THERMODYNAMIC OPTIMIZATION OF RECUPERATED S-CO_2 BRAYTON CYCLES FOR WASTE HEAT RECOVERY APPLICATIONS

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ABSTRACT

Accelerating growth of electricity demand, fuel cost and environmental pollution along with concerns of energy resources becoming scarce necessitate looking for effective energy saving solutions. To this end, waste heat recovery from energy intensive plants presents great potential. In order to efficiently convert the waste heat to power, Supercritical Carbon Dioxide (S-CO_2) power cycles have recently been proposed. The S-CO_2 Brayton cycles offers promising features such as high cycle efficiency, compactness and low capital cost. The S-CO_2 cycles are suitable for a broad range of heat source temperatures, and can be used in stand-alone applications as well as the replacement of steam cycles in CHP and combined cycle power plants. In this paper, two configurations of the S-CO_2 Brayton cycles (i.e., the single-recuperated and recompression cycles) are thermodynamically modeled and optimized. In contrast to traditional optimization approaches for solar or nuclear power applications in which the cycle efficiency is to be maximized, in waste heat recovery applications, the optimization objective is to maximize the power generation in the bottoming cycle. The proposed optimization framework is carried out by means of a genetic algorithm which is a robust method for multidimensional, nonlinear system optimization. The optimization process is comprehensive, i.e., all the decision variables including the inlet temperatures and pressures of turbines and compressors, the pinch point temperature differences, and the mass flow fraction of the main compressor are optimized simultaneously. The results demonstrate that the optimum turbine inlet temperature does not reach to its maximum allowable temperature and the

cycle is not necessarily fully recuperated. Ultimately, the optimum cycle design points including the optimum CO_2 mass flow rates are presented for various heat source temperatures.

INTRODUCTION

The power generation industry is facing new challenging issues regarding accelerating growth of electricity demand, fuel cost and environmental pollution. Electric power is expected to remain the fastest growing form of worldwide end-use energy through 2040, as it has been for several decades. At least one-third of the forecasted increase in worldwide energy demand through 2040 will be attributed to electric power generation. It has also been estimated that the worldwide net electricity generation will nearly double by the year 2040, from 18.5 trillion kilowatt-hours in 2010 to 35.3 trillion kilowatt-hours in 2040 (The Outlook for Energy: A View to 2040, 2014). Producing electricity via waste heat recovery is considered as a sustainable solution to the aforementioned accelerating demand. The U.S. DOE estimates that 280,000 MW discharged annually in the U.S. industries as waste heat. Electricity Potential from only industrial waste recovery is equal to 20% of U.S. Electricity Demand. Annual monetary saving is estimated to be between 70 to 150 Billion USD, with substantial reduction in greenhouse gases (U.S. Energy Information Administration, 2013 and BCS, 2008). The worldwide potential is even more considerable as these numbers represent the situation in only the United States. This paper discusses the use of a Brayton cycle with supercritical carbon dioxide (S-CO_2) as the working fluid for converting a portion of the waste heat into electric power.

Waste heat recovery can significantly help energy intensive industries which include Chemical and Petrochemical Plants, Iron, Steel and Aluminum Industries, Pulp & Paper Industry, Cement, Glass & Nonmetallic Minerals Industry. Over these different types of industries, there is a large variation in the temperatures at which waste heat is available (Table 1). Careful optimization, both at cycle level and at component level, is necessary to arrive at the optimal cycle configuration for S-CO_2 cycles for a specific application. This work involves initial cycle-level optimization so that a subsequent and more detailed component-level optimization can be performed later.

Compared to other conventional alternatives, S-CO_2 cycles are more efficient for low and medium temperature heat sources, are less expensive in both capital and life-cycle costs, have smaller footprints, and have lower pressure ratios. On the other hand, S-CO_2 cycles may be disadvantaged with significantly higher operating pressures as compared to most power plant cycles, and may involve multiple gas-to-gas heat exchangers. In particular, as compared to Rankine cycles for conventional steam turbine power plants, S-CO_2 cycles involve single phase turbine with no propensity for erosion, have better heat utilization and do not have water treatment issues. As compared to conventional Brayton cycles for gas turbines, S-CO_2 cycles are more efficient at lower turbine inlet temperatures, are efficient as stand-alone cycles as opposed to combined cycles, and do not require blade cooling for similar or superior efficiencies.

The S-CO_2 power cycles were originally patented by Sulzer (1950), and further studied by Feher (1968), Hoffman (1971), and Angelino (1967 and 1968). Although the cycle proposed by Feher operates entirely above the critical pressure, the compression takes place in the liquid phase taking advantage of a low temperature heat rejection process. Angelino also studied several condensation configurations of the S-CO_2 power cycles and compared them to the steam and perfect gas cycles. His comparison results clearly demonstrated superior performance of the partial condensation S-CO_2 cycle over the reheat steam cycle at turbine inlet temperature above 650 °C. However, considering the low critical temperature of carbon dioxide, his proposed cycles are feasible only where cooling water at temperature not higher than 12-15 °C is available year-round.

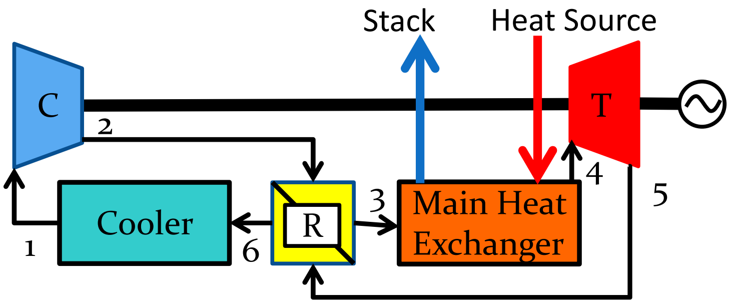
In one of the most extensive and comprehensive research studies, Dostal et al. (2004) investigated the S-CO_2 Brayton cycles with many modifications for nuclear power applications taking into account the sizing and performance of heat exchangers in detail. They state that the compactness of the system components and the size of an S-CO_2 gas turbine can be several folds smaller than the one that operates with Helium. Numerous studies and research on the S-CO_2 cycles have also been conducted by the U.S. National Labs. Harvego et al. (2011), from Idaho National Lab. (INL), evaluated the thermal efficiency of the recompression S-CO_2 Brayton cycle for different heat source temperature ranges and mass flow rates. Extensive studies have also been performed in Sandia National Lab. (SNL) and its sub-contractor, Barber Nichols Inc., to identify the important and critical technical issues in the design, development and application of the S-CO_2 power cycles published by Fleming (2012), Wright (2010), Conboy (2012), and Fuller (2012).

Table 1: Waste Heat Streams Classified by Temperature (Naik-Dhungel, 2012)

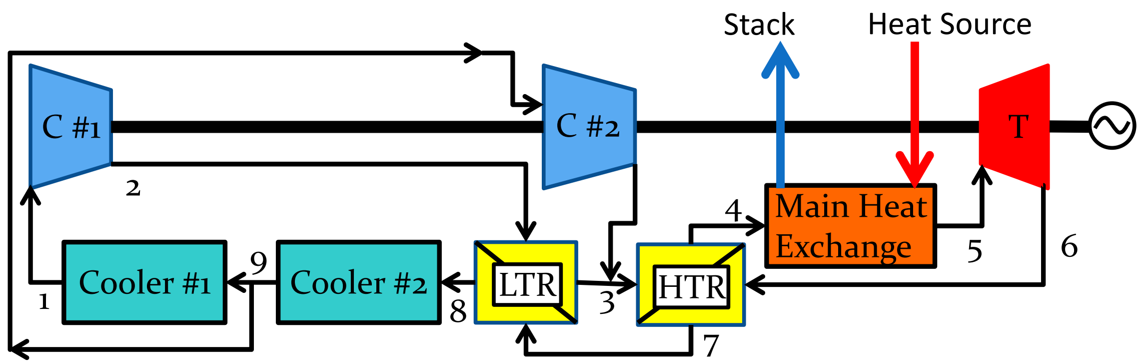
Temperature Classification	Waste Heat Source	Characteristics	Commercial Waste Heat to Power Technologies
High (>1,200 °F)	• Furnaces – Steel electric arc – Steel heating – Basic oxygen – Aluminum reverberatory – Copper reverberatory – Nickel refining – Copper refining – Glass melting • Iron cupolas • Coke ovens • Fume incinerators • Hydrogen plants	• High quality heat • High heat transfer • High power-generation efficiencies • Chemical and mechanical contaminants	• Waste heat boilers and steam turbines
Medium (500 –1,200 °F)	• Prime mover exhaust streams – Gas turbine – Reciprocating engine • Heat-treating furnaces • Ovens – Drying – Baking – Curing • Cement kilns	• Medium power-generation efficiencies • Chemical and mechanical contaminants (some streams such as cement kilns)	• Waste heat boilers and steam turbines (>500 °F) • Organic Rankine cycle (<800 °F) • Kalina cycle (<1,000 °F)
Low (< 500 °F)	• Boilers • Ethylene furnaces • Steam condensate • Cooling Water – Furnace doors – Annealing furnaces – Air compressors – IC engines – Refrigeration condensers • Low-temperature ovens • Hot process liquids or solids	• Energy contained in numerous small sources • Low power-generation efficiencies • Recovery of combustion streams limited due to acid concentration if temperatures reduced below 250 °F	• Organic Rankine cycle (>300 °F gaseous streams, >175 °F liquid streams) • Kalina cycle (>200 °F)

On the system level, the optimization of power cycles is very crucial. In an effort to find the global optimum deign point, in which the simultaneous optimization of all decision variables is indispensable, a new optimization approach based on a Genetic Algorithm (GA) is introduced to optimize the thermodynamic performance of S-CO_2 Brayton cycles.

Moreover, the previous studies on S-CO_2 cycles are mostly for applications such as nuclear and solar power generation, while S-CO_2 cycles also present great potential for waste heat recovery applications (CCGT and industrial waste heat). Echogen is currently the only manufacturer of S-CO_2 cycles for waste heat recovery applications. There have been numerous studies conducted at Echogen to compare the CO_2 and steam-based heat recovery systems published by Persichilli *et al* (2011), Kacludis *et al* (2012), and Persichilli *et al* (2012). Nevertheless, comprehensive studies, which present the optimum design variables of S-CO_2 Brayton cycles for various heat source temperatures, have not been conducted sufficiently. In this paper, two configurations of the S-CO_2 Brayton cycles i.e., the single-recuperated (RC) and recompression recuperated cycles (RRC) (as illustrated in Fig. 1) are thermodynamically modeled and optimized for different temperature ranges of heat recovery applications. Ultimately, a number of optimum design guidelines are presented.



a) Configuration: RC



b) Configuration: RRC

Figure 1: Plant layout for each power cycle configuration

THERMODYNAMIC MODELING

The modeling of the thermodynamic cycles follows the same process as reported by Mohagheghi and Kapat (2013). For easy reference of the readers, the primary steps are repeated here. The design and development of the S-CO_2 Brayton cycles are multidisciplinary efforts with several important aspects such as the integrity of mechanical systems, material compatibility and strength, vibration, machining methods, assembling, maintenance, etc. that appear in various layers of the design process. However, the focus of this study is the thermodynamic performance optimization of the S-CO_2 Brayton cycle; and it does not cover all the details involved in various layers of the design process. Nevertheless, the values of design parameters have been chosen in reasonable ranges to avoid any conflict with other aspects of the design and manufacturing.

S-CO_2 Properties

The first step in computational modeling of the S-CO_2 cycles is the calculation of thermodynamic properties of the working fluid. In order to calculate the thermodynamic properties of carbon dioxide, a FORTRAN code developed by the National Institute of Standards and Technology (NIST) in a software package named “REFPROP” (Lemmon *et al*, 2010) is used as subroutines in the main body of the modeling program. REFPROP is a computer program written to generate several databases for thermodynamic properties of working fluids. The generated data bases are based on multi-parameter equations of state that involve properties such as critical and triple points, and utilize experimental data for curve-fitting. The selected equations are applicable over the entire vapor and liquid regions of the fluid, including supercritical states; the upper temperature limit is usually near the point of decomposition of the fluid, and the upper pressure (or density) limit is defined by the melting line of the substance. In the case of carbon dioxide, the employed equations are obtained from the original work of Span and Wagner (1996).

Assumptions and Parameters

The basic assumptions in this paper are (1) steady state condition, (2) negligible heat losses in the recuperators and main heat exchanger (3) adiabatic turbines and compressors. Moreover, as suggested by many authors including Angelino in (1967) and Turchi et al. (2012), the pressure losses in the heat exchangers and ducts are also taken into account by introducing a fractional pressure drop (FPD) immediately downstream of the compressors.

It is also assumed that the minimum allowable temperature in the S-CO_2 cycles is 310 (K). It is noteworthy that this temperature is higher than the critical temperature of CO_2 (304.2 K), which means the cooling process in the cycle should happen in the temperature above the critical temperature. This condition ultimately leads to the exclusion of condensation CO_2 power cycles. The aforementioned assumptions are introduced to the modeling by means of certain input parameters which are summarized in Table 2.

Table 2: Input parameters

Input Parameters (unit)	Values
Ambient Temperature (K)	300
Minimum Allowable Cycle Temperature (K)	310
Fractional Pressure Drop (FPD)	0.02
Isentropic Efficiency of Turbines	0.9
Isentropic Efficiency of Compressors	0.89

In addition to the input parameters, the decision (design) variables are also inputs to the model. The optimization is performed on the decision variables which determine the thermodynamic performance of the cycles. The domain of optimization is specified by limiting the decision variables values between pre-assigned lower and upper bounds. The decision variables list and their variation domains are presented in Table 3.

The decision variables should be mathematically independent from each other. Thus, their selection is very crucial. The authors have tried to identify as many decision variables as possible in a way that the modeling tool demonstrates high flexibility and the optimization displays meaningful and valuable results.

Power Cycle Modeling

As illustrated in Fig. 1, the major components in the recuperated S-CO_2 Brayton cycles are compressors, turbines, recuperators, main recovery heat exchangers and coolers. The lumped modeling approach has been used in which each component is modeled by applying the conservation of mass and energy. Depending on the type of component (either turbomachineries or heat exchangers), supplementary equations such as Eq. (1), (2) and (3) are also employed to complete the sets of equations. The input parameters and decision variables are the known inputs to the model; and the unknowns are the dependent variables such as the outlet thermodynamic states of the compressors, turbines, and recuperators. Moreover, there are other unknowns (dependent variables) such as the efficiency, specific power, mass flow rates, etc. that can be calculated after all the thermodynamic states of the cycle are fixed. Note that a thermodynamic state is considered as a fixed (or known) state if two independent thermodynamic properties are known. This means all the required thermodynamic properties can be computed by knowing any combination of pressure with temperature, enthalpy, or entropy.

Table 3: Decision variables in the RC and RRC configurations

Decision Variables in Configuration: RC	Lower Bound	Upper Bound
Compressor Inlet Temperature, T(1)	310	410
Turbine Inlet Temperature, T(4)	350	Variable
Terminal Temperature Difference in Recuperators, ΔTt	10	40
Compressor Inlet Pressure, P(1)	2	Variable
Compressor Outlet Pressure, P(2)	Variable	24
Pinch Point Temperature Difference in Main Heater, ΔTPP	10	400
Decision Variables in Configuration: RRC	**Lower Bound**	**Upper Bound**
Main Compressor Inlet Temperature, T(1)	310	410
Turbine Inlet Temperature, T(5)	350	Variable
Inlet Temperature of Recompression, T(9)	T(1)	T(8)
Terminal Temperature Difference in Recuperators, ΔTt	10	40
Main Compressor Inlet Pressure, P(1)	2	Variable
Main Compressor Outlet Pressure, P(2)	Variable	24
Main Compressor Mass Flow Fraction, f	0	1
Pinch Point Temperature Difference in Main Heater, ΔTPP	10	400

The inlet and outlet pressures of the compressors are known (decision variables). Therefore, the pressure of all states can be calculated by using the value of FPD. Since the inlet temperatures of all compressors and turbines are also known, the inlet states of the compressors and turbines are concluded to be fixed. Considering the fact that the outlet specific entropy is equal to the inlet specific entropy in an isentropic process, the isentropic outlet of the compressors and turbines are also fixed. By using Eq. (1) and Eq. (2), the actual outlet states of the compressors and turbines are known.

$$\eta_{Comp.} = \frac{h_{out,is} - h_{in}}{h_{out} - h_{in}} \tag{1}$$

$$\eta_{Turb.} = \frac{h_{in} - h_{out}}{h_{in} - h_{out,is}} \tag{2}$$

The equation set for the recuperators is formed by applying the conservation of mass and energy. The set of equations is completed by using the definition of terminal temperature difference (ΔT_t). This procedure is slightly more complicated for the RRC configuration. In the RRC configuration, states 2 and 3 are fixed (compressor outlet). The mass flow rate going from point 2 to 3 can generally be any value depending on the mass flow fraction value, *f* (decision variable). Therefore, the minimum terminal temperature difference can occur at any end (that is, either at the hot end between 3 and 7, or at the cold end between 2 and 8) of the low temperature recuperator (LTR). As a first guess, the minimum terminal temperature difference is assumed to be at the cold end between points 2 and 8. The temperature of point 8 is calculated by Eq. (3), which makes point 8 as a fixed state. Equation (4) determines the enthalpy of point 7.

$$T_8 = T_2 + \Delta T_t \tag{3}$$

$$h_7 = h_8 + \left[(f)_{2-3} \times (h_3 - h_2) \right] \tag{4}$$

In order to test the first guess on the location of the minimum terminal temperature difference, the temperature of point 7 should be compared to the temperature of point 3. If the temperature difference between points 3 and 7 is less than the minimum terminal temperature difference, our first guess was wrong, and the minimum terminal temperature difference location is at point 3. Then, the same procedure can be employed to calculate the enthalpy of point 8 by using Eq. (5) and Eq. (6).

$$T_7 = T_3 + \Delta T_t \tag{5}$$

$$h_8 = h_7 - \left[(f)_{2-3} \times (h_3 - h_2) \right] \tag{6}$$

The last unknown state (point 4) can be fixed by using Eq. (7).

$$h_4 = h_3 + (h_6 - h_7) \tag{7}$$

The presented procedure is a simplified description of the actual algorithm that has been coded. The model is designed to be fully flexible so that it can easily be integrated with the optimizer tool (Genetic Algorithm) in a black box approach. When running the genetic algorithm, it is possible to encounter infeasible solutions. To avoid infeasible regions, several remedies and check points are contrived into the code.

Heat Recovery Considerations

As illustrated in Fig. 2, one key difference between S-CO_2 cycles for waste heat recovery and S-CO_2 cycles for solar and nuclear applications is the thermodynamic implication of how heat is added to the cycle. For those applications in which heat is added via a closed loop system, the conservation of energy implies that the amount of added heat to the cycle is equal to the generated heat in the heat source. This type of heat sources (solar receiver or nuclear reactor) is usually in the form of heat flux source in which the heat utilization is not constrained by temperature variation. Therefore, the heat input can be imposed onto the cycle modeling as a constant parameter. In contrast, for waste heat recovery applications, the heat input is applied through a heat exchange process between the working fluid (that is, carbon dioxide) and the hot waste gas. In such arrangements, the temperature of the waste gas stream decreases as the heat is transferred to the cycle's working fluid.

The products of mass flow rates and constant pressure specific heats (C_P) for the waste gas and the working fluid (CO_2) are not necessarily equal. Therefore, depending on the mass flow ratio of CO_2 and the waste gas, there can be a pinch point at either cold end or hot end of the recovery heat exchanger. The pinch point temperature difference is basically a very important factor that governs the amount of recovered heat and the CO2 mass flow rate in the cycle. In most common practices, the remaining thermal energy in the waste gas stream is ultimately discharged to the environment via a stack system. This implies that a portion of thermal energy in the waste gas stream is recovered in the heat exchanger and the rest is still wasted through the stack. In other words, the input heat to the cycle cannot be assumed as a constant parameter in waste heat recovery applications. On the contrary, the input heat is a nonlinear function of all decision variables in the power cycle, plus the pinch point temperature difference in the recovery heat exchanger. That is why the pinch point temperature difference in the recovery heat exchanger (main heat exchanger) is considered as a decision variable. This allows us to take into account the thermodynamic interactions of the heat recovery system and the power cycle, and ultimately find out the overall optimum design point. To this end, the following assumptions are made in the heat recovery system modeling:

(1) The waste gas is assumed to have the thermodynamic properties of air. It should be noted that because of the use of REFPROP, any arbitrary composition of the waste gas could have been considered in this calculation.

(2) The waste gas mass flow rate is fixed at 100 kg/s. Nevertheless, the results which are in the form of extensive properties can be linearly adjusted for other waste gas mass flow rates.

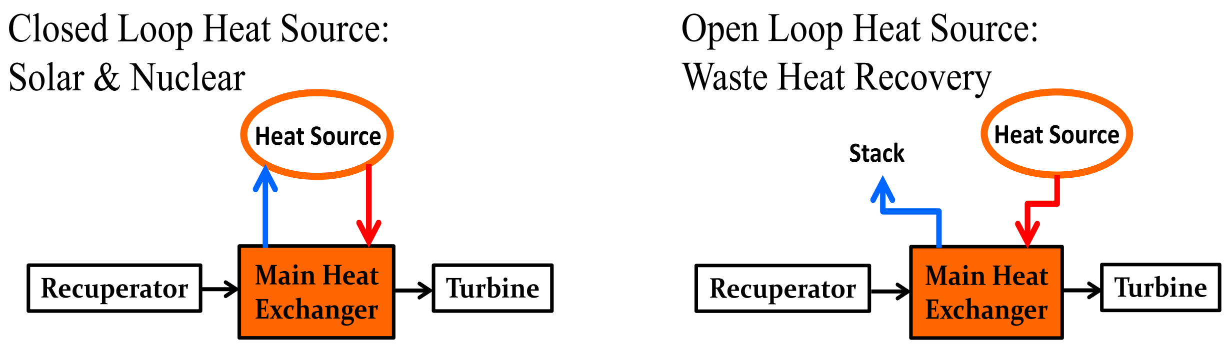


Figure 2: Heat Addition Process - Closed Loop vs. Open Loop

OPTIMIZATION APPROACH

The thermodynamic performance of energy systems is generally nonlinear, discontinuous; and has several local optima. In many cases, several decision variables exist which makes the optimization space multi-dimensional. As the number of decision variables increases, the interaction between subsystems and mathematical relations become tremendously complex; and traditional gradient based optimization algorithms become more tedious and in some cases even impractical. In contrast with gradient based optimization methods, Genetic Algorithm (GA) is a powerful evolutionary optimization method and can be competently adopted to address almost any optimization problem. The main advantage of GA is that it does not require analytical (or numerical) derivatives of the system's governing equations, but analyzes the system behavior. In other words, it treats the system as a black box. In the black box approach, the only interaction between the optimizer and the model is in the form of the decision variables and the corresponding values of the fitness (objective) function.

Brief Description of GA

The operation of Genetic Algorithm can be described in five steps:

1. A random population of individuals is generated. The identity of each individual is determined by a combination of values for the decision variables.

2. The fitness function for each individual is evaluated and individuals are sorted based on this criterion.

3. The fittest individuals, or in other words, individuals with greater values of fitness function are selected as parents to the next generation. For this purpose, fundamental genetic rules (marriage, mutation and talent preservation) are applied to this selected group and a new generation with same number of individuals as the previous generation is generated.

4. The new generation is again evaluated based on the fitness function. It is expected that the new generation that had healthy parents is better than the previous generation.

5. This process continues till health or fitness of the best individual does not change for several generations.

For more information on GA and its convergence criteria one may refer to Haupt (1998), Goldberg (1996), and Greenhalgh (2000).

Objective Function

In this study, optimization is merely based on thermodynamic analysis. The major focus in any power plant is to generate as much power as possible from available resources. Since the generated power is a product of input heat and cycle efficiency (Eq. 8), for a fixed input heat, this interpretation usually leads to maximization of the cycle efficiency.

$$P = \eta \times \dot{Q} \tag{8}$$

As explained in the modeling section, the amount of generated heat in the heat source is equal to the input heat to the cycle for applications such as solar and nuclear power. In other words, the heat input can be imposed onto the cycle modeling as a constant parameter. Therefore, maximizing the cycle efficiency guarantees the maximum power generation for such applications with the closed loop heat addition.

In contrast to traditional optimization approaches for solar or nuclear power applications in which the cycle efficiency is to be maximized, in waste heat recovery applications, the original optimization objective should be employed; that is, the objective is to maximize the power generation in the bottoming cycle. Therefore, the objective or fitness function in this paper is defined as the maximum cycle power.

RESULTS AND DISCUSSION

The presented results in this paper are based on 100 kg/s mass flow rate of the waste gas stream. As mentioned in the modeling section, some results are in the form of extensive properties; thus, they can be linearly adjusted for any other values of waste gas mass flow rates. And the results based on intensive properties are valid for any waste gas mass flow rates without further adjustments. Net power output for different values of the available (input) waste gas temperature is presented in Fig. 3 for each of the two cycle configurations: RC and RRC, and for two different optimization strategy: maximizing power output and maximizing efficiency. For waste heat recovery applications, the maximization of net power output is of practical interest, while maximization of efficiency is more of academic interest and is shown for comparison only.

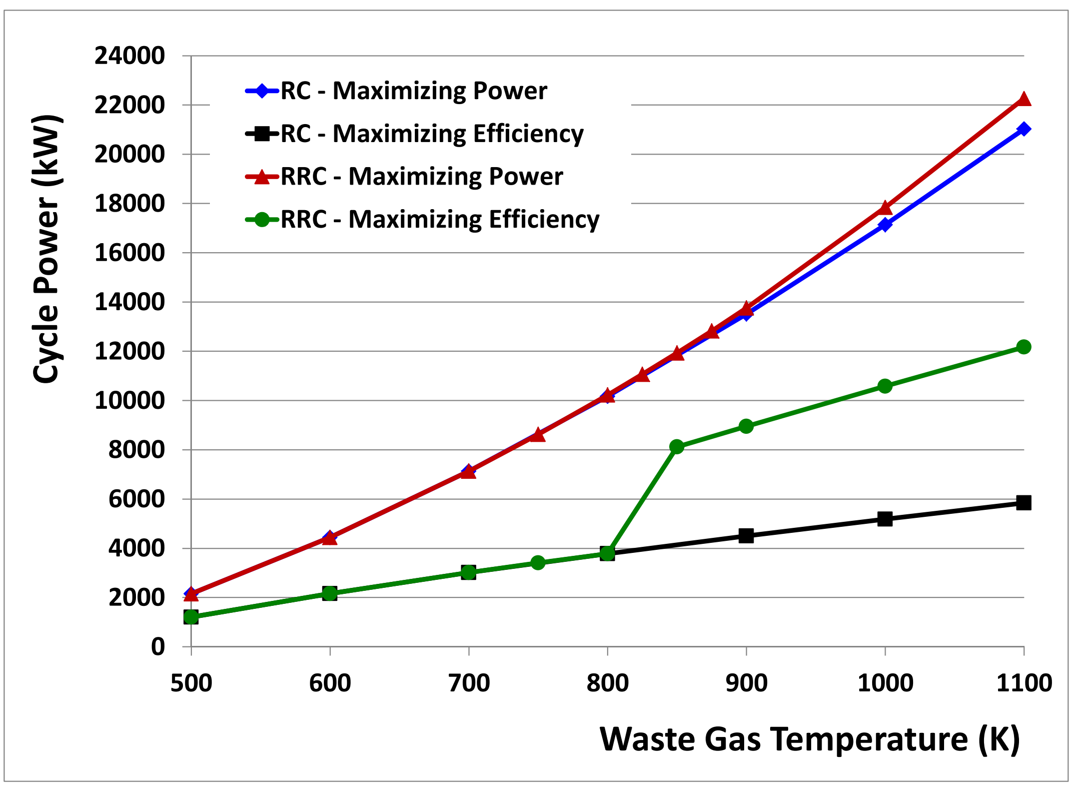


Figure 3: Net Power Output under Various Conditions

The results show that RRC configuration, while involving more complexity and more turbomachines, do not provide significantly higher power output. The RRC configuration does provide higher efficiency, especially for higher waste gas inlet temperature. Even then, the RRC configuration is transformed to the simple RC configuration for low waste gas inlet temperature (less than 700 to 800 K), suggesting that recompression does not provide any benefit, not even in efficiency, for low waste gas inlet temperature.

It should be noted that the pinch point temperature difference in the main heater plays an important role in the cycle performance. Optimization, as expected, always leads to the lowest bound of the main heater pinch point. While maximizing efficiency, the CO_2 mass flow rate gets automatically adjusted such that the temperature difference between waste gas and CO_2 remains constant so as to maximize the CO_2 mass flow rate for the same cycle efficiency. However, when the optimization strategy is to maximize the power, the pinch point always occurs at the low temperature end of recovery heat exchanger.

Another interesting point to note is that the pinch point for the low temperature recuperator can be in the middle of the heat exchanger and not at either end. However, the difference between the pinch point temperature difference and the minimum terminal temperature difference is quite small, typically a few tenths of 1 K, and hence the minimum terminal temperature difference is used as the decision variable in this work for the purpose of simplicity.

As maximizing net power output, rather than maximizing efficiency, is of practical interest, remaining results and discussion are presented only for the case of maximization of net power output. The mass flow rate of CO_2 needed to maximize the net power output for a given amount of waste gas mass flow rate and for various waste gas inlet available temperature is presented in Fig. 4.

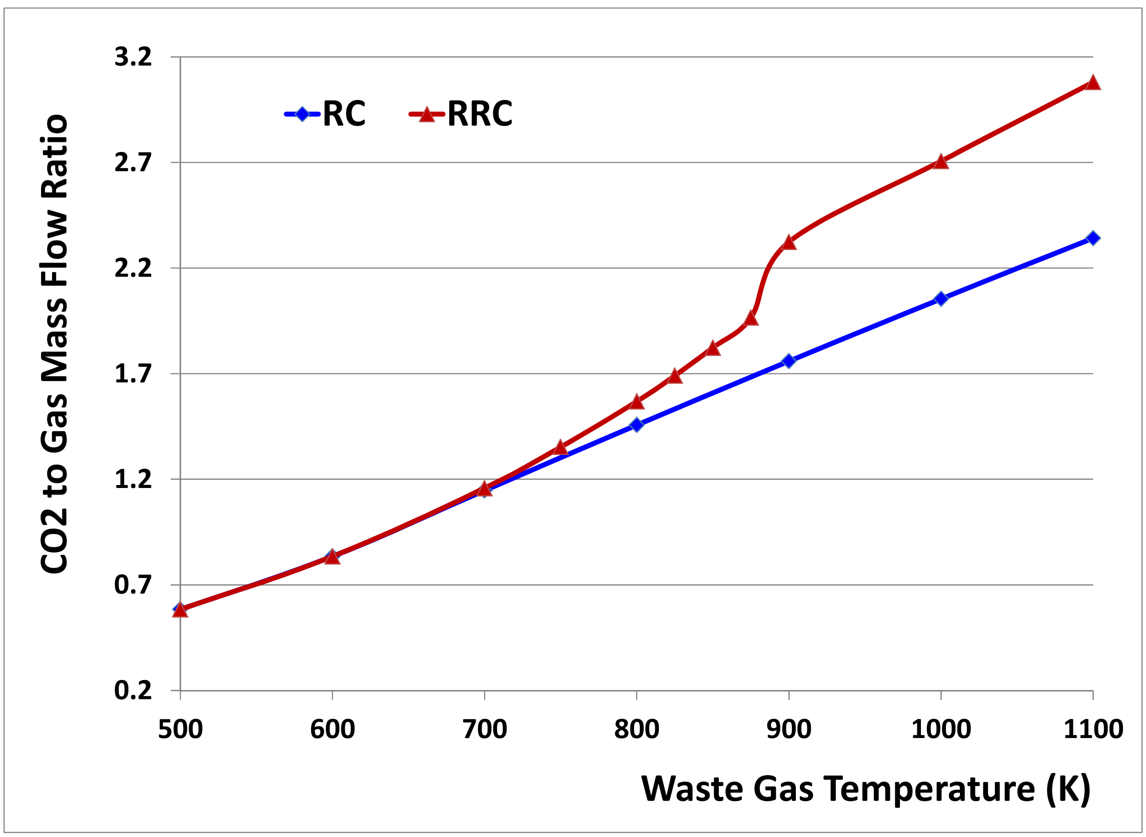


Figure 4: CO_2-to-Waste-Gas Mass Flow Ratio for Maximization of Net Power Output

For simple RC configuration, the CO_2 mass flow rate increases almost linearly with increasing waste gas temperature in order to take advantage of higher heat input, and hence higher power output, of the cycle. Optimization of RRC configuration leads to the RC configuration for waste gas temperature less than 700

K. For higher waste gas temperature, the optimal CO_2 mas flow rate in RRC configuration increases faster than that for RC configuration, until about waste gas temperature of 900 K, after which optimal CO_2 mass flow rate starts to increase linearly with waste gas temperature. Since RRC configuration does not provide any significant increase in net power output, higher CO_2 mass flow rate puts the RRC configuration at a disadvantage. Fig. 5 presents the optimum turbine inlet temperatures and main heater inlet temperatures as functions of the waste gas temperature. The optimum turbine inlet temperature is not significantly affected by the configuration, although RRC configuration requires slightly lower turbine inlet temperature. It should be noted that the optimal turbine inlet temperature increases almost linearly with the waste gas temperature above 600 K, but at a much slower rate. For example, when waste gas temperature increases from 700 K to 1100 K, the TIT increases from 600K to about 750K for the RC configuration. The low temperature end of the main heater also increases with increasing waste gas temperature, which also indicates the increase in the stack temperature.

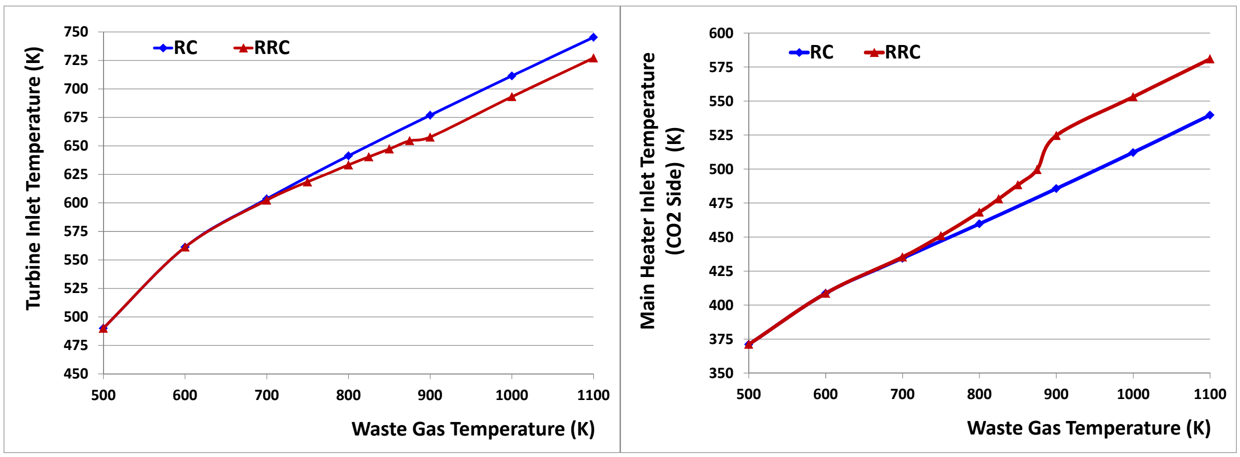


Figure 5: Turbine Inlet Temperature and CO_2-side Inlet Temperature for Main Heater

CONCLUSIONS

Maximization of net power output is of paramount practical interest rather than the maximization of thermodynamic efficiency for the application of waste heat recovery. In such applications, very efficient cycle designs may suffer from low heat recovery and low power generation. That is why in addition to the cycle design parameters; the heat exchange process between the waste gas and the working fluid needs to be considered in order to arrive at the proper optimal solutions. In spite of increased complexity, the RRC configuration does not provide any appreciable benefit as compared to the RC configuration in terms of net power output.

NOMENCLATURE

S-CO_2 = Supercritical Carbon Dioxide
CCGT = Combined Cycle Gas Turbine
INL = Idaho National Lab
SNL = Sandia National Lab
NIST = National Institute of Standards and Technology
RC = Recuperated Cycle
RRC = Recompression Recuperated Cycle
h = Enthalpy, kJ/kg
η = Efficiency
FPD = Fractional Pressure Drop
f = Mass Flow Fraction
ΔT_{pp} = Pinch Point Temperature Difference, K
ΔT_t = Recuperator Terminal Temperature Difference, K
LTR = Low Temperature Recuperator
GA = Genetic Algorithm
$\dot{Q}$ = Input Heat, kW
P = Power, kW

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