (3)$$

$$\dot{m}_r \cdot c_r \cdot \left(T_r^i - T_r^{i+1}\right) = \dot{m}_c \cdot c_r \cdot \left(T_c^i - T_c^{i+1}\right) \quad (4)$$

Pitla et al. [7] proposed a correlation for estimating mean Nusselt number for CO_2 as given by equation (5) and the same is adopted here. In equation (5), Nu_{bt} and Nu_{wt} are the Nusselt number calculated at bulk and wall temperature based thermo-physical properties respectively. Bulk and wall temperatures are resolved for every element employing appropriate iterative method by initially assuming them as the refrigerant and coolant temperature respectively. Nusselt number for each case is calculated using correlation given by Gneilinski [14] given in equation (6) in which f is the friction factor estimated as given by equation (7). Further, equations (8) and (9) are utilized to estimate heat transfer coefficient and total pressure drop respectively for refrigerant side.

$$Nu_r = \{0.5 \cdot (Nu_{wt} + Nu_{bt})\} \cdot (K_{wt}) \cdot (K_{bt}^{-1}) \quad (5)$$

$$Nu = \{(f/8) \cdot (Re-1000) \cdot Pr\} \cdot \{(12.7 \cdot (f/8)^{0.5} \cdot (Pr^{(2/3)}-1)) + 1.07\}^{-1} \quad (6)$$

$$f = (0.79 \cdot \ln(Re) - 1.64)^{-2} \quad (7)$$

$$H_r = Nu_r \cdot K_r \cdot (d)^{-1} \quad (8)$$

$$\Delta P_r = \left(f \cdot G_r^2 \cdot x\right) \cdot (2 \cdot \rho_r \cdot d)^{-1} \quad (9)$$

2.2 Coolant side modelling

For water, temperature dependent thermo-physical properties are considered in this analysis as given in equations (10) to (13) [15]. Further, Nusselt number, heat transfer coefficient and pressure drop for water are calculated using correlations given previously in equations (6) to (9). For modelling NF, single phase approach is employed. In this approach, mixture of NPs and base fluid (water in our case) are treated as homogenous solution assuming no relative motion between NPs and basefluid particles. Buongiorno [16] investigated possible seven slip mechanisms for two phase behavior of NFs and reported Brownian motion and thermophoresis to be the most dominating mechanisms. Similar conclusion was reported by Sadeghi et al. [17]. However, Ahmed et al. [18] reported that for NFs flow with Re higher than 100, both Brownian motion and thermophoresis may be safely neglected. This implies that adopting single phase approach for our range of investigation (i.e. turbulent flow conditions) is justified. However, two phase modelling approach is more accurate but complex and time consuming [19,20]. Haghighi et al. [9] claimed that heat transfer and pressure drop behavior of NFs can also be predicted within 10 % error using classical correlations developed for base fluid provided that appropriate thermo-physical properties of NFs are utilized. Following this, Nusselt number, heat transfer coefficient and pressure drop of NFs are also calculated using equations (6) to (9).

Density and specific heat of Al_2O_3/water NF can be estimated accurately by the weighted average of the densities and specific heats of the NPs and the base fluid respectively as shown in equation (14) and (15) [9,11]. Buongiorno et al. [21] normalized experimental data of NF's thermal conductivity from thirty organizations worldwide and reported that predictions from Maxwell model [22], as given in equation (16), to be in good agreement with and the same is utilized in this study. Correlation proposed by Maiga et al. [23] for effective viscosity for alumina NFs based on scarce experimental data collected from literature is adopted here as well, given in equation (17). The absolute values of thermo-physical properties of alumina NPs are taken from Purohit et al. [11]. The overall heat transfer coefficient for gas cooler is calculated as given by equation (18). The pump work required for coolant as shown in equation (19), is calculated as product of total pressure drop and discharge. Finally, cooling capacity, compressor work and COP of refrigeration system are calculated as given by equation (20), (21) and (22) respectively.

$$\rho_{bf} = 999.78 + (0.068 \cdot T) - (0.0107 \cdot T^2) + (0.00082 \cdot T^3) - (2.303 \cdot 10^{-5} \cdot T^4) \quad (10)$$

$$K_{bf} = 0.56112 + (0.00193 \cdot T) - (2.601 \cdot 10^{-6} \cdot T^2) - (6.08 \cdot 10^{-8} \cdot T^3) \quad (11)$$

$$\mu_{bf} = 0.00169 - (4.25 \cdot 10^{-5} \cdot T) + (4.92 \cdot 10^{-7} \cdot T^2) - (2.09 \cdot 10^{-9} \cdot T^3) \quad (12)$$

$$c_{bf} = 4217.4-(5.61 \cdot T)+(1.299 \cdot T^{1.52})-(0.11 \cdot T^2)+(4149.6 \cdot 10^{-6} \cdot T^{2.5}) \quad (13)$$

$$\rho_{nf} = \rho_{bf} \cdot (1-\emptyset)+\rho_{NP} \cdot \emptyset \quad (14)$$

$$c_{nf} = ((1-\emptyset) \cdot (\rho \cdot c)_{bf}+\emptyset \cdot (\rho \cdot c)_{NP}) \cdot (\rho_{nf})^{-1} \quad (15)$$

$$K_{nf} = (K_{NP}+2 \cdot K_{bf}+2 \cdot (K_{NP}-K_{bf}) \cdot \emptyset) \cdot \{K_{NP}+2 \cdot K_{bf} - (K_{NP}-K_{bf}) \cdot \emptyset\}^{-1} \cdot K_{bf} \quad (16)$$

$$\mu_{nf} = \mu_{bf} \cdot (1+7.3 \cdot \emptyset+123 \cdot \emptyset^2) \quad (17)$$

$$UA = (Rc_r+Rw+Rc_c)^{-1} \quad (18)$$

$$PP = \Delta P_r \cdot (A_c \cdot u_c) \quad (19)$$

$$CC = \dot{m}_r \cdot (h_1-h_4) \quad (20)$$

$$\dot{W} = \dot{m}_r \cdot (h_2-h_1) \quad (21)$$

$$COP = (CC)/(\dot{W}+PP) \quad (22)$$

The simulation is developed in MATLAB environment and REFPROP is utilized for estimating refrigerant properties. The double pipe gas cooler is divided into number of segments in order to resolve large varying CO_2 thermal and transport properties. Fig. 2 shows the grid independence test and it can be observed that beyond 20 number of divisions, the COP and the gas cooler capacity values does not vary more than 1%. The evaporator temperature of 0^oC and compressor efficiency of 0.65 is adopted throughout investigation. The mass flow of refrigerant for entire range of investigation is taken as 0.02 kg. s^{-1}. Gas cooler pressure and coolant inlet temperatures are taken as 10 MPa and 30^oC respectively, unless specified other way. The heat loss from the gas cooler into the surroundings is neglected.

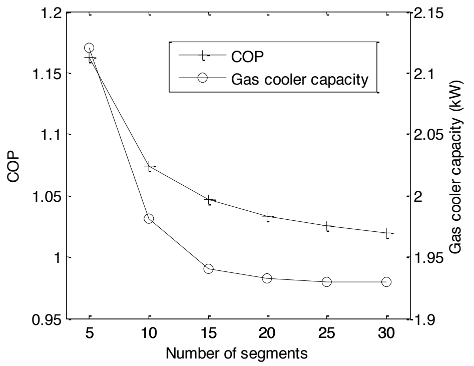


Fig 2. Grid independence test

3. Results and discussion

Adding NP's to the water leads to increase in thermal conductivity and enhances the heat transfer coefficient. However, at the same time pumping power also increases owing to increase in effective viscosity of NF. Equal Re and pumping power comparison criteria are chosen for assessing alumina NF as a coolant in gas cooler. The comparison of COP for refrigeration system employing water as cooling medium in gas cooler to that employing alumina NF is shown in Fig. 3. It can be observed from Fig. 3 that with increase in Re, pumping power and particle volume fraction for NF, the COP increases due to rise in heat transfer coefficient of coolant. At equal Re comparison criterion and in comparison with water, the increments in COP for refrigeration system employing 0.5%, 1.5% and 2.5% alumina NF as cooling medium are respectively equal to 3.35%, 8% and 13.24% at Re of 16000. This is attributed to the increased velocity of NF to maintain equal Re comparison criterion. While, comparison at equal pumping power yields 1.01% increment in COP for 0.5 % NF cooled system, while decrement of 1.03 % and 3.83 % for 1.5 % and 2.15 % volume fraction respectively. This can be ascribed to the reduced velocity of NF at equal pumping power resulting in negligible enhancement in heat transfer coefficient only at low volume fraction. However, irrespective of comparison basis, increase in the mass flow rate of NF is observed in this study. The results obtained are found consistent with those reported by Haghighi et al. [9].

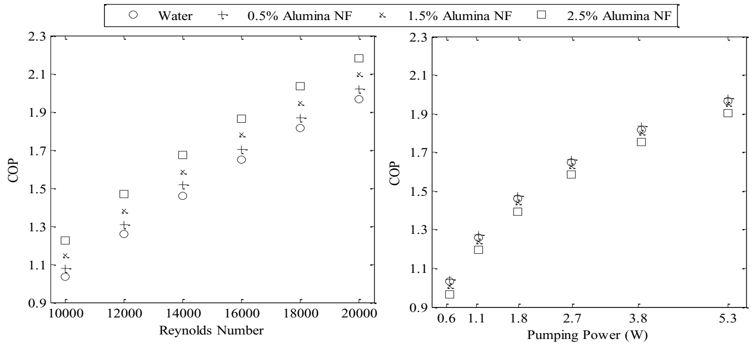


Fig.3 Comparison of variation of COP of water and Alumina NF with Reynolds Number and Pumping Power

Fig. 4 shows variation of overall conductance of gas cooler with varying Re and Pumping power. As conductance is a function of heat transfer coefficient of refrigerant and coolant side, it increases with increase in Re and Pumping power. At equal Re of 16000, NF cooled gas cooler with 0.5%, 1.5% and 2.5% volume fraction shows an increment in overall conductance by 4.71%, 8.01% and 11.63 % respectively. While for same pumping power comparison, 0.5% NF cooled gas cooler show an increment by 2.64% and gas cooler cooled with NF having 1.5% and 2.5% volume fraction shows decrement by 0.28% and 4% respectively.

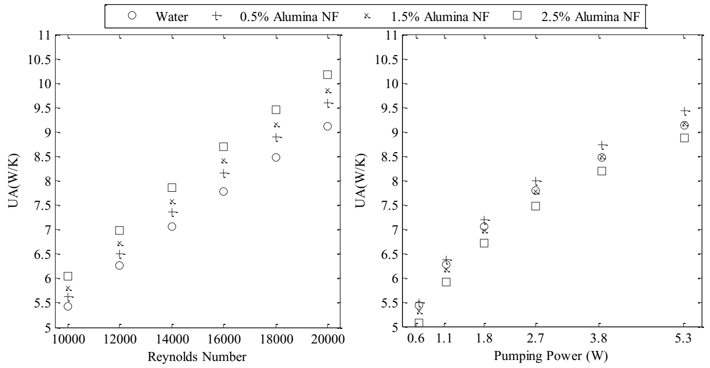


Fig.4 Comparison of variation of overall conductance of coolants with Reynolds Number and Pumping Power

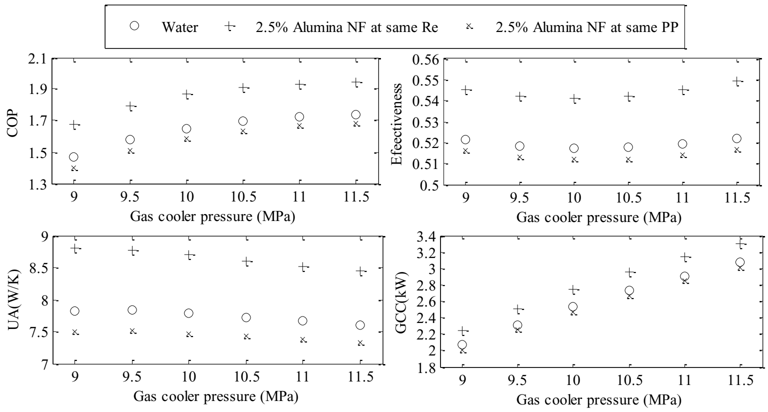


Fig.5 Comparison of variation of COP and gas cooler performance parameters with gas cooler pressure.

The gas cooler effectiveness and capacity shows similar trend as that of overall conductance. Hence, not shown here graphically. Fig. 5 shows variation of COP, overall conductance, effectiveness and gas cooler capacity with gas cooler pressure for both the comparison criteria. The COP is found to increase with increase gas cooler pressure and this is attributed to the required high gas cooler pressure for refrigerant outlet temperature of above 40^{o}C in all investigated cases. Due to fixed mass flow rate of coolant and refrigerant, effectiveness decreases when gas cooler pressure is increased from 9 to 10 MPa, while above 10 MPa effectiveness increases with increase in gas cooler pressure. As the gas cooler pressure moves away from vicinity of pseudo critical zone, overall conductance decreases. Gas cooler capacity is found to increase with increase in gas cooler pressure. As expected, at equal Re comparison criterion, NF cooled system performance is found superior while reverse is true at equal pumping power comparison base. Table 1 shows absolute values of operating parameters for NF (2.5 %) and water cooled system. From the Table 1, it can be observed that pump work is negligible in comparison to compressor work and gas cooler capacity. Owing to this, both coolants performance is also compared at equal mass flow rate (mass flow rate corresponding to Re of water) and it is clear from the Table 2 that water cooled system exhibits higher performance than NF cooled system also at equal mass flow rate comparison base. Further, the NF based system shows degradation in performance with increase in particle volume fraction and this implies that one can achieve better performance even by increasing the mass flow rate of base fluid rather than adding expensive nano sized particles.

Table 1 Comparison of performance of water and NF cooled refrigeration system performance at equal Re and pumping power.

Parameters	***Same Reynolds number***		***Same Pumping power***	
	Water	**Alumina NF**	**Water**	**Alumina NF**
Refrigerant outlet temperature (^{0}C)	45.79	44.47	45.79	46.14
Coolant outlet temperature (^{0}C)	43.65	41.6	43.65	42.91
Coolant mass flow rate (kg/s)	0.0428	0.0548	0.0428	0.043
Reynolds number	16000	16000	16000	13119
Pumping power (W)	2.694	4.896	2.694	2.694
Cooling COP	1.64	1.86	1.64	1.58
Overall conductance (UA) (W. K^{-1})	7.7866	8.6926	7.7866	7.4745
Gas cooler effectiveness	0.5173	0.5412	0.5173	0.5120
Gas cooler capacity (kW)	2.5317	2.7447	2.5317	2.4717

Table 2 Comparison of water and NF cooled refrigeration system performance at equal mass flow rate (water/0.5 % NF/1.5% NF/2.5% NF)

Mass flow rate (kg · s-^{1})	COP	Effectiveness	Overall conductance (W · K^{-1})	Gas cooler capacity (kW)
0.0257	1.033/1.032/0.986/0.929	0.449/0.448/0.440/0.430	5.46/5.45/5.22/4.95	1.95/1.94/1.89/1.84
0.0313	1.260/1.262/1.214/1.156	0.0477/0.478/0.470/0.461	6.32/6.33/6.07/5.77	2.15/2.16/2.11/2.06
0.0372	1.459/1.468/1.419/1.359	0.501/0.503/0.495/0.486	7.14/7.16/6.88/6.56	2.31/2.35/2.31/2.25
0.0428	1.646/1.653/1.604/1.542	0.520/0.523/0.516/0.507	7.91/7.94/7.65/7.30	2.49/2.53/2.49/2.43
0.0487	1.816/1.822/1.770/1.709	0.532/0.540/0.533/0.525	8.62/8.68/8.37/8.01	2.632.70/2.65/2.59
0.0547	1.963/1.971/1.924/1.861	0.545/0.555/0.548/0.540	9.23/9.37/9.05/8.68	2.74/2.85/2.80/2.74

4. Conclusion

In this study, performance of alumina NF cooled double pipe gas cooler in trans-critical CO_2 refrigeration cycle is theoretically compared to that of water cooled gas cooler. A more practical equal pumping power comparison criterion is adopted besides conventional equal Re comparison base. NFs are loaded with 0.5%, 1.5% and 2.5% of particle volume fraction under turbulent flow conditions. Both refrigerant side and coolant side modelling is made with extra care of CO_2 thermo-physical property variations in vicinity of pseudo critical temperature zone. The prominent conclusions drawn are as follows:

- COP of the NF cooled refrigeration system is found higher only at equal Re comparison criterion while for equal pumping power the water cooled system dominates. Gas cooler overall conductance, effectiveness and capacity shows similar trend under both comparison criteria. As expected, gas cooler performance is improved only under same Re comparison base.
- With the increase in gas cooler pressure, COP is found to increase irrespective of comparison criteria due to imposed high gas cooler outlet temperature. Gas cooler effectiveness shows a distinct behavior with respect to increase in gas cooler pressure: first it decreases till 10 MPa and then it increases. This is due to the imposed mass flow rates of both refrigerant and coolant. Overall conductance of gas cooler decreases with increase in

gas cooler pressure, while reverse is true for gas cooler capacity. Performance of NF cooled gas cooler with respect to gas cooler pressure is found better only at equal Re comparison base.

- The pump work for coolant is found negligible as compared to compressor work and gas cooler capacity. Even at equal mass flow rate comparison base, NF cooled gas cooler shows no potential improvement in performance. This implies that by increasing only mass flow rate of water, the performance of water cooled system can be improved as much as that of NF cooled systems. The results obtained are consistent with those reported by Haghighi et al. [9], however still requires further testing either by employing more accurate two phase modelling approach or by extensive experimentation involving environmental and economic analysis.

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