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Turbocompound Reheat Gas Turbine Combined Cycle

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ABSTRACT

This paper discusses a new power generation cycle based on the fundamental thermodynamic concepts of constant volume combustion and reheat. The turbo-compound reheat gas turbine combined cycle (TC-RHT GTCC) comprises three pieces of rotating equipment: A turbo-compressor and two prime movers, i.e., a reciprocating gas engine and an industrial (heavy duty) gas turbine. Ideally, the cycle is proposed as the foundation of a customized power plant design of a given size and performance by combining different prime movers with new "from the blank sheet" designs. Nevertheless, a compact power plant based on the TC-RHT cycle can also be constructed by combining off-the-shelf equipment with modifications for immediate implementation. The paper describes the underlying thermodynamic principles, representative cycle calculations and value proposition as well as requisite modifications to the existing hardware. The operational philosophy governing plant start-up, shut-down and loading is described in detail. Also included in the paper is a 110 MW reference power block concept with 57+% net efficiency. The concept has been developed using a pre-engineered standard block approach and is amenable to simple “module-by-module” construction including easy shipment of individual components.

INTRODUCTION

Brief History

Internal combustion engines can be classified into two major categories based on the heat addition portion of their respective thermodynamic cycles: "constant volume" and "constant pressure" heat addition engines (cycles) [1]. Either process is an idealized conceptualization of the actual fuel-air combustion that takes place inside the actual engine. In particular,

- **Constant volume heat addition** is closely approximated by the combustion of fuel-air mixture within the cylinders of a reciprocating or piston engine, e.g., a car or truck engine.

- **Constant pressure heat addition** is closely approximated by the combustion of fuel-air mixture inside the combustor of a gas turbine.

Thermodynamic cycle analysis, whether using the idealized air-standard approach or "real fluid" approach, clearly demonstrates the superiority of constant volume heat addition or combustion process in terms of cycle thermal efficiency [1, 2]. The reason for that, in layman's terms, is that constant volume combustion is essentially an "explosion" leading to *simultaneous* increase of temperature and pressure of the working fluid. Everything else being equal, this leads to a better thermal efficiency because part of the compression is achieved within the heat addition part of the cycle and, for the same amount of heat addition, leads to higher net cycle power output (less compression work).

In terms of practical applications, this advantage is implicit in the efficiencies of modern gas fired reciprocating engine gen-sets pushing nearly 50% (cf. around 40% for modern heavy-duty industrial gas turbines or 45% for smaller aeroderivative units with very high cycle pressure ratios).

The idea of constant volume combustion (CVC) within the context of gas turbines goes back to Holzwarth's explosion turbine in the early years of the 20th century [1, 2]. Due to its intermittent nature, strictly speaking a "closed system" process (envision the "explosion" within the engine cylinder in the space between the piston and cylinder head), CVC is in direct conflict with the steady flow nature of the turbine expansion, i.e., an "open system" or steady-state steady-flow (SSSF) process. As such, CVC dropped off the evolutionary trajectory of the gas turbine technology for land-based electric power generation.

Nevertheless, the idea has always been around for more than a half century within the context of aircraft propulsion. The specific version of the *quasi* CVC in this context is the "pulse detonation combustion". The engine comprising the pulse detonation combustor is known as a "pulse detonation engine" (PDE) [1, 2]. As the name suggests, the concept involves creation of a detonation wave within a semi-closed tube filled with fuel-air mixture. The resulting wave simultaneously compresses and heats the mixture, which is then discharged into the axial turbine. The same dichotomy mentioned earlier, namely “steady flow open system” and “intermittent flow closed system”, results in mechanical design difficulties, which so far presented the transition of PDE or similar CVC concepts into viable commercial products.

While CVC as an integral part of a gas turbine engine (or cycle) never made that transition, combination of the two types of internal combustion processes and respective engines as a “compound” system has found some commercial success [1]. Two of the earliest examples of such a “turbocompound” engine are Allison’s V-1710-127(E27) and Napier Nomad aircraft engines. Despite the tremendous fuel efficiency offered by turbocompound engines (Nomad is still the most efficient internal combustion engine flown with less than 0.35 lb/bhp-hr in flight delivering about 3,000 bhp), aircraft industry bypassed them in favor of rapidly emerging gas turbines. Many factors played into the decision; e.g., high weight/thrust ratio, high cost, low reliability and cheap fuel (back then, that is). Eventually, the technology found a permanent home in land-based propulsion applications, e.g., turbocompound diesel engines in Scania (former Saab) trucks.

(Note that a turbocompound engine is fundamentally different than a “turbocharged” engine. In the latter, the exhaust gas driven turbine-*cum*-compressor unit is simply an engine accessory to compress the air before it enters the engine cylinders to increase the working fluid mass for increased shaft power. In the turbocompound arrangement, the gas turbine is an “equal partner” with its reciprocating/piston counterpart in terms of contributing to the total shaft power generation.)

Recently, a turbocompound gas turbine combined cycle concept has been proposed [3,4]. The concept is named alternatively as

1. Engine Turbo-Compound System (ETCS), or
2. Engine Reheat Gas Turbine (ERGT)

The ETCS/ERGT concept involves gas turbine exhaust gas heat recovery in a heat recovery steam generator (HRSG) for additional power generation in a steam turbine (ST). As such, it is a *combined cycle* system. In particular, ETCS is a true turbocompound concept where the two distinct internal combustion engines are separate entities in their own right as shown in Figure 1.

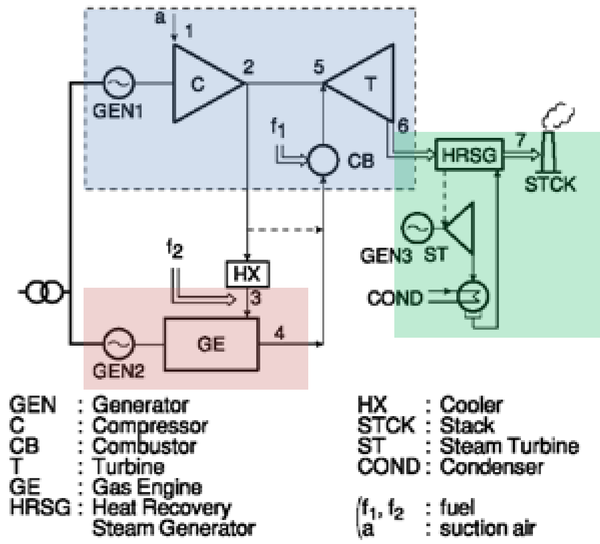

Figure 1 Engine Turbo-Compound System (ETCS) from Ref. [3]

The ETCS system as shown in Figure 1 works as follows:

1. A modified gas turbine GT; in particular
   a. A portion of the air from the compressor (C) discharge sent to the **gas engine** (GE[1]) after first having been cooled in a heat exchanger (HX)
   b. Exhaust gas from the GE piped back into the GT at the combustor (CB) inlet after having been mixed with compressed air bypassing the GE
2. A modified gas engine (GE) with no turbocharger (the turbocharger's task is assumed by the compressor of the GT) so that its exhaust gas is at a pressure high enough to satisfy the turbine (T) requirements.

The ETCS is an integrated system, which – at least on paper, that is – can be developed by combining two existing "off-the-shelf" units, i.e., a GT and a GE, with extensive modifications and additional piping and heat exchangers. (The ERGT is more like an *explanatory* conceptual model that illustrates the thermodynamics underlying the ETCS system. As such, it does not require further attention here.) The presence of double combustion in Figure 1, first inside the gas engine cylinders (fuel flow f_2) and then in the combustor of the gas turbine (fuel flow f_1), is sufficiently clear to illustrate the "reheat" concept implicit in ETCS.

The ETCS performance reported in Ref. [3] is summarized in the table below.

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[1] Hereafter the term gas engine (GE) is used for the gas-fired reciprocating (piston-cylinder) engine gen-set.

Table 1 ETCS performance from Ref. [3]

	ETCS (1)	ETCS (2)
Type and Number of Gas Turbine and Gas Engine	6MW-Class GT×1 (TIT 1150°C) Gas Engine ×2 (900°C Exhaust)	150MW-Class GT×1 (TIT 1350°C) Gas Engine ×32 (900°C Exhaust)
Power Output		
Gas Turbine Gas Engine Steam Turbine	6,700 kW 11,500 kW 4,000 kW	160,400 kW 200,100 kW 99,100 kW
ETCS	22,200 kW	459,600 kW
Thermal Efficiency (Gross, LHV Base)		
ETCS	49.8 % LHV	56.7 % LHV

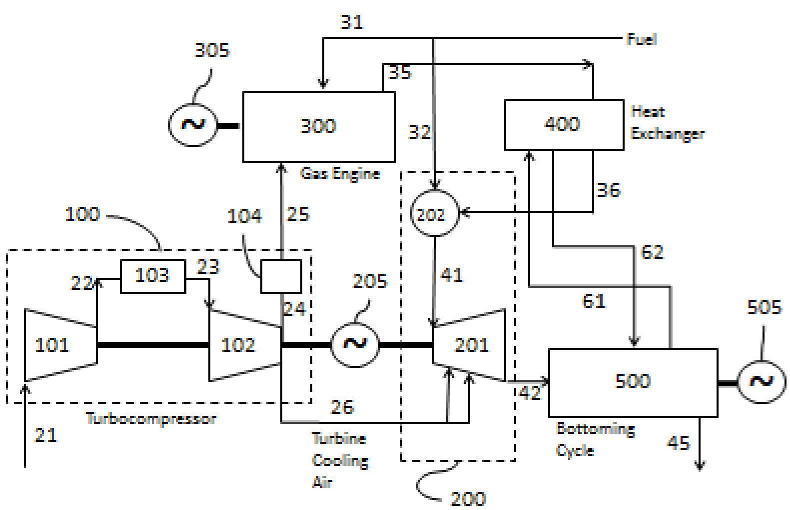

Figure 2 Detailed system diagram of the current TC-RHT GTCC concept

Current Concept[2]

The current concept is a **turbocompound reheat** (TC-RHT) gas turbine combined cycle (GTCC) comprising **three** pieces of major equipment (a detailed system diagram is shown in Figure 2):

1. An intercooled (most probably, integrally geared) centrifugal turbocompressor (TC) with an aftercooler

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[2] US Patent Application 14/304,089 “Turbo-Compound Reheat Combined Cycle Power Generation” filed on 06/13/2014

2. Advanced gas engine with the turbocharger removed
3. An industrial (heavy duty) gas turbine with the compressor section removed

Air compressed in the turbocompressor (100) is sent to the gas engine (300) intake after being cooled in an aftercooler (104) to a suitable temperature (typically, ~120°F).

Since the air is already compressed, there is no need for the engine turbocharger. Some of the compressed air is sent to the turbine (201) for component cooling (not shown is the "booster" compressor).

The gas engine (300) burns natural gas fuel to generate power. The exhaust of the gas engine is sent to a heat exchanger (400) for adding heat to the feed water of the steam Rankine bottoming cycle 500.

The exhaust gas leaving the heat exchanger 400 is sent to the GT combustor (202), which burns natural gas fuel to generate hot gas for expansion in the turbine section (201) for power generation.

The turbine can be E class (TIT of 2,175°F) or early F class (TIT is 2,475°F) to keep development and/or modification costs down. In any event, reduced O_2 content in the GE exhaust gas (about 9-10% by volume) puts a cap on TIT. The exhaust gas from the turbine (a three-stage design with a nominal pressure ratio (PR) of 13) varies between 1,050°F and 1,200°F (maximum allowable) depending on the TIT and PR. It is utilized in the bottoming cycle 500 to generate additional power.

The bottoming cycle (500) herein is a Rankine steam cycle comprising a two-pressure reheat or no-reheat **heat recovery steam generator** (HRSG), a **steam turbine** (ST) with

- Water-cooled, once-through, open-loop condenser (ISO conditions) or
- Dry air-cooled condenser (hot day conditions)

and the **balance of plant** (BOP) including myriad pipes, valves, pumps and heat exchangers. Superheated steam generated in the HRSG is expanded in a ST for additional power generation.

A temperature-entropy (T-s) diagram of the system in Figure 2 is shown in Figure 3.

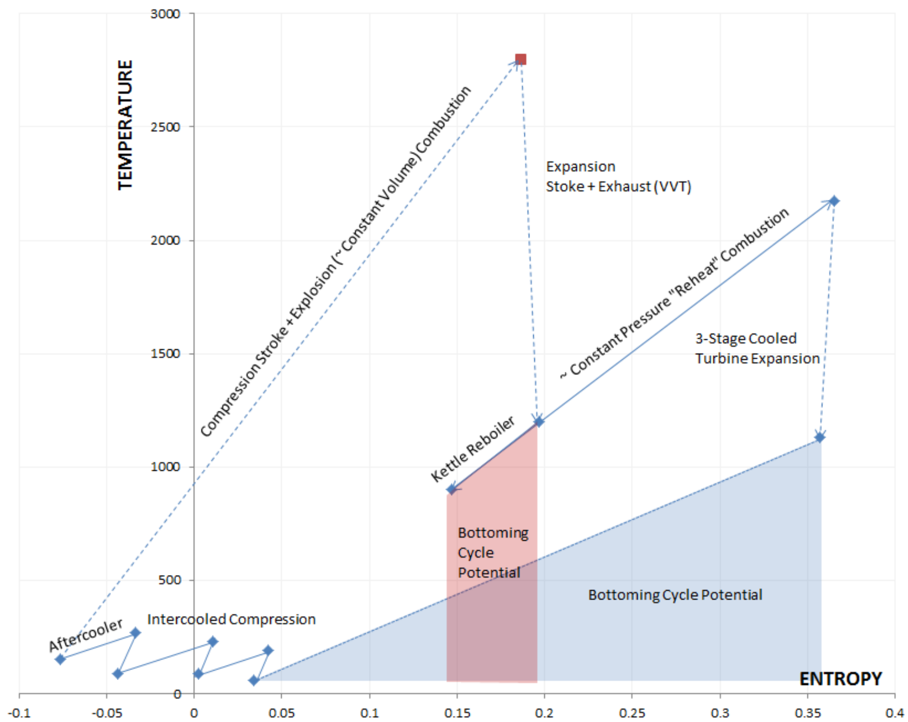

Figure 3 Conceptual T-s diagram of the current TC-RHT GTCC concept (three-stage compressor with two intercoolers).

SYSTEM PARTICULARS

Modified Gas Engine Operation

A base (original) engine value of 5.5 is assumed for the calculations herein. Thus, the engine intake air is at 116 psia, which otherwise would have been achieved in a compressor driven by an exhaust gas turbine. (In modern gas engines, two-stage turbocharging offers pressure ratio capability up to 10 and turbocharger efficiency up to 75% – see Figure 4.)

Modification of the engine to fit into the turbocompound system of the concept, increases the supercharging PR by about 25-30% and allows for much higher air mass to be "squeezed" into the cylinders (also by ~25-30%), i.e.

$$\dot{m} = \dot{m}_0 \cdot \frac{T_{in,0}}{T_{in}} \cdot \frac{P_{in}}{P_{in,0}}$$

$$\dot{m} = \dot{m}_0 \cdot \frac{T_{in,0}}{T_{in}} \cdot \frac{PR_{schg}}{PR_{schg,0}}$$

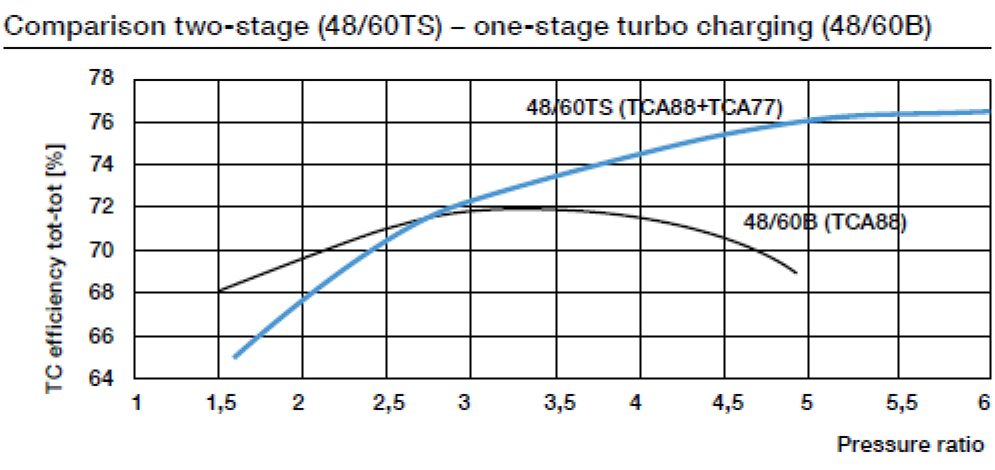

Figure 4 GE Jenbacher turbocharger (TC) efficiency as a function of compression pressure ratio [5].

In its introductory phase, the aim of the TC-RHT GTCC concept is to enable a highly efficient and cost-effective power plant system that can be built using "off-the-shelf" components without excessive modification and obviating the need for exotic/expensive materials for the balance of plant such alloy pipes. In order to achieve this goal there are two key requirements:

i. Engine exhaust pressure to be high enough to allow a reasonably high gas turbine PR (about 9)

ii. Engine exhaust gas temperature to be low enough (~ 1,300°F) to keep the cost of piping and heat exchanger tubing downstream at a reasonable level and ensure gas engine exhaust valve life

The unmodified, supercharged, spark-ignition gas engine operation is represented conceptually by the standard p-V diagram shown below.

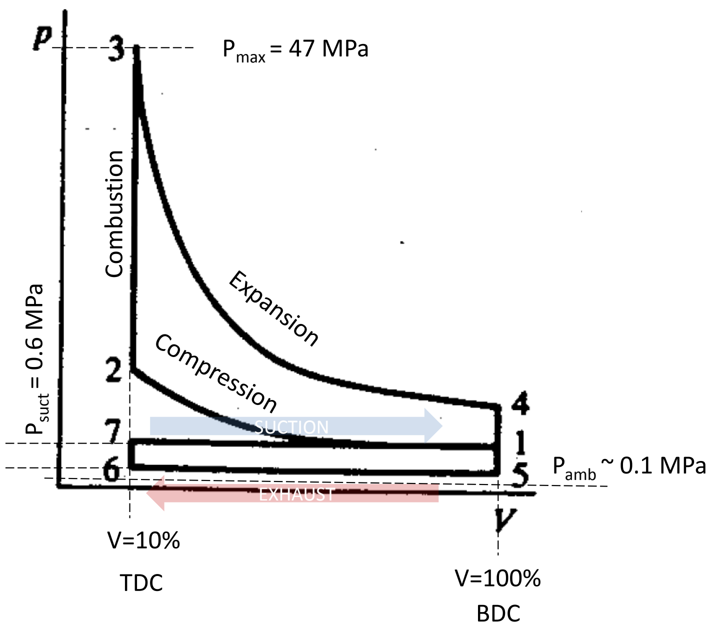

Figure 5 Pressure-volume diagram of spark-ignition gas engine with compression ratio, CR = V_1 / V_2. The gas engine cycle processes are as follows:

1. **Supercharged air intake (6-7-1)**
2. **Compression stroke (1-2)**
3. **Constant volume combustion (2-3, $v_2 = v_3$, $u_2 = u_3$)**
4. **Expansion stroke (3-4)**
5. **Exhaust (4-5-6)**

This is an idealized cycle with valve events taking place at top and bottom dead centers (TDC and BDC, respectively) and no change in cylinder volume as pressure difference across open valves drop to zero. Inlet and exhaust pressures are constant with negligible velocity effects. The modified (idealized) gas engine operation is shown on the p-V diagram below.

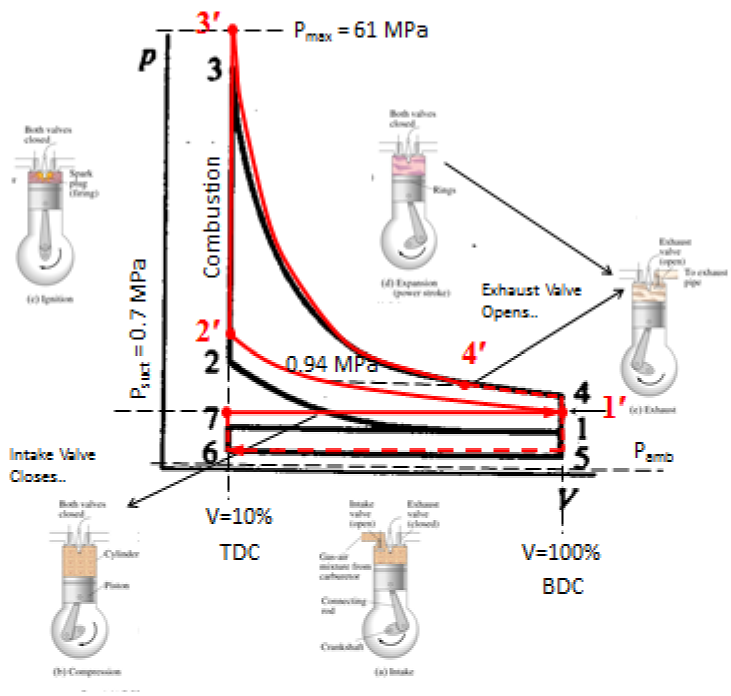

Figure 6 Pressure-volume diagram of modified spark-ignition gas engine. The modified gas engine cycle processes are as follows:

1. **Air intake (7-1′) – supercharging outside the engine**
2. **Compression stroke (1′-2′)**
3. **Constant volume combustion (2′-3′, $v_2 = v_3$, $u_2 = u_3$)**
4. **Expansion stroke (3′-4′)**
5. **Exhaust (4′-4-7)**

The key to the modified operation is the control of the timing of intake valve closing and exhaust valve opening. **Variable valve timing** (VVT) allows the engine controls to open the exhaust valve at a point to achieve the desired gas pressure. However, under normal engine operating conditions, the gas temperature at that point (140-160 psia) would be very high, i.e., around 1,700°F (925°C). Combined with the high suction pressure described earlier, this requires careful evaluation of fuel injection and supercharging to keep the cylinder maximum pressure and temperature at levels to ensure that the gas temperature at the exhaust valve opening is at the desired level. It is unavoidable that such a modification will severely reduce the GE efficiency.

Heat Recovery from Gas Engine Exhaust Gas

The exhaust gas from the gas engine (in reality, multiple gas engines) 35 is sent to the heat exchanger (evaporator) 400 (please refer to Figure 2).

Hot fluid is exhaust gas from the gas engines 35 (at 1,300°F), which is cooled to 900°F.

"Cold" gas 36 is sent to the combustor 202 of the gas turbine 200.

Steam generated in the evaporator 62 is sent back to the bottoming cycle 500. The optimal steam pressure is 1,650 to 1,850 psia (i.e., high pressure steam).

One possibility for the heat exchanger 400 is a **kettle evaporator**, which is a shell-and-tube type heat exchanger. Kettle evaporators are used to boil a liquid in a manner very similar to a teakettle (hence the moniker *kettle*), i.e., with a large open boiling surface, utilizing the heat from a hot fluid flowing in U tubes immersed in the boiling liquid (see Figure 7). In this case the boiling liquid is feedwater 61 from the bottoming cycle 500.

Ultimately, the most appropriate type of evaporator should be determined after a careful evaluation of process conditions in a cost-performance trade-off study.

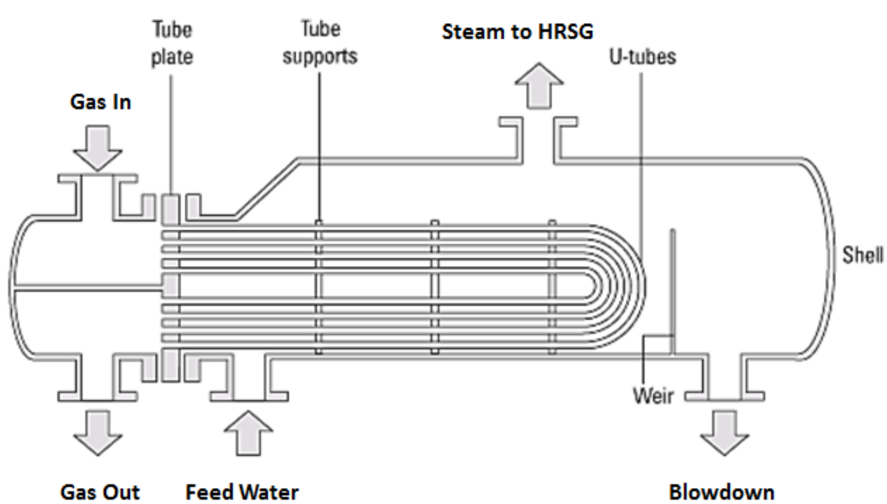

Figure 7 Kettle type reboiler (shell-and-tube heat exchanger) for steam generation

Combustion and Emissions

In the combustor 202, the second (i.e., *reheat*) combustion takes place. This combustor is envisioned to be the same component as in the original GT. Due to different airflow (about 30% less than in the original GT), oxygen content (10% vis-à-vis 21%), inlet air/gas temperature (900°F vis-à-vis ~680-700°F) and pressure (about 25% less), it is fully expected that the OEM would make some adjustments to ensure stability and emissions. The exact nature of such adjustments and/or design modifications cannot be foreseen at this time.

Nevertheless, some predictions can be made about the magnitude of emissions from the TC-RHT GTCC. Combustion with vitiated gas in the GT combustor is akin to the well-known EGR (**Exhaust Gas Recirculation**) concept. Since the O_2 content of the incoming hot gas is considerably lower than that of normal air, less oxygen is available for the NOx formation.

Assuming that the GT combustion results in 10 ppmvd (15% O_2) NOx production (instead of typical 25 ppmvd), with the addition of ~250-300 ppmvd (5% O_2) NOx produced in the reciprocating engine cylinder, the equivalent stack NOx emission would be around 75 ppmvd (15% O_2). This, of course, assumes that the modified gas engine cycle does *not* impact its nominal NOx emission rating. Actual gas engine emissions should come from the OEM.

In any event, it is reasonably certain that the final stack NOx emission for the TC-RHT GTCC concept would be around 75 ppm or less, which is about 25% lower than the original gas engine NOx emission rating on the same basis. Further reduction to single digit NOx emission can be readily achieved with an SCR in the HRSG.

Carbon dioxide emission is calculated as 24pps, which translates into ~760 lbs/MWh and is in line with advanced F class GTCC. Furthermore, due to its (i) higher CO_2 and (ii) lower O_2 content on a molar basis, about 7%(v) and 5%(v), respectively, TC-RHT GTCC is more advantageous for post-combustion carbon capture in an amine-based absorption-stripper process vis-à-vis a modern GTCC.

Bottoming Cycle

The bottoming cycle design specification is as follows:

- Two-pressure, reheat HRSG and steam turbine (nominal 50 MW)
- 1,650 psia / 1,000°F steam cycle
- 15°F pinch point delta T evaporator design
- Water-cooled (deaerating) condenser (1.2 in. Hg) at ISO, or
- Dry air-cooled condenser (3.0 in. Hg) at 85°F and 40% humidity
- HP feed water supplied to the kettle reboiler (KRB) at ~1,800 psia
- Saturated HP steam returned to the HP evaporator
- IP feed water used to heat gas fuel to 365°F

For the two-pressure reheat bottoming cycle, General Electric's **A Series** steam turbine is a good fit (up to 250 MW for small fossil applications, maximum steam conditions 2,520 psig/1,050°F/1,050°F). This product features a compact design using a separate HP casing along with a single-casing, intermediate pressure/low pressure section. For the single-flow LP section, several last stage bucket (LSB) sizes are available for the best fit to a particular application.

For the two-pressure, **no reheat** steam bottoming cycle, General Electric's **SC/SAC Series** condensing unit offers an *off-the-shelf* solution (maximum steam conditions 1,800 psig/1,000°F).

It should be noted that no attempt has been made to integrate the GE heat rejection via its cooling water and the compressor intercooler and/or aftercooler heat rejection into the bottoming cycle (e.g., feed water heating) in an optimal manner. In that sense, calculated performances are somewhat conservative.

Shaft Arrangement

As shown in Figure 2, turbocompressor 100, generator 205 and turbine 201 are on the same shaft. This single-shaft configuration is similar to the GT/GEN/FGC power train configuration originally developed by General Electric in 1990s for steel mill blast furnace gas (BFG) applications. In this case, a Frame 9E gas turbine is connected to the fuel gas compressor (FGC), which is a two-stage intercooled centrifugal unit, with the 9A5 generator (GEN) between the two (see Figure 8). This configuration has been successfully operational in many steel mills in Europe and China (most recently in Wuhan and Handan).

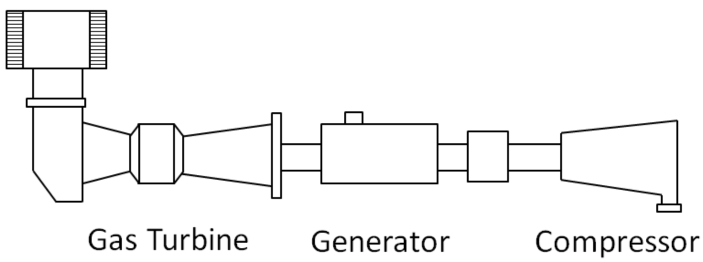

Figure 8 Single-shaft BFG firing gas turbine power train configuration (e.g., General Electric 9E in Wuhan and Handan) [6]

The optimal configuration envisioned for the introductory TC-RHT GTCC plant is similar to that as shown in Figure 9

Figure 9 Single-shaft TC-RHT GTCC power train configuration

Turbocompressor

The performance of the system is highly dependent on the turbocompressor configuration and technology. The particular unit in the system diagram of Figure 2 is a two-stage compressor with a single intercooler between the stages. There are other possible configurations; e.g., a three-stage compressor with two intercoolers or an axial compressor (no intercooling) as shown in Figure 10. (The term “stage” is used herein in lieu of the more accurate term “casing”. There might be several stages (each stage comprising one stationary and one rotating row of blades, i.e., *stators* and *rotors*, respectively) within a casing.)

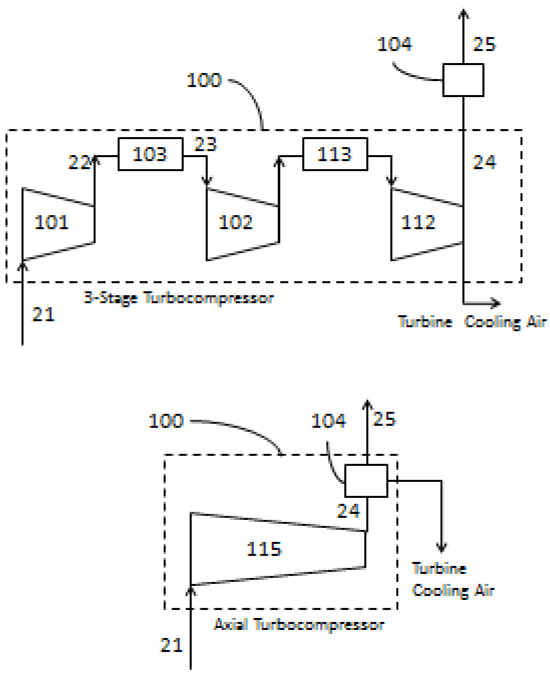

Figure 10 Different turbocompressor configurations

Axial compressors have typically higher efficiencies and they are well suited to high flow applications up to around 100 psia (note that multistage axial compressors in aircraft or land-based industrial gas turbines are designed for much higher pressure ratios and discharge pressures). Beyond that limit discharge temperatures become too high requiring specialized seals. Axial compressors have typically low stage PRs (about 1.1 to 1.2) so they require a large number of stages (about 10) to accomplish the desired pressure rise. While they are smaller than the centrifugal compressors and more efficient, they also cost more. From an operability perspective, axial compressors have a narrow band of stable operation between their operating and surge lines.

Centrifugal compressors are more suitable to high-pressure, low-flow applications in multi-stage intercooled configurations. They have typically lower efficiencies than axial compressors on a stage-by-stage basis but the overall efficiency of a multistage intercooled unit is significantly higher than that of a multistage axial unit for the same pressure ratio and inlet conditions. (This is so because the overall compressor train performance with intercooling between stages approaches the isothermal compression ideal with increasing stages.) Centrifugal compressors have a broader band of stable operation between their operating and surge lines. Operational flexibility of motor-driven units is enhanced by a VFD. In fixed-speed units (electric motor or prime mover driven), at low loads recirculation might be required to prevent surge.

OPERABILITY

Key operational factors must be defined by the compressor 100 and GT 200 OEMs for acceptable combinations of compressor 100 airflow and GT combustor 202 airflow, inlet temperature and oxygen concentration. A minimum combustor 202 temperature limitation to ensure maximum flame stability could, for example, require inclusion of a bypass from the inlet of the intercooler 24 to the GE exhaust 35 during start-up or during plant turndown when some GE units are not operating.

The basic startup sequence is described below:

- GT and ST units on turning gear
- Load Commutating Inverter (LCI) starts the GT and the compressor (generator acts as motor)
- Discharge of the compressors is bypassed from aftercooler104 inlet 24 and/or discharge 25 to GE exhaust 35 by a GE inlet pressure regulating bypass
- When GT 200 output torque is sufficient to drive the compressor 100, the LCI disengages
- When the generator 205 reaches full speed, it is synchronized and electrical generation commences
- Gas engines are started individually by their own compressed air starter systems and their generators are synchronized upon achieving full speed
- As gas engines start, the GE inlet pressure regulating bypass progressively closes as airflow is taken by the GE units
- When HRSG steam is ready, ST starts rolling and is synchronized when full speed is reached in the Figure 2 configuration, or it engages though the SSS clutch in the single shaft configuration shown in Figure 9.

A key area of operational interest will be the acceptable range for throttling the compressor inlet flow to allow staging of the GE units to meet swings in load demand. This may provide significant capability for the TC-RHT GTCC to adapt to the demands of a system with significant renewable generation. As previously discussed, selection of the elements of the compressor 100 should include consideration of the operational range of the available compressor configurations. Due to the significance of the GT combustor 202 oxygen concentration, the closer the system airflow can be matched to the required number of operating GE units for the system electrical demand, the closer the emissions will match the nominal full load values.

PERFORMANCE AND COST

The TC-RHT GTCC concept can be evaluated from two perspectives:

1. A novel power generation cycle with a "blank sheet" design for high efficiency at reasonably low TIT
2. A power plant configuration combining existing, "off-the-shelf" technology in a novel manner for high efficiency at lower ratings

The logical sequence of development would start with the second approach to prove the concept in the field. Once that is achieved, a new product development phase can be started so that the gas engine can be designed for the requisite high-pressure, high-temperature exhaust operation with minimal sacrifice in thermal efficiency.

The advantage of a "small" TC-RHT GTCC (around 110 MW ISO base load) is especially striking for certain markets, i.e.

1. Small grid unable to accommodate large units
2. High natural gas price (more than $10-$12 per million BTU)
3. Ability to use diesel fuel (gas supply is subject to uncertainty)
4. High ambient temperature and humidity
5. High altitude
6. Combination of the above

With that in mind, the following TC-RHT GTCC configuration is proposed:

1. A 5x1x1x1 configuration with
    a. 5 gas engines
    b. 1 F-Class gas turbine
    c. 1 two-pressure, no reheat HRSG
    d. 1 steam turbine
2. Site conditions
    a. ISO
    b. Hot (85°F, 40% RH[3])
    c. Hot and High Altitude (95°F, 30% RH, 6,150 ft elevation[4])

[3] Similar to Kinyerezi, Tanzania

[4] Similar to Plains End (I and II), located in Arvada, CO

3. Conceptual plant layout as shown in Figure 11

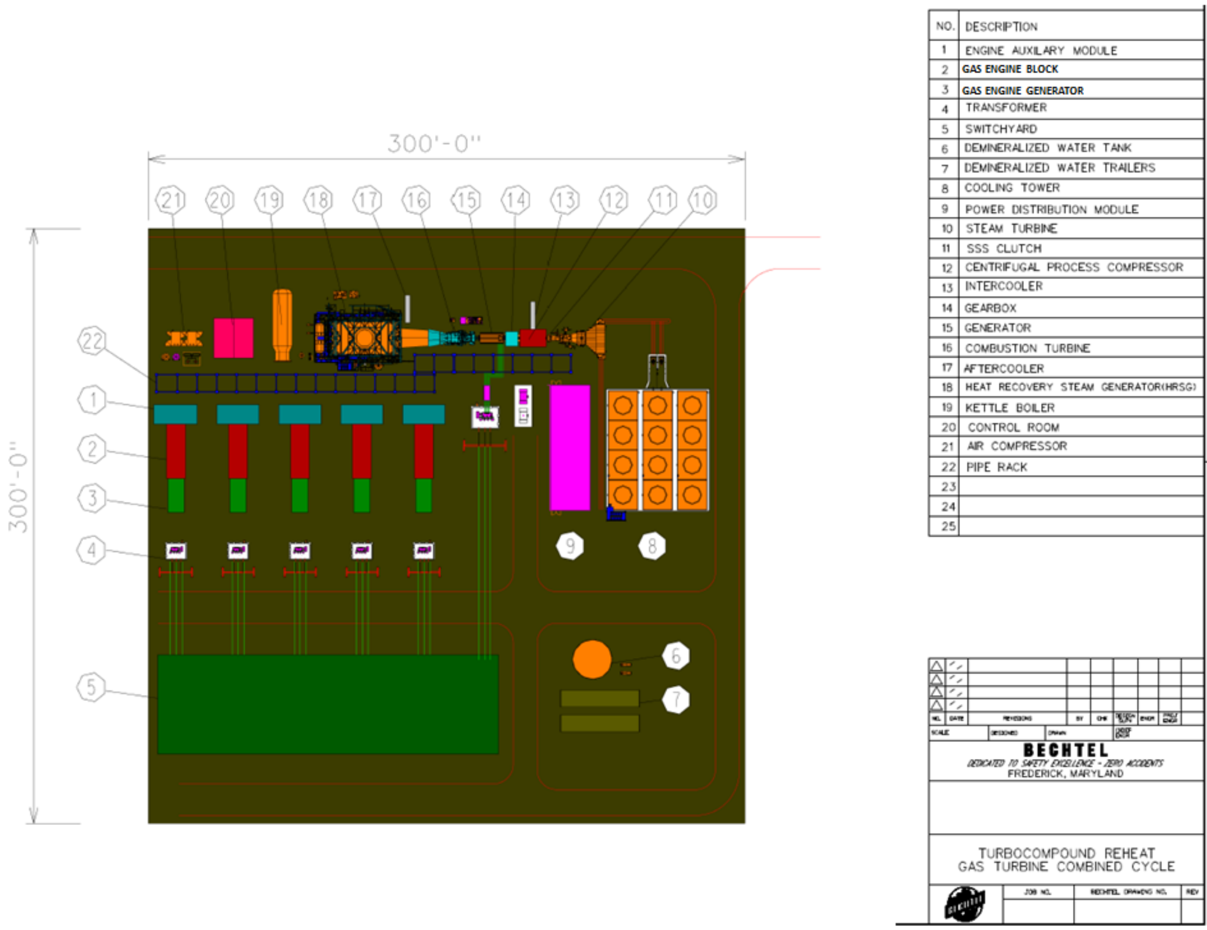

Figure 11 Conceptual TC-RHT GTCC Plant Layout – Plane View

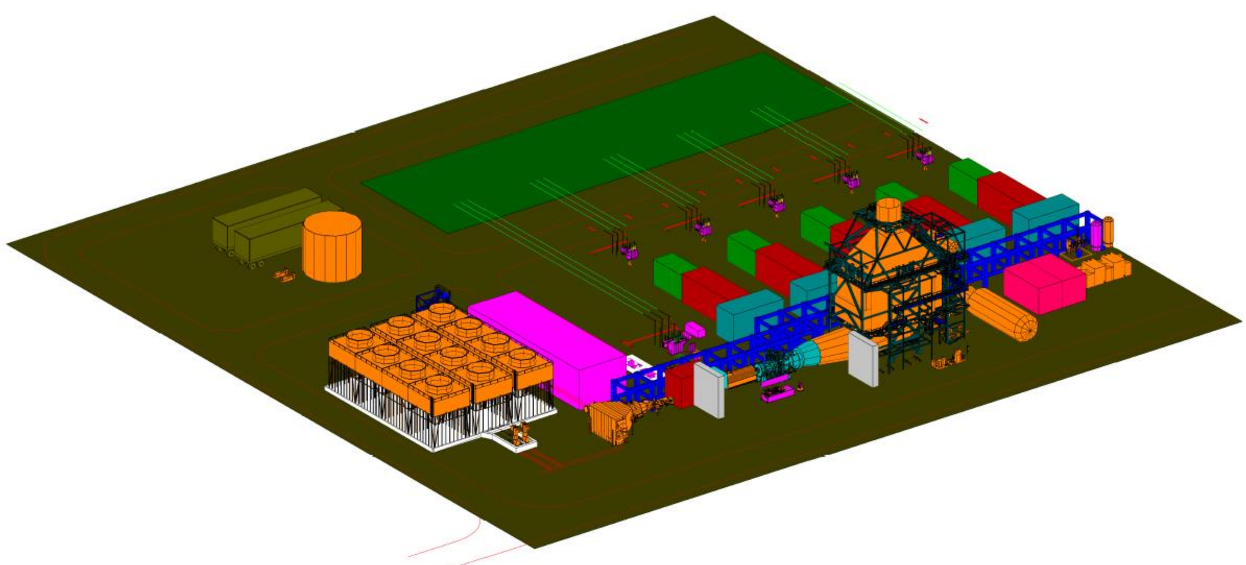

Figure 12 Conceptual TC-RHT GTCC Plant Layout – Isometric View

Compressor (including intercooling and after-cooling coils), gas turbine (including combustor), kettle reboiler and compressor cooling water heat rejection system performance, design, sizing and costing calculations are done in Thermoflow Inc.'s **Thermoflex** software (including the **PEACE** add-in for costing) [7]. Thermoflow Inc.'s **GTPRO** software is used for the performance, design, sizing and costing calculations of the bottoming cycle (including the **PEACE** add-in for costing) [7].

Table 2 Base and modified gas engine performance.

	BASE	ISO	Hot	Hot & High
Electric Power, kW	9,525	9,050	9,000	8,425
Efficiency	48.2%	40.2%	39.9%	37.4%
Heat Rate, Btu/kWh	7,080	8,490	8,545	9,115
Compression Ratio	13.5	13.5	13.5	13.5
Turbocharger PR	5.5	7.0	7.0	7.0
Intake Pressure, psia	80	103	101.5	83
P_3 / P_1	85	86	86	86
Fuel-Air Ratio	0.028	0.029	0.029	0.029
MEP, MPa	2.20	2.09	2.08	1.95
Cylinder Pr. @ Exhaust, psia	58.0 (1)	184.6	184.6	184.6
Exhaust Flow, pps	32.0	35.6	35.6	35.6
Exhaust Temperature, °F	1,025 (1)	1,300	1,300	1,370

1: At the turbocharger turbine inlet

Table 3 TC-RHT GTCC performance details.

	ISO	Hot	Hot & High
Turbine Output, kW	65,257	65,427	65,394
Gas Compressor Power, kW	22,832	23,935	24,360
Net GT Output, kW	41,588	40,669	40,216
GT + GE Output, kW	86,836	85,632	82,361
GT Exhaust Flow, pps	212.3	212.4	212.1
Total Fuel Flow, pps	8.90	8.91	8.72
GT Fuel Temperature, °F	365.0	365.0	365.0
GT Exhaust Temp, °F	1,095.2	1,096.1	1,095.7
Kettle Evaporator Duty, Btu/s	17,731	17,731	17,731
ST Output, kW	26,984	25,388	25,371
Gross CC Output, kW	113,820	111,020	107,733
Auxiliary Load, kW	1,033	1,958	2,312
Net CC Output, kW	112,787	109,062	105,421
Heat Consumption, kW	195,264	195,485	191,392
Net CC Efficiency	57.76%	55.79%	55.08%

Table 4 TC-RHT GTCC total installed cost and performance. ACC: Air-Cooled Condenser, OT-OL: Once-Through, Open-Loop, 2PNR: Two-Pressure, No-Reheat; intercoolers are included in the compressor price, gas turbine comprises combustor and turbine section only (compressor removed), gas engine cost is on installed basis with all modifications and accessories (in millions of USD).

	ISO	Hot	Hot & High
Heat Rejection System	OT-OL	ACC	ACC
Condenser Pressure, in. Hg	1.2	3.0	3.0
Steam Cycle	2PNR	2PNR	2PNR
Gas Engines	$35.0	$35.0	$35.0
Turbocompressor	$6.5	$6.5	$6.5
Gas Turbine	$10.0	$10.000	$10.0
Kettle-Type Evaporators	$0.65	$0.65	$0.70
Aftercoolers	$0.65	$0.70	$0.80
Heat Rejection System	$0.75	$4.0	$5.0
Circ. Water Pump	$0.115	$0.070	$0.075
Piping	$0.65	$0.72	$0.72
Bottoming Cycle	$34.0	$40.0	$45.0
Total Installed Cost (TIC)	**$88.3**	**$97.6**	**$103.8**
GTCC Output, MW	113	109	105
TIC per kW	**$783**	**$895**	**$985**
GTCC Efficiency	57.8%	55.8%	55.1%
GTCC Heat Rate, Btu/kWh	5,907	6,116	6,195

The single metric to quantify the combined effect of output, efficiency and installed cost is the **levelized cost of electricity** (LCOE or simply COE).

The dollar equivalent of COE is referred to as the **value**. The value is quantified by the **maximum acceptable increase in capital cost** (MACC). MACC is the capital cost equivalent of a given plant performance improvement (output and/or heat rate) with no change in COE. For details, please refer to the article by Gülen [8].

The value calculations are carried out with the following assumptions:

- 6,000 annual operation hours
- 16% capital charge rate
- Levelization factor of 1.169
- Fuel cost $5 (typical USA) and $12 (e.g., in South Africa) per million Btu (HHV)
- Value of heat rate with a **realization factor** (RF) as described in Ref. [8]

Based on the performance and cost information presented in the tables above, the value proposition is as follows:

- 57+% efficiency at 112 MW (conservative)
- Pre-engineered standard block approach
- Can be custom designed as well (for different engine/turbine combinations)
- Performance insensitive to altitude
- Performance relatively insensitive to ambient temperature
- Bottoming cycle contribution to overall GTCC is lower than standard GTCC
- Simple "module-by-module" construction amenable to easy shipment of individual components
- At comparable size and high fuel cost (e.g., $12 gas fuel)
  - $125 to $200 million value vis-à-vis reciprocating engine or aero-derivative (simple cycle) plants with multiple units
  - $15 to $50 million value vis-à-vis most efficient GTCC at ISO
  - $40 to $100 million at high altitude/ambient (e.g., Plains End in Colorado)

A GLIMPSE INTO WHAT *COULD* BE

In this section a new gas engine design is envisioned to explore the entitlement performance of a TC-RHT GTCC system. The bottoming cycle is 2PRHT and the gas turbine is the same F class unit considered earlier (i.e., loosely based on a GE 6F machine). The major distinction is that the second (reheat) combustion takes place in a **pulse detonation combustor**.

The new gas engine (a “blank sheet” design) has a compression ratio of 15 and MEP of 2.2. The exhaust gas temperature is around 1,050°F. The feasibility of the conceptual new engine design (materials, size, cylinder wall thickness, vibration and stress management, cost, etc.) remains to be seen. What is not disputable is the tremendous performance at a “miniature” scale (compared to the modern advanced F, G, H and J class behemoths) with an E class firing temperature (not even that in fact) and a modest bottoming cycle.

The requisite engine design and how it compares to today’s state-of-the-art (SOA) is summarized in Table 5. A 136 MWe (ISO base load) power plant at nearly **64% net efficiency** is possible with that engine (see Table 6). For details, the reader is referred to a recent article by Gülen [9].

Table 5 Gas engine performance (1)

		BASE	NEW
Electric Power	kW	9,525	9,520
Efficiency		48.2%	46.7%
Heat Rate	Btu/kWh	7,080	7,310
Compression Ratio (CR)		13.5	15.0
Turbocharger PR		5.5	7.0
Intake Pressure	psia	80	102
P_3 / P_1		85	93
Fuel-Air Ratio		0.028	0.026
Mean-Effective Pressure (MEP)	MPa	2.20	2.20
Cylinder Pressure @ Exhaust	psia	58 (2)	103
Exhaust Flow	lb/s	32.0	35.5
Exhaust Temperature	°F	1,025 (2)	1,045

1: Within round-off error
2: At the turbocharger turbine inlet

Table 6 TC-RHT GTCC ISO base load performance (1)

	Units	NEW
Turbine Output	kW	78,850
Gas Compressor Power (2)	kW	26,575
Net Turbine Output	kW	51,315
Turbine + Engine Output	kW	98,920
Turbine Exhaust Flow	pps	237.4
Total Fuel Flow	pps	9.81
Turbine Fuel Temperature	°F	77
Turbine Exhaust Temperature	°F	1,250
Kettle Evaporator Duty	Btu/s	19,175
Steam Cycle		2PRHT
Heat Rejection System		OT-OL
Steam Turbine Output	kW	38,295
Gross CC Output	kW	137,220
Auxiliary Load	kW	1,220
Net CC Output	kW	136,000
Heat Consumption	kW	213,530
Net CC Efficiency		63.7%

1: Within round-off error

2: Includes the booster compressor

CONCLUSIONS

A new fossil fuel fired combined cycle power plant concept is proposed. The TC-RHT GTCC combines existing and proven cycle heat addition variants, specifically, constant volume and reheat combustion, in a well-known configuration: turbocompounding.

Utilizing existing products with some modification, the introductory TC-RHT GTCC variant is a compact power plant with nearly 58% net efficiency at slightly higher than 100 MW. It is an attractive option in markets with expensive fuel and demanding site ambient conditions (e.g., high altitude and/or high temperature).

While the introductory variant is impressive enough, the TC-RHT GTCC, when designed as a new product from a “blank sheet of paper”, is capable of nearly 64% net efficiency. The enabler of this performance is pulse detonation reheat combustor.

Another key benefit of TC-RHT GTCC is its amenability to carbon capture and sequestration, vis-à-vis standard GTCC, due to the high CO_2 and low O_2 content of its gas (as a direct result of reheat combustion). This results in lower parasitic power loss and much reduced amine degradation making it an ideal, cost-effective candidate for state-of-the-art MEA based absorber-stripper CCS technology.

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