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Design and comparison of direct and indirect cooling system for 25 MW solar power plant operated with supercritical CO_2 cycle

M. Monjurul Ehsan[a,*], Zhiqiang Guan[a], A.Y. Klimenko[a], Xurong Wang[b]

[a] The Queensland Geothermal Energy Centre of Excellence (QGECE), School of Mechanical and Mining Engineering, The University of Queensland, Brisbane, QLD 4072, Australia

[b] School of Energy and Power Engineering, Xi'an Jiaotong University, Xi'an, Shanxi 710049, People's Republic of China

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ABSTRACT

A validated MATLAB code is developed to design a dry cooling unit for a 25 MW solar power plant operated with supercritical CO_2 (sCO_2) Brayton cycle. Both direct and indirect cooling systems are designed for the power plant and their performance are compared. For direct cooling system, air cooled heat exchanger unit is designed whereas for indirect cooling system, optimum size of shell and tube heat exchanger is selected addressing the retainment of shell side heat transfer correction factor within a reasonable limit. The nonlinear property variation of sCO_2 near the critical condition is reasonably well captured by the present code in both shell and tube and air cooled heat exchanger units. The comparative study of direct and indirect cooling system is performed for inlet sCO_2 temperature of 71 °C, operating pressure of 7.5 MPa and ambient air temperature from 15 °C to 40 °C. For the same heat rejection, indirect cooling system requires a much higher cooling tower. During high ambient temperature period, direct dry cooling system shows superior cooling performance in terms of lower sCO_2 outlet temperature compared to indirect cooling system.

1. Introduction

Supercritical CO_2 due to its unique transport properties, has been discovered as one of the efficient and environmentally benign refrigerant substitutes in refrigeration and air conditioning technologies. The newly applications of sCO_2 have been proposed in thermal power plants with closed loop Brayton cycles. The sCO_2 power cycles are believed to be the next generation power cycles due to the lower cost, higher thermal efficiency and simplified plant design in comparison with traditional superheated Rankine power cycles [1]. One of the key challenges associated with sCO_2 power cycles is the cooling of sCO_2 near the critical temperature and pressure. A reliable and well-designed cooling system is required for efficient and smooth operation of thermal power plant. In arid areas, natural draft dry cooling towers (NDDCT) are preferred over the wet coolers due to the insufficient water supply, environmental concern and the high maintenance and operation cost related to wet cooling.

The first ever closed loop Brayton cycle was proposed by Feher [2] in 1967, where the cycle was modified by the introduction of a pump to handle the high density of sCO_2 during the compression process instead of using a compressor. Dostal et al. [1] performed comparative study among various sCO_2 Brayton cycles and higher thermal efficiency of 47% was observed with recompression Brayton cycle. Besarati and Goswami [3] conducted the thermal assessment of four different sCO_2 Brayton cycle configurations among which recompression with partial cooling showed highest cycle thermal efficiency of 50%. Conboy et al. [4] performed small scale experimentation with recuperated sCO_2 Brayton cycle to investigate the influence of operating conditions on the cycle performance. Al-Sulaiman and Atif [5] investigated the performance of different sCO_2 Brayton cycle layouts integrated with solar power tower. The recompression Brayton cycle showed promising performance with a thermal efficiency of 52%. Dyreby et al. [6] performed modelling study, where dry cooling unit is employed in a recompression sCO_2 cycles for concentrated solar power (CSP) application. The cycle overall efficiency of 43% with compressor inlet temperature of 50 °C was achieved with sCO_2 Brayton cycle using air cooled heat exchanger unit [7]. Wang and He [8] optimized the solar power tower with molten salt as heat transfer fluid in a sCO_2 recompression Brayton cycle with reheating. Zhang et al. [9] worked on the design and optimization of the heat source on the performance of sCO_2 Rankine cycle by exploring the influence of thermal boundary condition. Milani et al. [10] performed optimization studies in terms of thermal efficiency, water demand and solar power compatibility among various sCO_2 Brayton cycle layouts integrated with concentrated solar

* Corresponding author.
E-mail address: m.ehsan@uqconnect.edu.au (M. Monjurul Ehsan).

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Nomenclature

A	area, m^2
C_p	specific heat, J/kg K
C_{Dts}	drag coefficient
d	diameter, m
f	friction factor
F_{Dts}	drag force, N
G	mass velocity, kg/s m^2
H	height, m
h	heat transfer coefficient, W/m^2 K
J	shell side heat transfer correction factor
k	thermal conductivity, W/m K
K	loss coefficient
l_{ts}	length of tower support, m
L_t	length of finned tube, m
m	mass flow rate, kg/s
n_{ts}	number of tower supports
n_r	number of tube rows
n_{tr}	effective number of tubes per row
P_f	fin pitch, m
P_t	transversal tube pitch, m
P_l	longitudinal tube pitch, m
P_d	diagonal tube Pitch, m
ΔP	pressure drop, N/m^2
Q	heat transfer rate, W
R	shell side pressure drop correction factor
T	temperature, °C or K
t_f	fin thickness (mean), m
t_{ft}	fin tip thickness, m

Greek symbols

ε	surface roughness, m
η	efficiency
μ	dynamic viscosity, kg/ms
ρ	density, kg/m^3
σ	area ratio

Dimensionless number

Fr_D	Froude number
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
R_y	characteristic Reynold number
J	Colburn factor

Subscripts

a	air side
ct	cooling tower
ctc	cooling tower contraction
cte	cooling tower expansion
cv	control volume
he	heat exchanger
i	inner; inlet
o	outer; outlet
lm	log mean temperature difference
pc	pseudocritical
RB	Robinson and Brigg correlation
r	root
s	supercritical; shell side
t	tube side
to	tower outlet
ts	tower support
w	wall temperature

power systems. Song et al. [11] performed the thermal assessment of sCO_2 Brayton cycle by enhancing the preheating process. Padilla et al. [12] employed dry cooling system to assess the thermal efficiency with various sCO_2 Brayton cycles. Recompression cycle with intercooling achieved the highest cycle efficiency of 55.2% with turbine inlet temperature of 850 °C.

Experiments performed to assess the thermal performance of a small natural draft dry cooling tower coupled with 1 MW concentrating solar thermal sCO_2 power cycle [13]. Numerical computation conducted to investigate the cooling performance of a NDDCT under cross wind condition and factors influencing the deterioration of the cooling performance were also reported [14]. Numerical analysis with CFD model used to evaluate the cooling performance of NDDCT by optimizing the water distribution in cross wind condition [15]. Several recent studies were performed to investigate the thermal performance of NDDCT under cross-wind condition [16–20]. Sadafi et al. [21] performed experimentation to precool the inlet air in a NDDCT. Yubiao Sun et al. [22] also conducted experimentation to enhance the cooling performance by spraying water droplets at the inlet of the tower. The influences of nozzle arrangement, injection height and injection direction on the cooling performance were also carried out [23]. Trabelsi et al. [24] analysed the performance of dry cooled solar power tower with molten salt as a working fluid. The overall system was optimized with various solar filed size, size of turbine and type of heat transfer fluid.

So far researchers have explored the potential of sCO_2 as a working fluid in various closed loop Brayton cycle configurations. Few studies have investigated the cycle performance coupled with dry cooling units. Dai et al. [25] recently performed the comparison of direct and indirect dry cooling system operating with sCO_2 Brayton cycle under various ambient temperatures. The study was performed with an existing NDDCT of 20 m height and the design of air cooled heat exchanger unit was not taken into consideration. In the present work, the design and comparison of direct and indirect dry cooling system in a natural draft dry cooling unit are performed. One dimensional MATLAB code is generated to design the natural draft dry cooling tower for both direct and indirect cooling systems. The novelty of the present work includes the detailed analysis of an air cooled heat exchanger unit and shell and tube heat exchanger operated with sCO_2. The design of heat exchangers is a great challenge since the working fluid is sCO_2 and conventional method is not sufficient to predict the heat transfer accurately. The present code is able to predict the exact heat transfer mechanism and pressure drop characteristics of sCO_2 working with shell and tube and air cooled heat exchanger unit.

2. Dry cooling system classification

In dry cooling system, fresh ambient air is used to cool the turbine exhaust and the cooling medium never gets in contact with the working fluid. Fig. 1. shows a direct dry cooling system, where sCO_2 after following expansion process through the turbine is sent to the recuperator to preheat the sCO_2 stream from the compressor outlet. The sCO_2 is cooled by exchanging heat to the airstream flowing across the bundles of finned tube heat exchanger unit in a NDDCT. However, the cooling potential of a direct system depends on the dry air bulb temperature.

In indirect dry cooling system shown in Fig. 2, the turbine exhaust after flowing through the recuperator is sent to the pre-cooler, where the heat from the sCO_2 is transferred to the cooling water by shell and