

## Research Article

# Thermodynamic Analysis of Supercritical CO<sub>2</sub> Power Cycle with Fluidized Bed Coal Combustion

Chenchen Geng, Yingjuan Shao , Wenqi Zhong, and Xuejiao Liu

Key Laboratory of Energy Thermal Conversion and Control of Ministry of Education, School of Energy and Environment, Southeast University, Nanjing 210096, China

Correspondence should be addressed to Yingjuan Shao; yjshao@seu.edu.cn

Received 4 May 2018; Accepted 25 June 2018; Published 24 July 2018

Academic Editor: Xiaoke Ku

Copyright © 2018 Chenchen Geng et al. This is an open access article distributed under the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

Closed supercritical carbon dioxide (S-CO<sub>2</sub>) Brayton cycle is a promising alternative to steam Rankine cycle due to higher cycle efficiency at equivalent turbine inlet conditions, which has been explored to apply to nuclear, solar power, waste heat recovery, and coal-fired power plant. This study establishes 300MW S-CO<sub>2</sub> power system based on modified recompression Brayton cycle integrated with coal-fired circulating fluidized bed (CFB) boiler. The influences of two stages split flow on system performance have been investigated in detail. In addition, thermodynamic analysis of critical operating parameters has been carried out, including terminal temperature difference, turbine inlet pressure/temperature, reheat stages, and parameters as well as compressor inlet pressure/temperature. The results show that rational distribution of split ratio to the recompressor (SR<sub>1</sub>) achieves maximal cycle efficiency where heat capacities of both sides in the low temperature recuperator (LTR) realize an excellent matching. The optimal SR<sub>1</sub> decreases in the approximately linear proportion to high pressure turbine (HPT) inlet pressure due to gradually narrowing specific heat differences in the LTR. Secondary split ratio to the economizer of CFB boiler (SR<sub>2</sub>) can recover moderate flue gas heat caused by narrow temperature range and improve boiler efficiency. Smaller terminal temperature difference corresponds to higher efficiency and brings about larger cost and pressure drops of the recuperators, which probably decrease efficiency conversely. Single reheat improves cycle efficiency by 1.5% under the condition of 600°C/600°C/25Mpa while efficiency improvement for double reheat is less obvious compared to steam Rankine cycle largely due to much lower pressure ratio. Reheat pressure and main compressor (MC) inlet pressure have corresponding optimal values. HPT and low pressure turbine (LPT) inlet temperature both have positive influences on system performance.

## 1. Introduction

High efficiency and clean coal-fired power generation technologies are priority research directions in the traditional energy industry. Utilizing double reheat and raising steam parameters are two effective means to improve power plant efficiency, however, which is generally restricted by nickel-based alloys development and expensive equipment cost. More efficient and compact cycles integrated with conventional coal-fired boiler become a breakthrough in improving fossil fuels utilization rate. Closed supercritical carbon dioxide (S-CO<sub>2</sub>) Brayton cycle is considered as a promising substitute for steam Rankine cycle applied to thermal power generation. S-CO<sub>2</sub> is that both pressure and temperature of CO<sub>2</sub> are above its critical point (7.38Mpa 30.98°C) which is a suitable working fluid for power generation because of its near-critical

properties. CO<sub>2</sub> critical temperature is easily achieved and operating at high pressure allows more compact turbomachinery. Furthermore, S-CO<sub>2</sub> possesses high density in near-critical region similarity to its liquid phase resulting in less compressor work and higher efficiency. Besides, CO<sub>2</sub> is generally abundant, less corrosive, inexpensive, and nontoxic [1].

Previous researches show that S-CO<sub>2</sub> Brayton cycle is more efficient than supercritical steam Rankine cycle above moderate turbine inlet temperatures and can be coupled with various heat sources. The investigations of its potential applications have extended from nuclear, concentrated solar power (CSP), shipboard propulsion, waste heat recovery, and geothermal to fossil fuel [2, 3]. Detailed analysis began in the late 1960s. Feher [4] proposed CO<sub>2</sub> as the working fluid for the regenerative supercritical cycle that operated entirely above the critical pressure and was compressed in

the liquid phase. Hoffmann and Feher [5] further designed a 150 kWe power conversion module for a helium-cooled nuclear reactor in 1970. Angelino [6] analyzed different configurations of CO<sub>2</sub> condensing cycles including recompression cycle, partial cooling cycle, pre-expansion cycle, and pre-compression cycle. The study pointed out that the efficiency of recompression CO<sub>2</sub> cycle was better than reheat steam cycle at turbine inlet temperatures higher than 650°C. These studies have not been given sufficient attention in the following decades because of immature manufacturing technics as well as limitation of heat resource temperature. Dostal et al. [1] revived interest in S-CO<sub>2</sub> cycles in the early 20th century; they modified the recompression S-CO<sub>2</sub> cycle proposed by Angelino and carried out detailed component design, optimization, and economic analysis as well as control schemes for advanced nuclear reactors. After that, a growing number of studies were carried out involving system design [7], thermodynamic analysis and optimization [8, 9], and experimental loop tests [10, 11] as well as commercial-scale construction [12, 13] for different applications.

A considerable amount of heat recovered by recuperators from turbine exhaust brings about a narrow temperature range through the heat source. S-CO<sub>2</sub> cycle can well match with nuclear and CSP heat sources due to their nearly constant temperature [14]. However, modifications are indispensable for applications to the coal-fired power plant in order to effectively utilize flue gas heat and improve the gross plant efficiency. Several improvements have been put forward, such as utilizing residual heat to preheat air, improving cycle layouts, extracting a fraction of S-CO<sub>2</sub> from different points at high-pressure side to recover flue gas heat, heating recycled flue gas, and adopting cascaded or combined cycles. Muto et al. [15] proposed a double expansion turbine cycle to reduce the pressure difference between gas side and S-CO<sub>2</sub> side of heater and introduced an economizer to recover exit gas heat. Moullec [16] explored the potential performance of a coal-fired power plant based on double reheat recompression S-CO<sub>2</sub> cycle integrated with a post-combustion carbon capture and storage (CCS). A high temperature air preheating combined with a fraction of fluid split from the main compressor (MC) to the economizer can effectively utilize 100~500°C flue gas heat. The net lower heating value (LHV) plant efficiency without CO<sub>2</sub> capture for a maximal temperature and pressure of 620°C and 30Mpa could achieve 50.3% and 4.8% higher than supercritical steam power plant. Mecheri and Moullec [17] investigated the S-CO<sub>2</sub> system performance and still recommended the previous layout. Li et al. [18] set up an additional split flow economizer before the air preheater to heat the CO<sub>2</sub> from low temperature recuperator (LTR) and had accomplished 5MWe fossil-based S-CO<sub>2</sub> recompression and reheat integral test facility design. Bai et al. [19] provided a novel S-CO<sub>2</sub> intercooled recompression cycle with additional medium-temperature recuperator and an anabranch applied to a coal-fired power plant, which could achieve 49.5% net LHV efficiency with 29.6Mpa/650°C. Shelton et al. [20] constructed S-CO<sub>2</sub> recompression cycle with an oxy-coal circulating fluidized bed (CFB) and CCS, which could realize a nearly constant temperature heat source through heating the recycled flue gas and a fraction of CO<sub>2</sub> from the MC exit.

The efficiency increment of the S-CO<sub>2</sub> cycle plant configured with reheat and intercooling could achieve 2% compared to steam Rankine cycle power. Pratt and Whitney Rocketdyne [21] developed Zero Emission Power and Steam (ZEPS) plant which adopted a pressurized fluidized bed combustor (PFBC) with combined cycle that modified S-CO<sub>2</sub> cycle was used as the topping cycle and steam cycle was used as the bottoming cycle. The ZEPS plant efficiency could increase by 9.4% compared with oxy-combustion atmospheric boiler. Thimsen et al. [22] designed an inverted tower coal-fired S-CO<sub>2</sub> heater that applied to a 750MWe recompression S-CO<sub>2</sub> power cycle with 700°C turbine inlet temperature. A second cascading S-CO<sub>2</sub> cycle was employed as the bottoming cycle to cool the combustion products from 530°C to 370°C.

CFB boiler with relatively uniform, low heat flux and combustion temperature, and wide fuel flexibility as well as low pollutant emissions can be better coupled with S-CO<sub>2</sub> cycle compared to pulverized coal (PC) boiler [14, 23]. Combination of CFB boiler and S-CO<sub>2</sub> power cycle is one of the effective means for the coal-dominated energy structure in China to improve energy conversion efficiency and reduce pollutant emissions at their source. The gross system performance requires further analysis and optimization. This study establishes 300MW coal-fired power system based on recompression S-CO<sub>2</sub> cycle and single reheat integrated with CFB boiler. Secondary split flow extracted from high-pressure side of the cycle is adopted to accommodate narrow temperature variation through the boiler caused by the necessarily high extent of recuperated heat. A series of parameters sensitivity analysis have been carried out, including two stages split ratio, terminal temperature difference, turbine inlet pressure/temperature, and reheat pressure/temperature as well as compressor inlet pressure/temperature.

## 2. System Configuration and Model

*2.1. System Configuration.* The overall system layout based on modified recompression S-CO<sub>2</sub> cycle and single reheat integrated with CFB boiler and the corresponding T-s diagram are shown in Figures 1 and 2. Several different configurations have been put forward, including simple recuperated cycle, recompression cycle, partial cooling cycle, and pre-expansion cycle. However, recompression cycle is considered as a promising alternative with high performance and simple compositions [1]. The recompression cycle effectively alleviates pinch-point problem through adjusting flow rate to compensate specific heat difference and better match temperature variation between cold and hot side fluid in the LTR. Besides, single reheat, a common method used in steam Rankine cycle, can further increase plant efficiency. An additional high pressure CO<sub>2</sub> split flow extracted from LTR outlet to the economizer (ECO) can not only utilize flue gas heat but also increase high temperature recuperator (HTR) effectiveness, which is a valid method employed by aforementioned studies.

As Figure 1 shows, there exist two split flow locations in the system. Firstly, a portion of S-CO<sub>2</sub> exiting from LTR hot side is cooled in the precooler (PC), compressed by the MC and preheated in the LTR sequentially. The remaining flow

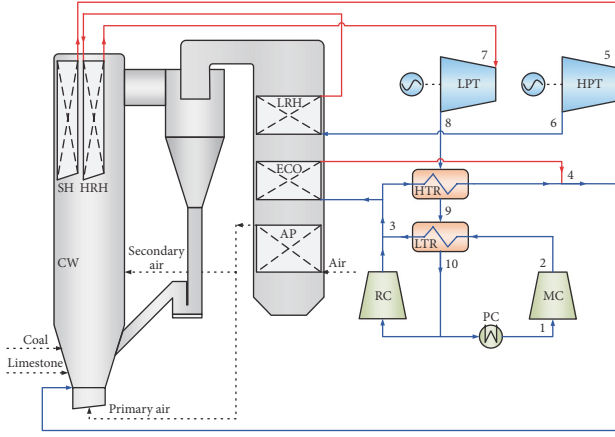


FIGURE 1: Schematic diagram of recompression S-CO<sub>2</sub> cycle with single reheat.

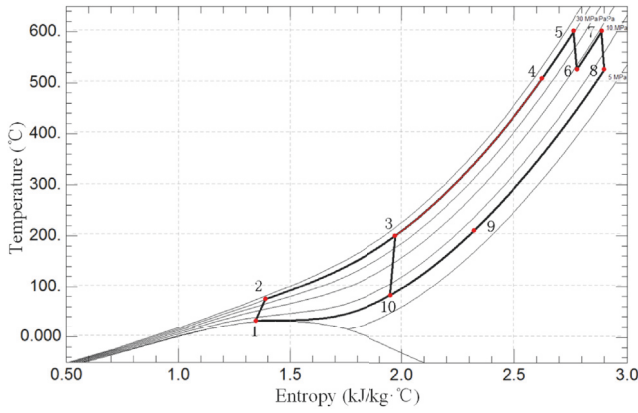


FIGURE 2: T-s diagram of recompression S-CO<sub>2</sub> Brayton cycle with single reheat.

is directly compressed by the recompressor (RC) and mixes with outlet fluid of LTR cold side. The mixture is divided into two branches again. One branch of S-CO<sub>2</sub> exiting from LTR cold side is diverted into the ECO to recover relatively high temperature flue gas heat and another branch is further preheated in the HTR. Two branches of fluid finally converge and are further heated to the desired temperature through the cooling wall (CW) and superheater (SH) of the CFB boiler sequentially. S-CO<sub>2</sub> with high pressure and temperature expands through the high pressure turbine (HPT) and is reheated in the low temperature reheater (LRH) and the high temperature reheater (HRH). S-CO<sub>2</sub> with moderate pressure expands through the low pressure turbine (LPT) and then passes through the HTR, LTR, and PC to the lowest temperature and pressure.

**2.2. Mathematical Model.** The thermodynamic model for the S-CO<sub>2</sub> power system has been established based on mass and energy balance. S-CO<sub>2</sub> properties are inquired from REFPROP database, which is developed by the National Institute of Standards and Technology (NIST) [24] based on Span and Wagner equation of state [25]. Thermal parameters

of different states in the system are calculated through iteration. In addition, CFB boiler heat balance and efficiency are obtained by preliminary thermal calculation [26].

Several assumptions need to be considered in this study:

- (1) The system operates under steady-state conditions.
- (2) Heat losses in pipelines and each component are neglected.
- (3) The temperature and pressure of mixing flows as well as respective branch flows are set to the identical value in order to reduce irreversible loss. The outlet states of LTR cold side and RC are equal. Similarly, the outlet states of HTR cold side and ECO are same.
- (4) Pressure drops in each component are assumed to be constant with the minimum and maximum pressure variations. The CFB boiler and PC pressure drops are set to 0.7Mpa and 0.05Mpa, respectively. The cold side and hot side pressure drops of LTR and HTR are set to 0.15Mpa and 0.1Mpa separately.

Two concepts of the split ratio are defined. The first split ratio ( $SR_1$ ) is defined as the ratio of RC mass flow to the total mass flow and the second split ratio ( $SR_2$ ) is defined as the ratio of ECO mass flow to the total mass flow, both of which are given by

$$SR_1 = 1 - \frac{(h_9 - h_{10})}{(h_3 - h_2)} \quad (1)$$

$$SR_2 = 1 - \frac{(h_8 - h_9)}{(h_4 - h_3)} \quad (2)$$

The compression and expansion in the MC, RC, HPT, and LPT are non-isentropic process. Work consumption of MC and RC and work output of HPT and LPT are expressed as

$$\begin{aligned} W_{MC} &= m(1 - SR_1) \cdot \frac{(h_{2s} - h_1)}{\eta_{MC}} \\ &= m(1 - SR_1) \cdot (h_2 - h_1) \end{aligned} \quad (3)$$

$$W_{RC} = \frac{m \cdot SR_1 (h_{3s} - h_{10})}{\eta_{RC}} = m \cdot SR_1 (h_3 - h_{10}) \quad (4)$$

$$W_{HPT} = m \cdot (h_5 - h_{6s}) \cdot \eta_{HPT} = m \cdot (h_5 - h_6) \quad (5)$$

$$W_{LPT} = m \cdot (h_7 - h_{8s}) \cdot \eta_{LPT} = m \cdot (h_7 - h_8) \quad (6)$$

The temperature difference and effectiveness are frequently used for recuperator model. However, the terminal temperature difference, defined as the achievable minimum temperature at the end of recuperators, is a preference for calculation [27]. The terminal temperature differences of LTR and HTR are set to 8°C, which is a reasonably accomplishable value for current technical equipment. The smaller terminal temperature difference can improve heat transfer capacity, but cause the nonlinear increase of recuperator volumes and capital costs. The temperature relations in the LTR and HTR are defined as

$$t_2 + \Delta t \leq t_{10} \leq t_9 \quad (7)$$

$$t_3 + \Delta t \leq t_9 \leq t_8 \quad (8)$$

The energy balance equations for the LTR and HTR are expressed as

$$Q_{LTR} = m(1 - SR_1) \cdot (h_3 - h_2) = m \cdot (h_9 - h_{10}) \quad (9)$$

$$Q_{HTR} = m(1 - SR_2) \cdot (h_4 - h_3) = m \cdot (h_8 - h_9) \quad (10)$$

Heat absorption in the CFB is expressed as

$$Q_{CFB} = m [SR_2 \cdot (h_5 - h_3) + (1 - SR_2) \cdot (h_5 - h_4) + (h_7 - h_6)] \quad (11)$$

Heat rejection in the PC is expressed as

$$Q_{PC} = m(1 - SR_1) \cdot (h_{10} - h_1) \quad (12)$$

Thermal efficiency and net efficiency are defined as

$$\eta_{th} = \frac{(W_{LPT} + W_{HPT} - W_{MC} - W_{RC})}{Q_{CFB}} \cdot 100\% \quad (13)$$

$$\eta_{net} = \frac{(W_{LPT} + W_{HPT} - W_{MC} - W_{RC}) \cdot \eta_G \cdot 100\%}{(m_{coal} \cdot Q_{net,ar})} \quad (14)$$

$$= \eta_{th} \cdot \eta_G \cdot \eta_{CFB}$$

Initial parameters of recompression S-CO<sub>2</sub> cycle with CFB, specific compositions of coal, and desulfurizer are listed in Tables 1 and 2.

**2.3. Validation of Thermodynamic Model.** The thermodynamic model is validated by literature data [9]. MC inlet pressure of 7.38Mpa and compressor isentropic efficiency of 89% are selected. Other input parameters are identical to Table 1. As Figure 3 shows, the relative errors are below 3%, which are primarily caused by recuperator model. Terminal temperature difference of 8°C corresponds to nearly 95% effectiveness of LTR; however effectiveness of HTR is higher than 95%, leading to larger thermal efficiency.

### 3. Results and Discussion

The influences of critical operating parameters on system performance have been investigated comprehensively based on cycle thermodynamics model and CFB thermal balance.

**3.1. Split Ratio to the RC (SR<sub>1</sub>).** Figure 4 presents effects of SR<sub>1</sub> on cycle efficiency at different terminal temperature differences. SR<sub>1</sub> is a key manipulated variable influencing the thermal efficiency of the S-CO<sub>2</sub> cycle, which has optimal values under given working conditions. Specific heat of S-CO<sub>2</sub> varies dramatically with the pressure and temperature, especially near the critical point, and therefore the terminal temperature difference will appear at the cold or hot end of the LTR as SR<sub>1</sub> changes. At first, with the increase of SR<sub>1</sub>, less amount of S-CO<sub>2</sub> passes through the PC and MC, resulting in work consumption decrease in the MC and increase in the RC; accordingly, the total work consumption increases. In addition, the outlet temperature of the cold side

TABLE 1: Initial parameters.

Parameter	Symbol	Unit	Value
Electric power output	$W_e$	WM	300
MC inlet pressure	$P_1$	Mpa	7.6
MC inlet temperature	$t_1$	°C	32
HPT inlet pressure	$P_5$	Mpa	25
HPT inlet temperature	$t_5$	°C	600
LPT inlet pressure	$P_7$	Mpa	14
LPT inlet temperature	$t_7$	°C	600
Terminal temperature difference	$\Delta t$	°C	8
MC and RC isentropic efficiency	$\eta_{MC}, \eta_{RC}$	%	90[16]
HPT and LPT isentropic efficiency	$\eta_{HPT}, \eta_{LPT}$	%	93[16]
Generator efficiency	$\eta_G$	%	98.5
Exhaust temperature	$t_{py}$	°C	110
Environment temperature	$t_0$	°C	20

TABLE 2: Compositions of coal and desulfurizer.

Coal and desulfurizer	Compositions	Unit	Value
Ultimate analysis of bituminous coal (As-received)	Carbon	%	62.96
	Hydrogen	%	4.13
	Oxygen	%	6.73
	Nitrogen	%	1.46
	Sulfur	%	1.22
	Moisture	%	10.00
Heat value (LHV)	Ash	%	13.50
	$Q_{net,ar}$	MJ/kg	24.72
Limestone	Calcium carbonate	%	95
	Moisture	%	0.2
	Impurities	%	4.8

fluid in HTR continues to rise, leading to a reduction in boiler heat absorption. The extent of decrease of boiler heat absorption is greater than the increase of compressor work consumption, which constantly improves the cycle efficiency until LTR reaches the temperature difference restriction at the hot end. In such a condition, the minimum temperature differences are achieved at both ends of LTR. LTR realizes an excellent temperature matching in which the negative impact of specific heat difference falls to zero and thermal efficiency reaches the maximum. Then, continuing to raise SR<sub>1</sub> leads to the sharp decline of LTR effectiveness and thermal efficiency as a result. The later analyses are based on optimal SR<sub>1</sub> values under different operating conditions.

Figure 4 also indicates impacts of different terminal temperature differences on thermal efficiency. The terminal temperature difference represents recuperators heat transfer capability, smaller temperature difference gets better heat transfer capacity and higher cycle efficiency, but it will need greater volumes and higher cost. In addition, larger heat transfer areas increase pressure drops in recuperators, which decrease efficiency conversely. The relevant study indicates that minimum temperature difference lower than 4°C is not economical [28].

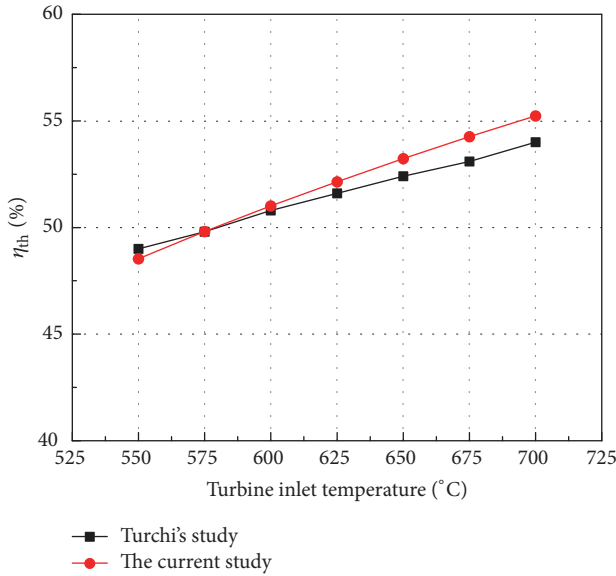


FIGURE 3: Validation of thermodynamic model by Turchi's study.

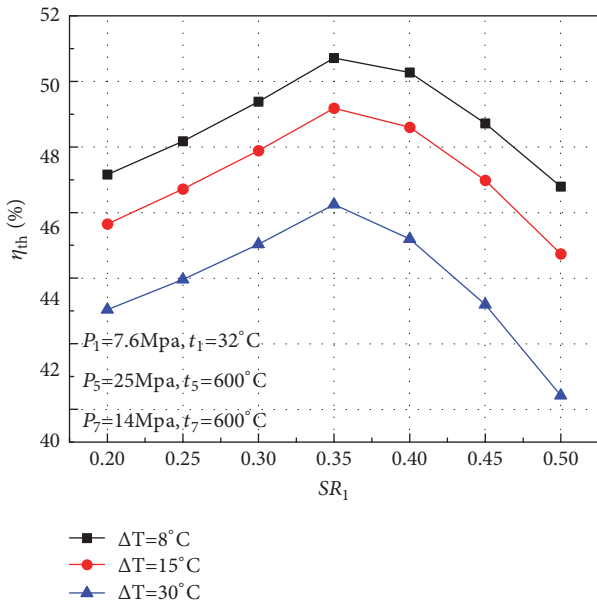


FIGURE 4: Effects of SR<sub>1</sub> on cycle efficiency at different terminal temperature differences.

Figure 5 illustrates effects of HPT inlet pressure on optimal SR<sub>1</sub>. The optimal SR<sub>1</sub> declines in the approximately linear proportion to HPT inlet pressure. The same mass flow in recuperator of simple recuperated cycle causes larger heat addition while bypassing the LTR of recompression cycle leads to higher average heat absorption temperature and thermal efficiency improvement. As Figure 5 presents, average specific heat differences between high-pressure and low-pressure side fluids are gradually narrowing with increasing HPT inlet pressure and a greater proportion of S-CO<sub>2</sub> is permitted to access to the PC and MC; thereby optimal values of SR<sub>1</sub> decrease.

3.2. *Split Ratio to the ECO (SR<sub>2</sub>)*. Figure 6 shows effects of SR<sub>2</sub> on boiler efficiency, thermal efficiency, and net efficiency. It demonstrates that the split flow exiting from high-pressure side of the LTR can effectively recover moderate flue gas heat and improve boiler efficiency. Simultaneously, cycle thermal efficiency maintains constant and net efficiency increases accordingly. Boiler heat losses and corresponding efficiency are preliminarily calculated according to semiempirical formulas and historical operation data of steam CFB boiler. Suppose the minimum temperature approach of countercurrent LRH located above the ECO is 55°C and hot air is preheated to 300°C, exhaust gas temperature gradually decreases and boiler efficiency rises with the increase of SR<sub>2</sub>. Furthermore, thermal efficiency maintains constant because heat recovered by the HTR does not change with a low proportion of S-CO<sub>2</sub> entering into the ECO. When SR<sub>2</sub> is equal to 0.076, the exhaust temperature falls to 110°C, and thus boiler efficiency is improved to 94%; the net efficiency can reach about 47.2%.

3.3. *HPT Inlet Parameters*. Figure 7 shows effects of HPT inlet pressure on cycle efficiency at different HPT inlet temperature. The cycle efficiency is improved with the increase of HPT inlet pressure, but the rate of increase gradually slows down and eventually reaches a stable state. At first, the increment of specific work output of the HPT is greater than total specific work consumption of the MC and RC as HPT inlet pressure increases, 1Mpa increase in HPT inlet pressure leads to about 0.5% improvement in cycle efficiency. The increment rate of specific net work continues to fall with HPT inlet pressure rising. In these conditions, the benefit of efficiency improvement is less obvious and materials pressure-resistance requirement becomes stricter, leading to larger material cost.

It can also be observed from Figure 7 that HPT inlet temperature has a positive influence on system performance. The degree of efficiency improvement caused by HPT inlet temperature is more noticeable under higher HPT inlet pressure. The cycle efficiency is improved by nearly 0.35% with HPT inlet temperature increasing by 20°C under HPT inlet pressure of 20Mpa. The cycle efficiency is improved by nearly 0.6% with HPT inlet pressure of 35Mpa under the same HPT temperature interval.

3.4. *Reheat Stages and Parameters*. Reheat is an effective approach for steam Rankine cycle efficiency improvement by increasing average heat absorption temperature. However, S-CO<sub>2</sub> Brayton cycle with lower pressure ratio has different results. As Figure 8 presents, single reheat can improve cycle efficiency, especially under higher HPT inlet pressure. The thermal efficiency is improved by 1.5% with HPT inlet pressure of 25Mpa. However, the efficiency of double reheat is enhanced by only 0.2% under the same condition. Larger reheat stages are undesirable due to more complex configurations and additional capital cost, which may offset the benefit of efficiency increment.

Figures 9 and 10 show effects of LPT inlet pressure on cycle efficiency at different HPT inlet pressure and LPT inlet temperature. As these figures reveal, LPT inlet pressure has optimal values corresponding to different HPT inlet pressure

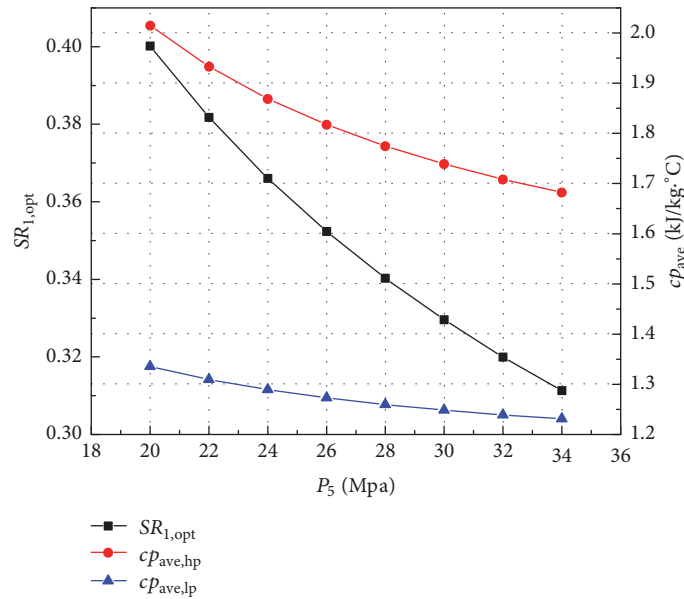


FIGURE 5: Effects of HPT inlet pressure on optimal  $SR_1$ .

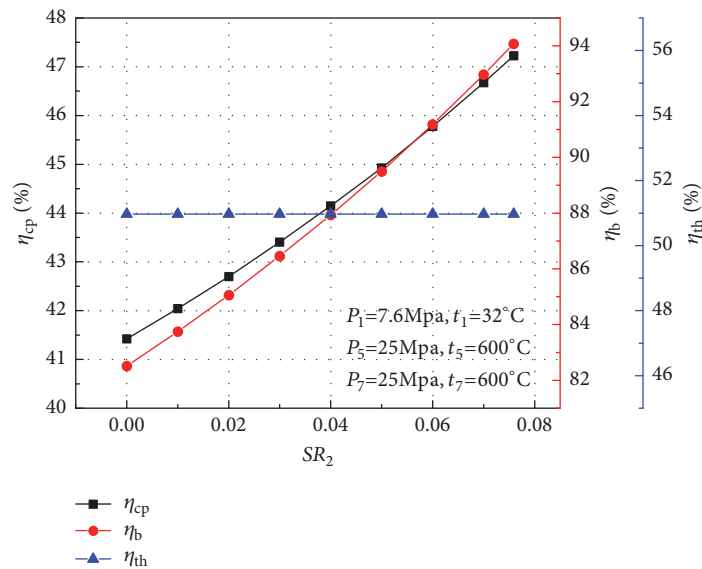


FIGURE 6: Effects of  $SR_2$  on boiler efficiency, thermal efficiency, and net efficiency.

and LPT inlet temperature. Firstly, the tendency of the change of HPT work output is opposite to the variation of LPT work output with increasing reheat pressure. HPT work output decreases and conversely LPT work output increases, which has an optimal reheating pressure. The corresponding optimal reheat pressures with different HPT inlet pressure of 20Mpa, 25Mpa, and 30Mpa are 13Mpa, 14.5Mpa, and 16Mpa respectively, which are slightly higher than the average of minimum and maximum pressures (12.6Mpa, 14.1Mpa, and 15.5Mpa). Figure 10 indicates that the efficiency is improved by nearly 0.5% with the reheat temperature increment of 20°C. Moreover, optimal reheat pressures are similarly affected by reheat temperature. Higher reheat temperature

corresponds to larger reheat pressure. Pressure ratios between HPT and LPT need to be redistributed due to various work output under different reheat temperature.

Figure 11 summarizes optimal reheat pressure values under different HPT inlet pressure and LPT inlet temperature, which can be used as a reference for the conceptual design of a power plant. These values are still affected by the pressure drop in reheater and detailed hydraulic calculation needs to be further carried out.

**3.5. MC Inlet Parameters.** Figure 12 shows effects of MC inlet pressure on cycle efficiency at different MC inlet temperature. MC inlet pressure similarity has optimal values,

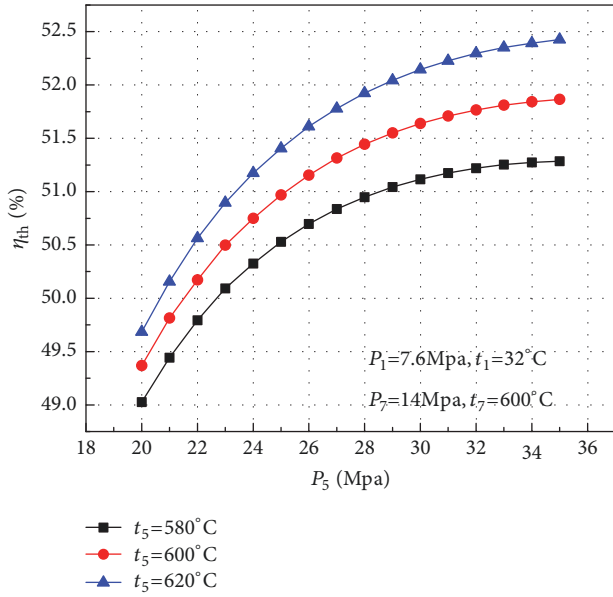


FIGURE 7: Effects of HPT inlet pressure on cycle efficiency at different HPT inlet temperature.

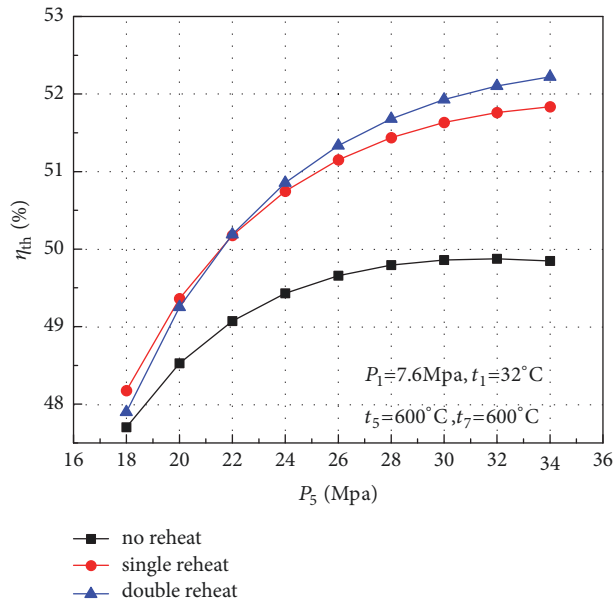


FIGURE 8: Effects of reheat stages on cycle efficiency at different HPT inlet pressure.

which correspond to the pseudocritical pressure of various MC inlet temperature. In addition, optimal MC inlet pressure increases and maximum cycle efficiency decreases with different MC inlet temperature. The specific reason is that turbine work output and compressor work consumption together decrease with increasing MC inlet pressure at first, but work consumption variation, especially in the MC, is greater due to the sharp decline of compressibility factor in the near-critical region, showing the incompressibility property of liquid phase, finally leading to continual rising of

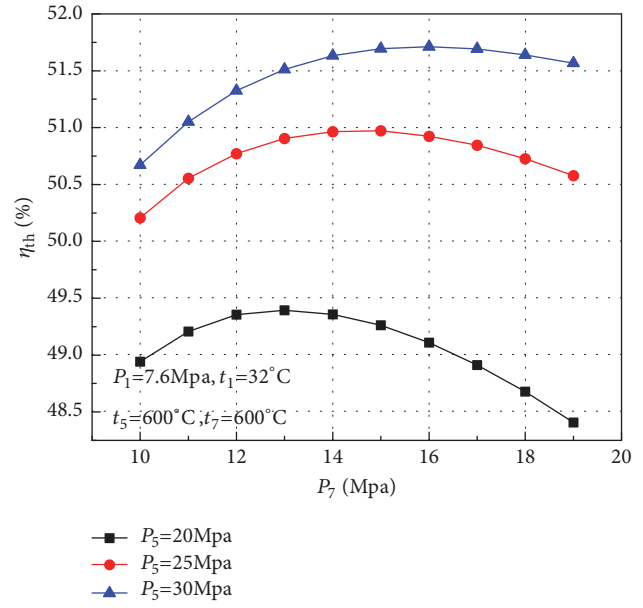


FIGURE 9: Effects of LPT inlet pressure on cycle efficiency at different HPT inlet pressure.

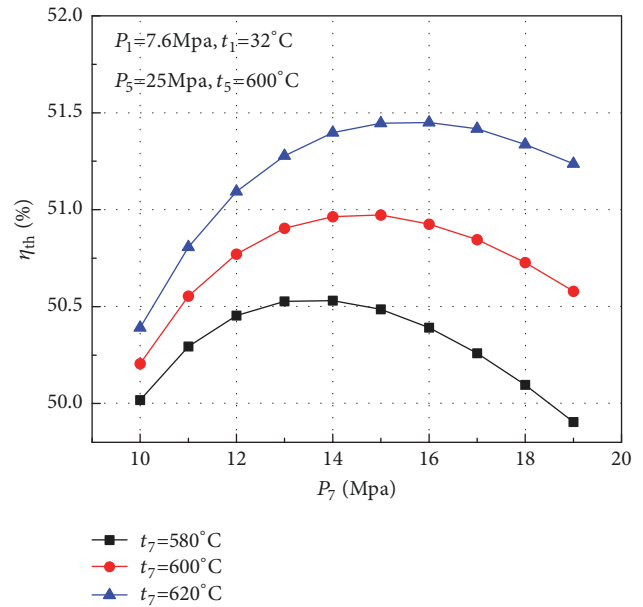


FIGURE 10: Effects of LPT inlet pressure on cycle efficiency at different LPT inlet temperature.

cycle efficiency. The benefit of lower compression work consumption near critical point gradually decreases with further increasing MC inlet pressure and temperature. Moreover, the total compressor work consumption increases with MC inlet temperature, resulting in a significant decrease of thermal efficiency especially under optimal MC inlet pressure.

#### 4. Conclusions

S-CO<sub>2</sub> recompression Brayton cycle is a promising alternative to be applied in coal-fired power generation due to special

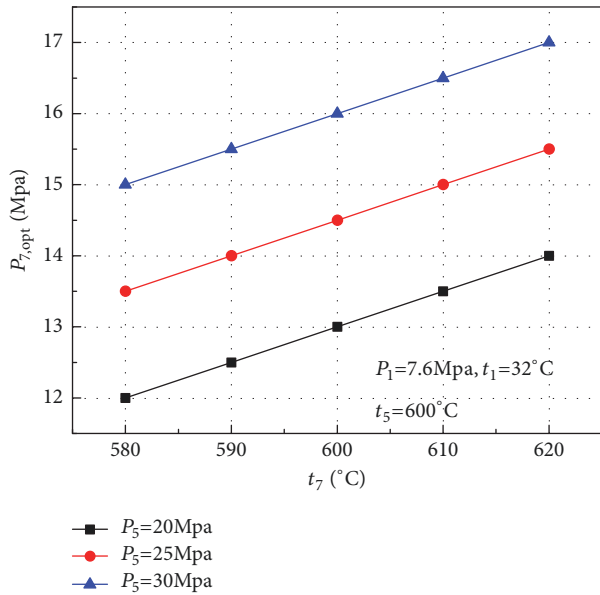


FIGURE 11: Optimal reheat pressure under different HPT inlet pressure and LPT inlet temperature.

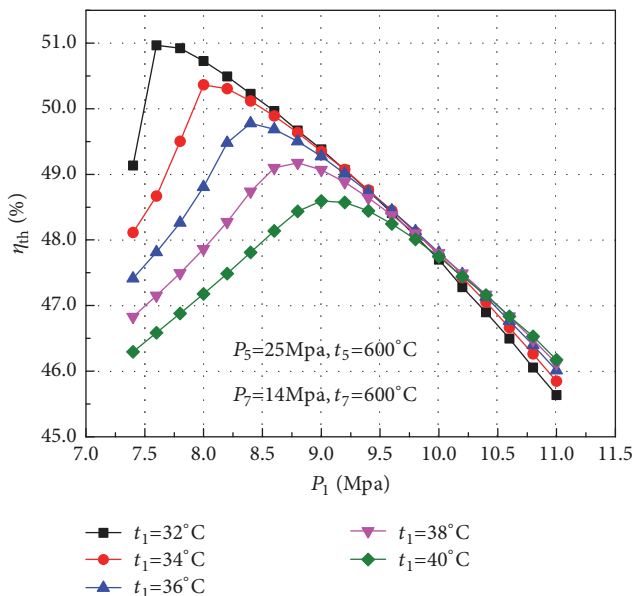


FIGURE 12: Effects of MC inlet pressure on cycle efficiency at different MC inlet temperature.

thermophysical properties in the near-critical region and high average heat absorption temperature as well as low pressure ratio, which contribute to higher efficiency and more compact turbomachinery compared to steam Rankine cycle.

CFB boiler with lower and uniform combustion temperature can be better integrated with S-CO<sub>2</sub> cycle. However, the overall system configuration needs to be further modified to match the narrow temperature range of heat source. This paper constructs recompression S-CO<sub>2</sub> Brayton cycle with single reheat based on the CFB boiler. Additional split flow extracted from LTR cold side is adopted in order to utilize flue

gas heat effectively. The net efficiency can reach 47.2% under the condition of 600°C/600°C/25Mpa.

Influences of critical parameters on system performance have been investigated systematically, including RC split ratio (SR<sub>1</sub>), ECO split ratio (SR<sub>2</sub>), HPT inlet pressure and temperature, and reheat layout as well as MC inlet pressure and temperature. The results show that SR<sub>1</sub>, reheat pressure, and MC inlet pressure have optimal values, among which appropriate temperature matching is achieved between the hot and cold side of the LTR, the optimal reheat pressure is slightly higher than the average pressure and is still affected by reheat temperature, and the optimal MC inlet pressure corresponds to pseudocritical pressure of various MC inlet temperature. Smaller terminal temperature difference corresponds to higher efficiency, but the cost and pressure drops of heat exchangers need to be taken into account. Reasonable SR<sub>2</sub> can recover moderate flue gas heat caused by narrow temperature range. Single reheat improves cycle efficiency by 1.5% with HPT inlet pressure of 25Mpa; however, double reheat for efficiency improvement is less obvious compared to steam Rankine cycle largely due to much lower pressure ratio. In addition, HPT and LPT inlet temperature both have positive influences on system performance. The cycle efficiency is improved by nearly 0.5% with HPT or LPT inlet temperature increasing by 20°C under HPT inlet pressure of 25Mpa.

## Abbreviations

S-CO<sub>2</sub>: Supercritical carbon dioxide  
 CSP: Concentrated solar power  
 CFB: Circulating fluidized bed  
 PFBC: Pressurized fluidized bed combustor  
 PC: Pulverized coal  
 CCS: Carbon capture and storage  
 LHV: Lower heating value  
 ZEPS: Zero Emission Power and Steam  
 NIST: National Institute of Standards and Technology

SR<sub>1</sub>: Split ratio to the recompressor  
 SR<sub>2</sub>: Split ratio to the economizer  
 LTR: Low temperature recuperator  
 HTR: High temperature recuperator  
 LPT: Low pressure turbine  
 HPT: High pressure turbine  
 MC: Main compressor  
 RC: Recompressor  
 PC: Precooler  
 ECO: Economizer  
 CW: Cooling wall  
 SH: Superheater  
 LRH: Low temperature reheater  
 HRH: High temperature reheater  
 AP: Air preheater.

## Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.



## Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.

## Acknowledgments

Financial supports from the National Key Research and Development Program of China (no. 2017YFB0601802) and Key Research and Development Program of Jiangsu Province (no. BE2017159) are sincerely acknowledged.

## References

- [1] V. Dostal, *A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors*, Massachusetts Institute of Technology, Massachusetts, Mass, USA, 2004.
- [2] Y. Ahn, S. J. Bae, M. Kim et al., "Review of supercritical CO<sub>2</sub> power cycle technology and current status of research and development," *Nuclear Engineering and Technology*, vol. 47, no. 6, pp. 647–661, 2015.
- [3] M.-J. Li, H.-H. Zhu, J.-Q. Guo, K. Wang, and W.-Q. Tao, "The development technology and applications of supercritical CO<sub>2</sub> power cycle in nuclear energy, solar energy and other energy industries," *Applied Thermal Engineering*, vol. 126, pp. 255–275, 2017.
- [4] E. G. Feher, "The supercritical thermodynamic power cycle," *Energy Conversion*, vol. 8, no. 2, pp. 85–90, 1968.
- [5] J. R. Hoffmann and E. G. Feher, "150 kwe supercritical closed cycle system," *Journal of Engineering for Gas Turbines and Power*, vol. 93, no. 1, pp. 70–80, 1971.
- [6] G. Angelino, "Carbon dioxide condensation cycles for power production," *Journal of Engineering for Gas Turbines and Power*, vol. 90, no. 3, pp. 287–295, 1968.
- [7] F. Crespi, G. Gavagnin, D. Sánchez, and G. S. Martínez, "Supercritical carbon dioxide cycles for power generation: a review," *Applied Energy*, vol. 195, pp. 152–183, 2017.
- [8] J. Sarkar, "Second law analysis of supercritical CO<sub>2</sub> recompression Brayton cycle," *Energy*, vol. 34, no. 9, pp. 1172–1178, 2009.
- [9] C. S. Turchi, Z. Ma, T. W. Neises, and M. J. Wagner, "Thermodynamic study of advanced supercritical carbon dioxide power cycles for concentrating solar power systems," *Journal of Solar Energy Engineering*, vol. 135, no. 4, article 41007, 2013.
- [10] S. A. Wright, R. F. Radel, M. E. Vernon, P. S. Pickard, and G. E. Rochau, *Operation and Analysis of a Supercritical CO<sub>2</sub> Brayton Cycle*, Sandia, New Jersey, NJ, USA, 2010.
- [11] B. D. Iverson, T. M. Conboy, J. J. Pasch, and A. M. Kruiuzenga, "Supercritical CO<sub>2</sub> Brayton cycles for solar-thermal energy," *Applied Energy*, vol. 111, pp. 957–970, 2013.
- [12] T. J. Held, "Initial test results of a megawatt-class supercritical CO<sub>2</sub> heat engine," in *Proceedings of the 4th International Symposium on Supercritical CO<sub>2</sub> Power Cycles*, 2014.
- [13] C. Turchi, *10 MW Supercritical CO<sub>2</sub> Turbine Test*, National Renewable Energy Laboratory, 2014.
- [14] K. Brun, P. Friedman, and R. Dennis, *Fundamentals and Applications of Supercritical Carbon Dioxide (SCO<sub>2</sub>) Based Power Cycles*, Woodhead Publishing, Cambridge, UK, 2017.
- [15] Y. Muto, S. Ishiyama, Y. Kato, T. Ishizuka, and M. Aritomi, "Application of supercritical CO<sub>2</sub> gas turbine for the fossil fired thermal plant," *Journal of Energy and Power Engineering*, vol. 9, pp. 7–15, 2010.
- [16] Y. Le Moullec, "Conceptual study of a high efficiency coal-fired power plant with CO<sub>2</sub> capture using a supercritical CO<sub>2</sub> Brayton cycle," *Energy*, vol. 49, no. 1, pp. 32–46, 2013.
- [17] M. Mecheri and Y. Le Moullec, "Supercritical CO<sub>2</sub> Brayton cycles for coal-fired power plants," *Energy*, vol. 103, pp. 758–771, 2016.
- [18] H. Li, Y. Zhang, and M. Yao, "Preliminary design study on a multi-megawatts fossil-based supercritical CO<sub>2</sub> recompression and reheat integral test facility," in *Proceedings of the 5th International Symposium on Supercritical CO<sub>2</sub> Power Cycles*, 2016.
- [19] Z. Bai, G. Zhang, Y. Li, G. Xu, and Y. Yang, "A supercritical CO<sub>2</sub> Brayton cycle with a bleeding anabranch used in coal-fired power plants," *Energy*, 2017.
- [20] W. W. Shelton and N. Weiland, "Oxy-coal-fired circulating fluid bed combustion with a commercial utility-size supercritical CO<sub>2</sub> power cycle," in *Proceedings of the 5th International Symposium on Supercritical CO<sub>2</sub> Power Cycles*, 2016.
- [21] G. A. Johnson, M. W. McDowell, G. M. O'Connor, C. G. Sonwane, and G. Subbaraman, "Supercritical CO<sub>2</sub> Cycle development at Pratt and Whitney Rocketdyne," in *Proceedings of the ASME Turbo Expo 2012: Turbine Technical Conference and Exposition*, 2012.
- [22] D. Thimsen and P. Weitzel, "Challenges in designing fuel-fired sCO<sub>2</sub> heaters for closed sCO<sub>2</sub> Brayton cycle power plants," in *Proceedings of the 5th International Symposium on Supercritical CO<sub>2</sub> Power Cycles*, 2016.
- [23] P. Basu, *Circulating Fluidized Bed Boilers: Design, Operation and Maintenance*, Springer, Berlin, Germany, 2015.
- [24] E. W. Lemmon, M. L. Huber, and M. O. McLinden, NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP. 9.0, 2010.
- [25] R. Span and W. Wagner, "A new equation of state for carbon dioxide covering the fluid region from the triple-point temperature to 1100 K at pressures up to 800 MPa," *Journal of Physical and Chemical Reference Data*, vol. 25, no. 6, pp. 1509–1596, 1996.
- [26] X. Sun and Z. Huang, *Technology and Engineering Application of Large Circulating Fluidized Bed Boiler*, China Electric Power Press, Beijing, China, 2013.
- [27] J. C. Bryant, H. Saari, and K. Zanganeh, "An analysis and comparison of the simple and recompression supercritical CO<sub>2</sub> cycles," in *Proceedings of the Supercritical CO<sub>2</sub> Power Cycle Symposium*, 2011.
- [28] R. Dennis, "Overview of supercritical carbon dioxide based power cycles for stationary power generation," in *Proceedings of the 4th International Seminar on ORC Power Systems*, 2017.



**Hindawi**

Submit your manuscripts at  
[www.hindawi.com](http://www.hindawi.com)

