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Technical Note

Conceptual design of a super-critical CO2 brayton cycle based on stack waste heat recovery for shazand power plant in Iran

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ABSTRACT

Conceptual design of a waste heat recovery cycle is carried out in attempt to enhance the thermal efficiency of a steam power plant. In the recovery system, super-critical an CO_2 is employed as the working fluid operating in a Brayton cycle. Low grade heat rejected by the flue gases through the stack is used as the primary heat source, while a secondary heat exchanger utilizes the hot gases leaving the economizer to heat of CO_2 up to desired temperature. In the present work, a case study for a 325 MW steam power plant of Shazand in Iran is carried out and a thermodynamic model is developed to predict the performance of the system. Regarding that the proposed recovery cycle may lead to less effective air preheating and affecting the combustion efficiency, an optimization process has been conducted to determine the optimum conditions. It's been also considered that excessive decline in flue gas temperature leaving the stack may result in the condensation and accumulation of corrosive substances on the inner surface of the stack. The results demonstrate that this waste heat recovery system can deliver up to 18 MW of net power which corresponds to an increase of 1.58 percent in thermal efficiency of the power plant. Obtained results magnify the importance of this innovative design, consequently illustrate the necessity for using waste heat recovery system.

Keywords: Brayton cycle, Super-critical CO_2, Waste heat recovery.

1. Introduction

Concerns about increasing energy demand, growing energy price and limited fossil fuel resources as well as environmental problems such as global warming and greenhouse gas emissions have heightened interest in usage of waste heat recovery systems. Furthermore, economic advantages may be obtained through devising methods for increasing the overall efficiency of power generation units. In steam power plants for example, the thermal efficiency hardly reaches 40 percent, meaning that 60 percent of the input energy is wasted. Thus, attempts have been made to recover this significant amount of waste heat. While heat is rejected from the condenser into the cooling towers in a steam power plant, it is generally of a low grade and is therefore used for cogeneration purposes. The high temperature flue gases, leaving the stack, can potentially be utilized in power generation waste heat recovery cycles.

Selection of a suitable working fluid is a major step in designing a waste heat recovery system. As a principal parameter, the properties of the working fluid decide the performance of the system. Diverse substances have been suggested to be employed as working fluid in heat recovery cycles. Steam, organic fluids and supercritical carbon dioxide are among the most popular. Steam is the primary choice for designing a gas turbine compound cycle whereas organic fluids such as R-32 are most commonly used in organic Rankine cycle (ORC). Recent studies, however, suggest that the supercritical Carbon dioxide ($ScCO_2$) has certain advantages compared to former working fluids.

Compared to organic and steam-based Rankine cycle systems, $ScCO_2$ can achieve high efficiencies over a wide temperature range of heat sources with compact components resulting in a smaller system footprint, lower capital and operating costs [1]. Non-flammability, non-toxicity and non-corrosiveness are characteristics that make $ScCO_2$ an ideal working fluid for closed loop power generation applications.

Furthermore, low cost and abundance of carbon dioxide are features that enhance the affordability of a system run by $ScCO_2$.

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Non-flammability, non-toxicity and non-corrosiveness are characteristics that make $ScCO_2$ an ideal working fluid for closed loop power generation applications. Furthermore, low cost and abundance of carbon dioxide are features that enhance the affordability of a system run by $ScCO_2$. In addition to the aforementioned favorable characteristics, the high density of $ScCO_2$ enables extremely compact turbo machinery designs and microchannel-based heat exchangers [1]. In the case of heat sources with approximately constant heat capacity such as hot flue gases, heat absorption can be performed more effectively due to the heat capacity characteristics of carbon dioxide in supercritical region.

Many approaches have been proposed to run a power generation cycle with $ScCO_2$ as the working fluid. Studies carried out in this area generally include the thorough planning of a power plant as well as designing waste heat recovery systems.

Le Moullec [6] conducted a conceptual design of a $ScCO_2$ power plant. Persichilli et al. [1] suggested a $ScCO_2$ power generation cycle, purposefully designed to capture waste heat in gas turbine exhausts. Di Bella [11] also proposed an inventive heat recovery power generation system.

In addition, remarkable attention has been given to application of $ScCO_2$ power generation cycles in renewable energy systemssuch as concentrating solar power [2, 9, 14, 16].

Recently a substantial amount of effort has been dedicated to generating concepts for nuclear power plants with aim to achieve higher efficiency and minimum safety issues [10, 12,15]. Yoon et al. [7] assessed the potential of $ScCO_2$ Brayton cycle as a favorable candidate for the next generation nuclear reactors power conversion systems. He also determined that $ScCO_2$ Brayton cycle pressure ratio has a significant influence on cycle efficiency [13].

recompression Brayton cycle with $ScCO_2$ adopted as working fluid.

Y.M.Kim *et al.* [8] proposed a novel transcritical or fully cooled $ScCO_2$ Rankine cycle, using both the low temperature and the high temperature heat sources.

Vidhi *et al.* [17], however, announced that between a carbon dioxide-based supercritical Rankine cycle and an organic fluid-based supercritical Rankine cycle, the former needs higher pressure to achieve the same efficiency and a heat recovery system is necessary to desuper heat that the turbine exhausts and pre-heat pressure charged liquid.

Much work has been carried out on the cycle performance, heat transfer and fluid flow in the $ScCO_2$ power generation cycles. Kulhánek and Dostál [5] analyzed various Brayton cycles and declared that the recompression cycle achieves higher efficiency.

In this paper, innovative techniques are implemented to enhance the efficiency of a 325 MW steam power plant in Shazand, Iran.

A Brayton cycle with $ScCO_2$ as the working fluid is coupled with the Rankine cycle and the stack gases of the power plant steam generator are used as the primary heat source of the Brayton cycle. Moreover, the path of the flue gases from the boiler furnace to the stack is altered in a way that lessens heat consumption in the air preheater. Instead, the surplus heat is used as a high temperature heat source for the $ScCO_2$ cycle which results in greater power obtained from the recovery system. The furnace combustion efficiency, however, highly depends on the temperature of preheated combustion air. Thus, considerations have been made in order to prevent intake air from entering the boiler furnace at low temperatures.

2. Thermal Modeling

Modeling of the power generation heat recovery system is developed based on the data from fossil-fuel steam generator of a 325 MW steam power plant in Shazand, Iran. Current working conditions of the aforementioned steam generator such as flow rates and temperatures are listed in Table 1 [19].

Table 1. Working condition data of Shazand 325 MW steam powerplant, Iran

Unit Thermal Efficiency (%)	39.45
Fuel consumption (kg/s)	17.75
Intake Air Mass Flow Rate (kg/s)	564.417
Flue Gas Mass Flow Rate (kg/s)	582.167
Flue Gas Temperature Leaving the Stack (^oC)	126
Flue Gas Temperature at Air-Preheater Inlet (^oC)	355
Air Temperature at Furnace Inlet(^oC)	269
Intake Air Temperature (^oC)	10
Ambient Temperature (^oC)	10
Excess Air (%)	5
Smoke Density (kg/ m^3)	0.765
Smoke Specific Heat (kJ/kg.C)	1108.25
Fuel Density (kg/m^3)	1.2906
Fuel High Heat Value (MJ/ m^3)	38.6
Fuel Low Heat Value (MJ/ m^3)	35.502

Since gases leaving the stack are still relatively hot approximately around 130°C, the recovery cycle uses the stack flue as the first heat source. Extracting heat from the stack, it should be considered that temperature of the gas leaving the stack should be kept well above the dew point of the water vapor within the gases in order to prevent condensation. Otherwise, it may lead to formation of acids and bring on corrosion. In addition, the flue gases should have enough buoyancy to rise in a high plume above the stack for proper atmospheric dispersion. For the present steam generator which burns natural gas (mostly methane) with 5 percent excess air, the minimum stack temperature to avoid condensation may be calculated by solving the combustion equation given as

$$CH_4 + 2.1\, O_2 + 7.896\, N_2 \tag{1}$$

$$\rightarrow CO_2 + 2\, H_2O + 0.1\, O_2 + 7.896\, N_2\,. \tag{2}$$

The partial pressure of the water component in the gas mixture is given by

$$P_{water} = P_{total} \times mole\, fraction_{water}\,, \tag{3}$$

which gives

$$P_{water} = 2.01 \times \frac{2}{1 + 2 + 0.1 + 7.896} = 0.182\ bar\,. \tag{4}$$

This pressure corresponds to a saturation temperature of 85°C according to saturated water thermodynamic tables. The average temperature of the gases must be kept well above this value in order to avoid local cool spots that might cause condensation.

3. ScCO2 Cycle Description

The schematic layout and principal arrangement of the waste heat recovery system is presented in Fig. 1. A Brayton ScCO2 cycle is based on a simple Brayton cycle, working at high pressures and utilizing ScCO2 as the working fluid in a closed loop. According to previous studies, a simple Brayton cycle provides a more reliable working condition and a simpler scheme [4,5]. The proposed power generation system consists of five main components:

1. A primary heat exchanger where waste heat energy is absorbed from the stack of the power plant.
2. Secondary heat exchanger utilizes the energy from the flue gases prior to the air preheater section of the steam generator to heat the $ScCO_2$ up to its maximal temperature.
3. A gas turbine working at 200 bars in order to convert $ScCO_2$ enthalpy to mechanical power. A higher Turbine Inlet Temperature (TIT) results in a higher thermal efficiency while a minimum value of 220°C is known to be the least feasible TIT for power generation in a closed loop $ScCO_2$ Brayton cycle [1]. Isentropic efficiency of the gas turbine is assumed to be 93% which is a reasonable assumption for a multi-stage turbine working with $ScCO_2$ as the working fluid [6]. The rotor tip geometry is also a key component influencing the performance of the gas turbine [20].
4. A Recuperative heat exchanger is utilized to recover the heat leaving the gas turbine and convey the energy to the intake combustion air of

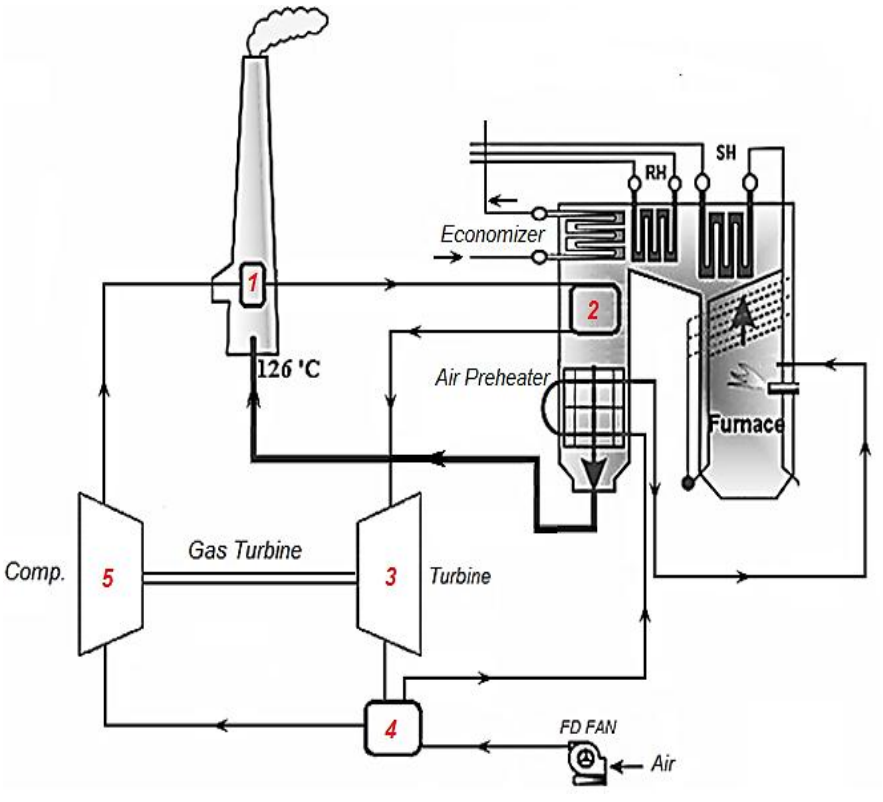

Fig. 1. Schematic layout and principal arrangement of the waste heat recovery system

the boiler furnace. The abovementioned Recuperator acts as a precooler to decrease the temperature of the outlet $ScCO_2$ from the turbine. Using ambient air at 10°C (according to meteorological data provided for the yearly average temperature in Shazand, Iran [19]) as the cold fluid, the recuperator cools the low pressure $ScCO_2$ down to 32°C. Moreover, it contributes to the air-preheater to warm upthe combustion air effectively and enhance the combustion efficiency of the boiler furnace.

5. A Compressor in which the low pressure $ScCO_2$ at 32°C and 78 bars is received from the precooler. The compressor inlet is chosen to be near the thermodynamic critical point of the carbon dioxide in order to boost the cycle performance and achieve greater output. The compressor working at the optimal pressure ratio of 2.56 [13], compresses the working fluid up to 200 bars. Considering the high density of the carbon dioxide in super critical state, compact and efficient designs of power cycle compressor are feasible. Isentropic efficiency of 90 percent is chosen for the compressor which is a typical value for a large heavy duty compressor. Power consumed by the compressor is taken to be equal to increase in enthalpy of $ScCO_2$, considering the isentropic efficiency.

Pressure drop is an important aspect in such power cycles; however, in a waste heat recovery supercritical system with compact dimensions, effects of pressure loss on the performance of the system may be neglected.

4. Thermodynamic Analysis

Regarding that an appropriate design of the recovery cycle has a strong influence on its effectiveness, a comprehensive thermodynamic analysis is necessary in order to select the optimum conditions. The function of this system is such that heating $ScCO_2$ in the secondary heat exchanger will result in greater net power in the recovery cycle; however, it would adversely affect the operation of the steam generator air preheater as there would be less heat available to warm up the combustion air. Since air preheating has increased boiler furnace combustion efficiency, calculations have been carried out to obtain the optimum working conditions. With the steam power plant, fuel saving rate being set as the objective great impacton the function, an algorithm has been generated; consequently, $ScCO_2$ temperature rises in the second heat exchanger and it is optimized through exhaustive search method.

Before beginning the solution, one has to predict the nature of the optimal solution. It is highly desirable to use the waste heat from the stack maximally, as this energy may not benefit the steam power plant any longer. However, in the second phase of heating (in the second heat exchanger), there must be a compromise between trying to utilize the heat for increasing the $ScCO_2$ cycle output and maintaining the combustion efficiency at the steam cycle boiler.

The fuel saving rate obtained from preheated combustion air has been investigated previously (Table 2) [18].

Data presented in Table 2 are then generalized to be used in the algorithm. In each step, $ScCO_2$ temperature is increased by 5 degrees in the second heat exchanger before entering the gas turbine. The net power delivered by the recovery system is then calculated this power corresponds to an increase in the overall power plant power output. The process results in a slump in the power plant fuel rate which can be interpreted as fuel saving. On the other hand, reduction in fuel saving caused by less effective air preheating in the steam generator is also determined by the generalized data from Table 2. Total fuel saving rate is finally calculated in each step and the search continues until $ScCO_2$ reaches the maximum possible temperature. Heat transfer constraints in the heat exchangers are considered so that the cold fluid temperature does not exceed the warm fluid inlet temperature.

Moreover, as the boiler furnace intake air passes through the recuperator and then enters the air-preheater before being combusted, the gas turbine outlet temperature is monitored to remain below the air-preheater temperature.

The objective function- assumed to be the overall

Table 2. Impact of air preheating on fuel saving rate [18]

Furnace Exhaust Temperature, °C	Preheated Air Temperature, °C					
	315	430	540	650	760	870
540	13	18	-	-	-	-
650	14	19	23	-	-	-
760	15	20	24	28	-	-
870	17	22	26	30	34	-
980	18	24	28	33	37	40
1090	20	26	31	35	39	43
1200	23	29	34	39	43	47
1310	26	32	38	43	47	51

fuel saving rate of the steam power plant- is an implicit function of the following form:

$$\Phi = f(TIT, Q) \tag{5}$$

As elaborated above, this problem is apparently constrained. It is desirable to maximize the objective function Φ, subject to the following constraints (in which all the temperatures refer to the heat recovery cycle):

$$C1: T_{Turbine,in} > 220°C \tag{6}$$

$$C2: T_{Turbine,in} < 359°C \tag{7}$$

$$C3: T_{Turbine,out} < T_{air\ preheater,in} \tag{8}$$

The optimum temperature of $ScCO_2$ before entering the gas turbine corresponds to the point where the maximum fuel saving rate is acquired. This analysis is carried out for different mass flow rates of $ScCO_2$.

5. Results

Figure 2 shows the fuel saving rate versus temperature at which $ScCO_2$ enters the gas turbine. It is evident that for mass flow rates of and below 200 kg/sec, the fuel saving rate increases as gas turbine inlet temperature is raised. However, for greater $ScCO_2$ mass flow rates, there would be an optimum point at which fuel saving is maximum as the objective function. Even though heating $ScCO_2$ above the optimum point in these cases will wind up in greater power in the gas turbine Its adverse effect on air preheating is great enough to bring on inefficient fuel combustion in the boiler furnace; consequently, fuel saving rate begins to dwindle. As it is shown in Fig. 3, it is also observed that using mass flow rates more than 300 kg/sec has little contribution to fuel saving enhancement. Moreover, excessive flow rates demand more complicated pipelines and equipment. Thus, it is sensible to keep $ScCO_2$ mass flow rate in range of 250-300 kg/secwith gas turbine inlet temperature of 275-325°C. The net power obtained from the recovery cycle in optimum conditions is calculated for different $ScCO_2$ mass flow rates as shown in Fig. 4.

With the mass flow rate of 300 kg/sec being set as the operating conditions, the following information is derived in Table 3. After determining the final design parameters and executing the thermodynamic analysis thoroughly, temperature of the flue gases at the terminal stage of the stack is calculated to be 100°C which is 26°C lower than its initial value. The decline in the temperature of the flue gas has negative impacts as previously discussed. Although, the temperature of the gases leaving the stack is still high above the calculated condensation limit, temperature reduction has a strong effect on the plume rise. In order to maintain the plume height for ecological considerations, certain calculations have to be conducted. Holland's equation is suitable for

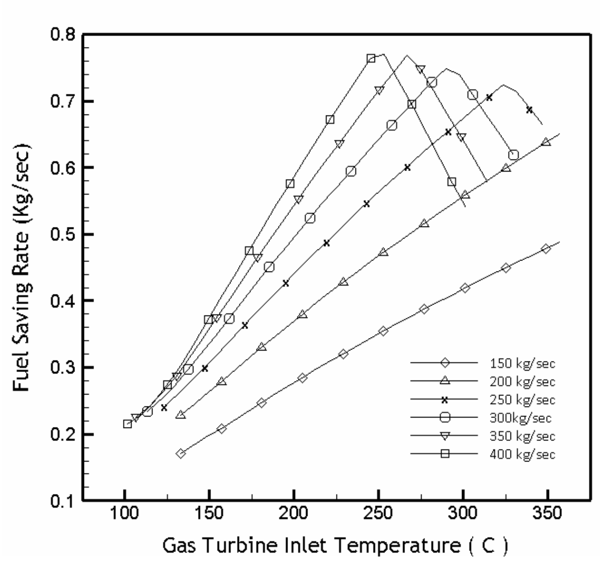

Fig. 2. Fuel saving rate VS. TIT for different mass flow rates

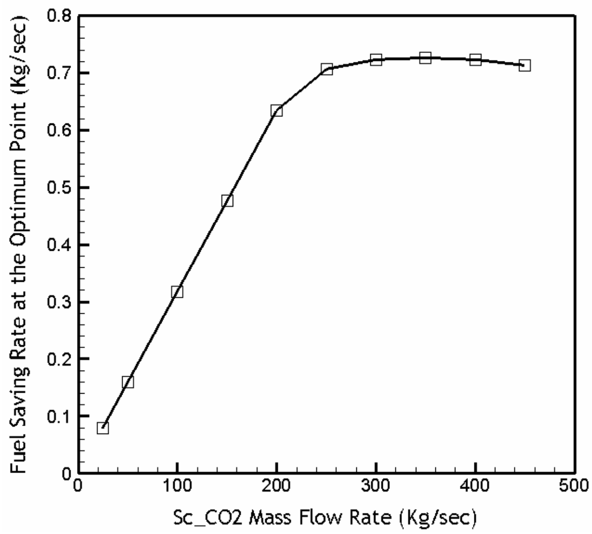

Fig. 3. Fuel saving rate at the optimum point VS.$SCCO_2$ mass flow rate

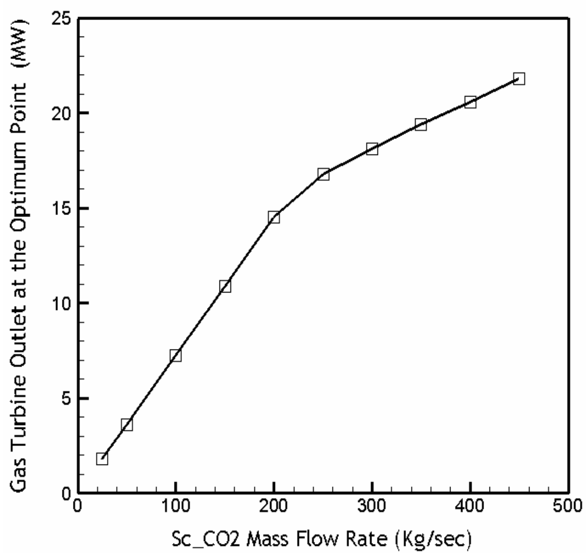

Fig. 4. Gas turbine outlet power at the optimum point VS.$ScCO_2$ mass flow rate

calculating the plume rise assuming stable situation of atmosphere in shazand, Iran:

$$\Delta h = \frac{V_{stack} D}{U} \times (1.5 + 2.68 \times 10^{-3} PD \frac{T_s - T_a}{T_s}) \qquad (9)$$

An increase of 22% in the operating pressure of the stack is mandatory to compensate for negative effects of flue gas temperature decrease on the plume height. The pressure at the stack can be adjusted favorably by increasing the power on the forced draft fan of the steam generator.

6. Conclusion

In this paper, the conceptual design of a waste heat recovery cycle is conducted for a 325 MW steam power plant in Shazand, Iran. A Brayton cycle with $ScCO_2$ as working fluid is utilized to absorb the heat rejected from the stacks. In addition, the structure of the steam generator is altered to permit the recovery cycle to gain additional heat from a heat exchanger installed prior to the air-preheater section. Optimization process has then been carried out to determine the optimum working conditions, so that inefficient air-preheating is prevented. Moreover, the design of the recovery cycle is such that its rejected heat contributes to combustion-air preheating. The results suggest that $ScCO_2$ mass flow rate of 250-300 kg/sec with gas turbine inlet temperature of 300-325°C are the optimum conditions which maximize the total fuel saving rate. With the net power of 18 MW generated, this system boosts the output power of the power plant up to 5.5 percent which is equivalent of the main cycle to 1.58% enhancement in the overall efficiency.

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