Energy xxx (2016) 1–13

Contents lists available at ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

Supercritical CO_2 Rankine cycles for waste heat recovery from gas turbine

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ARTICLE INFO

Article history:
Received 13 March 2016
Received in revised form
17 September 2016
Accepted 26 October 2016
Available online xxx

Keywords:
Supercritical CO_2
Rankine cycle
Waste heat recovery
Gas turbine
Cascade cycle
Split cycle

ABSTRACT

A supercritical carbon dioxide (S-CO_2) Rankine cycle for waste heat recovery (WHR) from a gas turbine can achieve high efficiency despite its simplicity and compactness in comparison to a steam/water cycle. With respect to WHR, it is very important to maximize the net output power by incorporating the utilization efficiency of the waste heat in conjunction with the cycle thermal efficiency. A simple S-CO_2 Rankine cycle used for a high-temperature source cannot fully utilize the waste heat because the working fluid is preheated by the recuperator to a high temperature to achieve a high cycle efficiency. To recover the remaining waste heat from a simple cycle, a cascade cycle with a low-temperature (LT) loop can be added to the high-temperature (HT) loop, or a split cycle—in which the flow after the pump is split and preheated by the recuperator and LT heater separately, before the HT heater can be used. This study presents a comparison of three cycles in terms of energy and exergy analyses of their systems. The results show that a split cycle can produce the highest power of the threes systems considered over a wide range of operating conditions. The reasons for this are explained in detail.

1. Introduction

A supercritical CO_2 (S-CO_2) power cycle has several advantages over a steam/water cycle, including simplicity, compactness, sustainability, and superior economy [1–5]. Given these advantages, interest in the use of S-CO_2 power cycles has increased for many applications, such as in fossil fuel, renewable energy, waste heat, and advanced nuclear power plants.

S-CO_2 power cycles can involve high-temperature (HT) heat sources such as nuclear power, concentrated solar power, and combustion. In S-CO_2 power cycles with high-temperature heat sources, a recuperator is needed to increase the turbine inlet temperature to a level sufficient to achieve high efficiency. However, heat transfer occurs from the turbine exhaust stream (which has a low specific heat) to the pump exit stream (which has a high specific heat), leading to large internal irreversibility in the recuperator [1]. In a recompression cycle, the recuperator is divided into low- and high-temperature parts. Each part has different flow rates to accommodate the large variations in the heat capacity of the fluid. Hence, the recompression cycle is considered to be a highly efficient cycle. If there is an additional low-temperature (LT) heat source, it can be used to compensate for the low specific heat of the turbine exhaust stream to minimize the internal irreversibility in the recuperator [6].

In using S-CO_2 power cycles for waste heat recovery, it is very important to maximize the net output power by incorporating the utilization efficiency of the waste heat along with the thermal efficiency of the cycle [7]. Therefore, the optimal system configuration for achieving the maximum power from the waste heat source is different from that for an HT heat source. Mahmood [8] compared the performance of two configurations of S-CO_2 Brayton cycles (i.e., single-recuperated and recompression cycles) for waste heat recovery of low-temperature waste gas (i.e., at temperatures less than 700–800 K), suggesting that the recompression cycle, despite its increased complexity, does not yield any appreciable benefit in terms of net power output.

Echogen Power Systems presented many of the advantages of an S-CO_2 power cycle for waste heat recovery (WHR) from a gas turbine, including the fact that it can achieve high efficiency despite its

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http://dx.doi.org/10.1016/j.energy.2016.10.106
0360-5442/

Please cite this article in press as: Kim YM, et al., Supercritical CO_2 Rankine cycles for waste heat recovery from gas turbine, Energy (2016), http://dx.doi.org/10.1016/j.energy.2016.10.106

Nomenclature

c_p	isobaric specific heat [kJ/(kgK)]
E	exergy (kJ)
$\dot{E}$	rate of exergy (kW)
ε	heat exchanger effectiveness
h	specific enthalpy (kJ/kg)
HT	high temperature
k	specific exergy (kJ/kg)
L	exergy loss (kJ)
$\dot{L}$	rate of exergy loss (kW)
LT	low temperature
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (kPa)
Q	heat (kJ)
$\dot{Q}$	rate of heat (kW)
s	specific entropy [kJ/(kgK)]
S-CO_2	supercritical CO_2
T	temperature (K)
W	work (kJ)
$\dot{W}$	rate of work (kW)
x	split ratio
η	isentropic efficiency
η	efficiency
η_{II}	second-law efficiency

Subscripts

0	atmospheric (environmental) state
C	condenser
cyc	cycle
e	expander
EG	exhaust gas
P	pump
CO_2	carbon dioxide
H	heater
HR	heat recovery
i	state point
in	inlet
max	maximum
net	net output
out	outlet
P	pump
R	recuperator
s	isentropic
sys	system
T	turbine

Superscripts

$+$	input
$-$	output

simplicity and compactness in comparison to a steam/water cycle [9,10]. Wright presented a bulk cold energy storage system run using a CO_2 refrigeration cycle and a power plant peaking concept, coupled with a S-CO_2 waste heat recovery system for a gas turbine [11]. However, a single-recuperated S-CO_2 Rankine cycle used for a high-temperature heat source cannot fully utilize the available waste heat. The reason for this is that the recuperator preheats the working fluid to a high temperature in order to maintain the high thermal efficiency of the cycle. Various cycle layouts have been proposed for use in recovering the remaining waste heat from a simple cycle. Cho et al. [12] compared the performance of seven cycle layouts of S-CO_2 cycles (including intercooling, cascading, and split concept cycles) as a bottoming power system with that of a steam Rankine cycle in a natural gas combined-cycle power plant.

Huck [13] compared the performance of the most efficient configuration for an S-CO_2 cycle with dual-expansion and dual-flow split versus steam-bottoming cycles for gas turbine combined-cycle applications. In the case of WHR for a heavy-duty gas turbine, it is difficult for the performance of an optimized S-CO_2 cycle to exceed that of a three-pressure reheat steam-bottoming cycle. However, when compared with a two-pressure non-reheat steam cycle that recovers heat from small industrial and aeroderivative gas turbines, an S-CO_2 cycle can outperform a steam-bottoming cycle [13]. In other research, the use of one or more thermoelectric generator (TEG) systems integrated with an S-CO_2 cycle were proposed to increase the power recovery from an MT30 gas turbine used in marine applications [14,15]. The use of TEG systems takes advantage of the temperature differences between the cycle components.

Wright et al. [16] performed thermo-economic analyses of four S-CO_2 WHR systems (a single-recuperated Brayton cycle, a cascade cycle, a dual-recuperated cycle, and a split cycle, the last of which is referred to as a preheating cycle in Ref. [16]) for a 25-MWe gas turbine. The cascade and dual-recuperated cycles were a combination of two S-CO_2 cycles that recover the remaining heat from the single S-CO_2 cycle, with one compressor used in common but with the flow after the compressor (pump) divided into two (HT, LT) streams and passed through each heating loop and turbine. In the cascade cycle, the HT stream passed directly through a primary heater in order to recover the waste heat from the gas turbine without passing through a recuperator. On the other hand, the LT stream passed through an LT recuperator and then an HT recuperator, in order to recover the residual heat of the expanded HT stream. A split cycle can be used in other ways, with the flow split after the compressor (pump) being preheated by the recuperator and LT heater separately, and then merged and passed through the same HT heater and turbine. Although Wright et al. [16] showed that the split cycle yielded the highest net electric power from the waste heat source of the four cycles considered, the optimization process was not given, and the reason for the highest efficiency of the split cycle in comparison with the other cycles was not explained.

This study presents the optimization processes for S-CO_2 Rankine cycles that are used to maximize the power obtained from the waste heat of a small industrial gas turbine. This study also compares three types of cycles (simple, cascade, and split) over a wide range of upper pressures in the cycles, and explains the cause of the efficiency loss in each cycle based on energy and exergy analyses of the systems.

The cascade cycle presented in this study is different and more advanced than the one presented by Wright et al. [16]. The differences in this cascade cycle allow for an increase the net power of the system. In the case of the S-CO_2 WHR system, it is very important to minimize the temperature difference for the heat transfer (exergy loss) between the heat source and the high pressure of CO_2 in the heater, while lowering the temperature of the exhaust gas through the heater as much as possible. However, in the cascade cycle presented by Wright et al. [16], it is very difficult to minimize the temperature difference for the heat transfer in the

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