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Modern Gas Turbine Combined Cycle

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MODERN GAS TURBINE COMBINED CYCLE

NET THERMAL EFFICIENCY RATINGS OF 60% ARE HERE — WHAT'S NEXT?

S. C. Gülen

The indisputable king of the fossil fired electric power generation realm is the gas turbine combined cycle (GTCC) power plant with modern F-, G-, H- and J-class machines. At 60+% net thermal efficiency (officially clocked in a commercial installation in 2011), it is ten percentage points ahead of its nearest challenger (an ultra-supercritical pulverized coal power plant). As such, especially under the light of the recent discovery of abundant shale gas reserves, natural gas burning GTCC is all but certain to be a major ingredient in a power generation mix for the foreseeable carbon-averse future.

The seventieth anniversary of the first modern mass-produced jet engine (Junkers Jumo-004 turbojet powering the world's first jet fighter, Messerschmitt 262) presents an apt occasion to recap the evolution of the technology and gauge its future potential. In order to avoid hyperbole and commercialism, it is imperative to ground the discussion in firm theory (to the extent possible in a short article) and knowledge of history (sometimes the obscure aspects of it).

Beginnings

Anselm Franz's Jumo-004 was a culmination of work done by many giants in the field, primarily Hans von Ohain and Sir Frank Whittle, who walked in the footsteps of earlier inventors from 18th and 19th centuries. In terms of basic engine architecture, Jumo-004 was no different from its modern descendants, including can-annular combustor and stacked-wheel rotor construction with serrated Hirth couplings (the same as in latest H-class units of one OEM).

The interested reader can find many excellent references discussing the engine in detail. Suffice to say that its hollow turbine blades, manufactured from folded and welded 12-% chrome alloy, were cooled from air bled from the compressor. While built around a modest cycle with pressure ratio (PR) of only about 3 and turbine inlet temperature (TIT) of 1,427 F (775 C), it is not a big stretch to claim that Franz and team (not to mention the competing teams in UK and USA at the time) could have designed a bona fide E-class gas turbine before 1950 if they had the right materials – in addition to removal of restrictions imposed by wartime considerations.

After all, when one looks beyond its intricate accessory systems for lubrication, fuel delivery, cranking, etc., the gas turbine is an extremely simple machine designed to compress air, add fuel to react with oxygen in the air and then expand the mixture of reaction products. In essence, it is the practical embodiment of the Brayton (Joule) cycle, which, like all heat engine cycles, is a valiant albeit very poor attempt to replicate the ultimate heat engine cycle: the Carnot cycle.

As such, gas turbine performance is dictated by two cycle parameters: PR and TIT. On an ideal (commonly referred to as air-standard) Brayton cycle basis, the former dictates the cycle efficiency and the latter the cycle specific work output. In real cycles with aero-thermodynamic, hydrodynamic, mechanical and cooling losses, both have positive impact on simple cycle efficiency while TIT is of prime importance to the combined Brayton-Rankine cycle efficiency. The bottom line is that there is one and only one path to further improvement of simple or combined gas turbine cycle efficiency: ever increasing TIT with commensurate rise in cycle PR. This was already predicted at the dawn of the jet age by Adolf Meyer in his 1939 paper presented at a meeting of the Institution of Mechanical Engineers in London, UK.

Carnot Limit

The Carnot cycle is the translation of the second law of thermodynamics into engineering jargon: One cannot build a heat engine operating in a cycle and more efficient than the equivalent Carnot engine. The impossibility of even approaching the Carnot limit in practice stems from the near impossibility of attaining heat transfer at constant temperature (yes, there is an exception and it will appear later in the narrative). Thus, each gas turbine Brayton cycle with known PR and TIT can be translated into its Carnot-equivalent via mean-effective heat addition and heat rejection temperatures, METH and METL, respectively (Figure1). Following the standard cycle notation, then

$$METH=\frac{T_3-T_2}{\ln(T_3/T_2)} \quad \text{with } T_2=T_1\cdot PR^{k_{air}} \qquad [1]$$

$$METL=\frac{T_4-T_1}{\ln(T_4/T_1)} \quad \text{with } T_4=T_3\cdot PR^{-k_{gas}} \qquad [2]$$

The Carnot-equivalent Brayton cycle efficiency is simply

$$\eta_{C,BC}=1-\frac{METL}{METH} \qquad [3]$$

where k_{air} = 0.2831 and k_{gas} = 0.2270 and T_4 is *the* TIT for the ideal engine. Note that the Carnot-equivalent efficiency of Eq. [3] is much lower than the *ostensible* Carnot efficiency given by

$$\eta_C=1-\frac{T_1}{T_4} \qquad [4]$$

which, in fact, is the first response to the inquiry of ideal efficiency of a given cycle. (Note how it completely ignores the cycle PR, which, in fact, is the primary driver of efficiency). Since the gas turbine industry is not in the business of building Carnot engines, what good is Eq. [3] to a practitioner? As it will be demonstrated below, Eq. [3] is a potent tool to estimate actual gas turbine performance. Furthermore, it highlights the fact that low temperature (heat rejection) is as important, if not more so, for achieving the highest possible cycle efficiencies. With the focus on the high temperature (heat addition) side of the cycle, this fact is sometimes ignored.

Which Temperature?

First consider the current gas turbine technology landscape where the main classification parameter is TIT (Figure 2). In terms of sheer numbers, it is dominated by standard E (1,300 C TIT) and F class (1,400 C) units with air-cooled (utilizing compressor bleeds) turbine hot gas path (HGP). Recent introduction of advanced F-class machines (one OEM refers to them as "H" class, herein referred to as H-OLAC or H with open-loop air-cooling to distinguish it from the steam-cooled H-class) brought the standard F-class into the realm of steam-cooled G- and H-class technologies (1,500 C TIT). The latter class (herein H-CLSC or H with closed-loop steam cooling but better known as the H-System per its OEM) with six units in commercial operation since 2003 is currently not offered by the OEM. However, it has a special place in the gas turbine technology map.

Apart from the reheat gas turbine (labeled as sequential combustion by its OEM) with much higher number of units in commercial operation, H-System with two fully

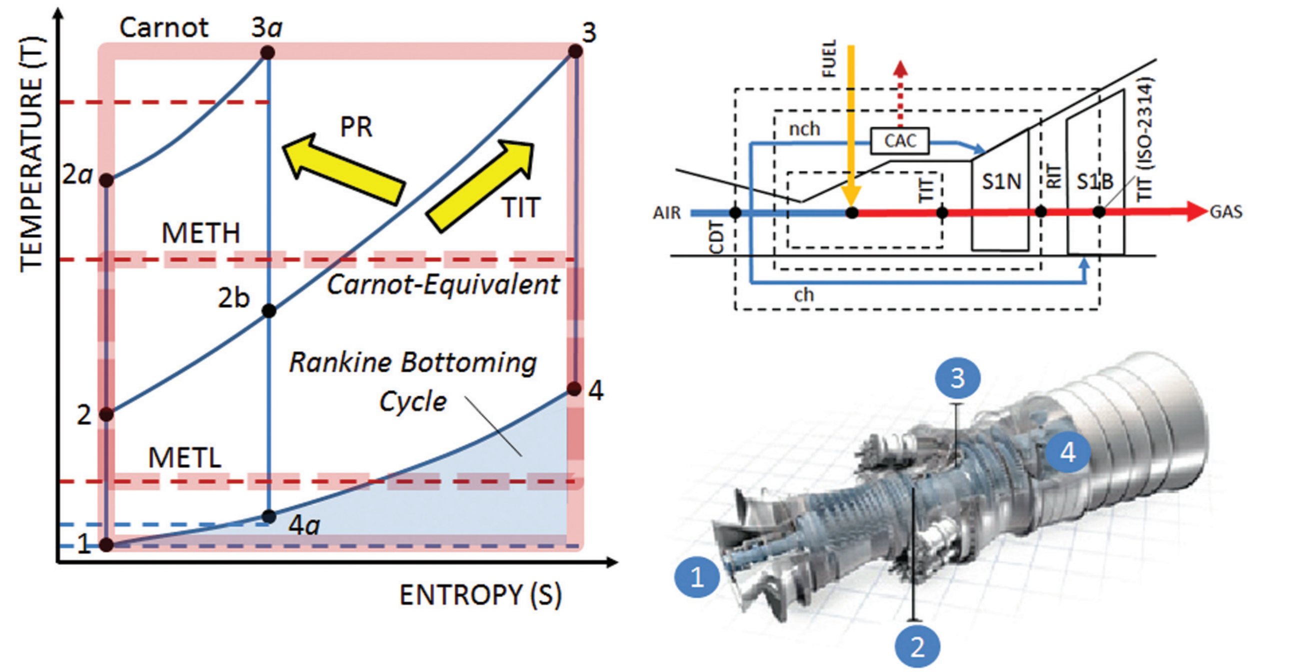

Figure 1: Figure 1 Gas turbine Brayton cycle. S1N: Stage 1 Nozzle (Stator), S1B: Stage 1 Bucket (Rotor), CAC: Cooling-Air Cooler, CDT: Compressor Discharge Temperature, nch: Nonchargeable (denotes compressor extraction air used to cool parts upstream of S1B inlet, ch: Chargeable (denotes compressor extraction air used to cool parts downstream of S1B inlet).

steam-cooled turbine stages (both stator and rotor) is the only proven non-standard industrial gas turbine architecture (G-class units with steam-cooled combustor transition piece and turbine rings can be classified as fortified air-cooled machines).

Note that TIT in Figure 2 is the temperature at the inlet of the turbine (or, equivalently, at the combustor exit). It is the best possible proxy for the highest Brayton cycle temperature in a real gas turbine with variable composition of the working fluid and myriad leaks and cooling flows (The true highest cycle temperature, by the way, is in the combustor's flame zone).

The TIT is frequently confused with two other temperatures, the so-called firing temperature and the TIT per ISO-2314 standard. The former is a real temperature in the sense that it can (in theory) be measured whereas the latter is a hypothetical number. Also known as Rotor Inlet Temperature (RIT), the firing temperature is arguably the most important gas turbine parameter (even more so than TIT) because it quantifies the true work generation ability of the cycle working fluid. The difference between TIT and RIT is a direct measure of the HGP component material durability (alloy and casting) and effectiveness of thermal barrier coating (TBC) and cooling technologies.

The ultimate limit of RIT = TIT is the holy grail of the turbine designer (or, more precisely, the metallurgist). As it is, the lowest registered delta between the two is about 80 F, which has been achieved in the H-System deploying buckets made from single crystal alloy (durability) with TBC (protection) and closed-loop steam cooling (no hot gas temperature dilution). In air-cooled gas turbines, the RIT-TIT delta is around 200 F, somewhat lower for the most advanced F/H class machines and somewhat higher for the others.

Rule of 75%

Now back to Eq. [3]: What good is it to the practitioner? As illustrated in Figure 3, the answer is "quite a lot". When the efficiencies of actual gas turbines reported in trade literature are plotted as a function of TIT, the regression line going through the data points is almost a perfect match with Eq. [3] multiplied by a factor of 0.75 – henceforth the Carnot factor. To get an idea about the historical development, Jumo-004 (2,000 lb thrust at ~700 ft/s speed) with a PR of only 3 and TIT of 775 C had a Carnot factor of about 0.54.

Several interesting observations can be made from Figure 2:

1. Modern gas turbine technology is

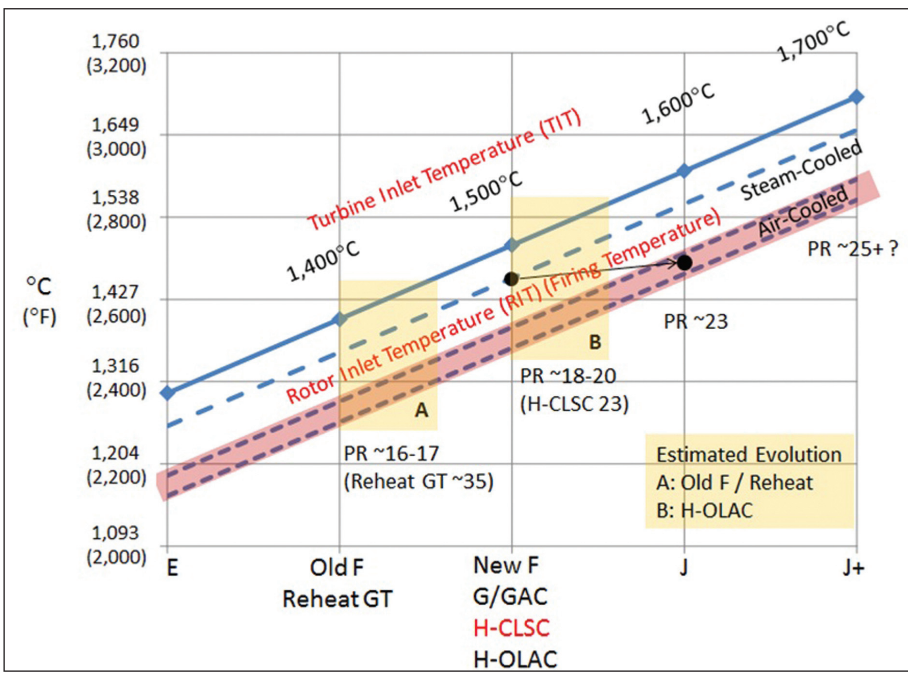

Figure 2: Gas turbine technology landscape

doing a laudable job of achieving 75% of the theoretical maximum. (Also shown in the Figure 3 is the ostensible Carnot efficiency, which should be best ignored – it puts the gas turbine engineering community under an undeserved bad light)

2. While "brute force" approach, i.e., ever higher TITs, is still the main driver of efficiency, advances in materials, coatings and cooling technologies make inroads without pushing the TIT further

3. One should also mention the reheat combustion, which is effective in reducing the combustion irreversibility without increasing the TIT.

No data point exists for the H-CLSC class gas turbine because it is only available in a combined cycle configuration (where the bottoming steam Rankine cycle is the source of HGP cooling steam). One could obviously estimate the equivalent simple cycle efficiency but, since no numbers are made public by the OEM, it is left out of Figure 3. Nevertheless, as shown in Figure 2, the firing temperature level of the H-CLSC (at 1,500 C TIT) can only be matched (or possibly surpassed) at ~1,600 C TIT of the J-class. This should provide some idea about 1,500 C TIT H-CLSC-class efficiencies.

As it turns out, the rule of 75% also applies to the bottoming steam Rankine cycle of the GTCC power plant. Note that the METL for the Brayton topping cycle of a GTCC given by Eq. [2] is the METH for the Rankine bottoming cycle (RBC) of the same. Thus, the Carnot efficiency for the RBC is

$$\eta_{C,RBC} = 1 - \frac{T_1}{METL}$$

Note that METL for the RBC is T1, i.e., the ambient temperature. In a real cycle, this would be the steam temperature in the condenser. The key observation here is that the METL for the RBC is constant. In other words, isothermal heat rejection is indeed a reality for the steam Rankine cycle (latent heat transfer of condensation at constant pressure and temperature).

The efficiencies of actual GTCC steam turbines reported in the trade literature have been plotted as a function of GT exhaust temperature (the plot is not shown due to space limitations; it can be obtained from the author). Expressed as a frac- [5] tion of the RBC Carnot efficiency in Eq. [5], performance of the 3PRH units (adjusted for the feed pump power consumption) are found to be, just like its Brayton cousin, about 75% of the theoretical maximum (0.75±0.03 to be exact).

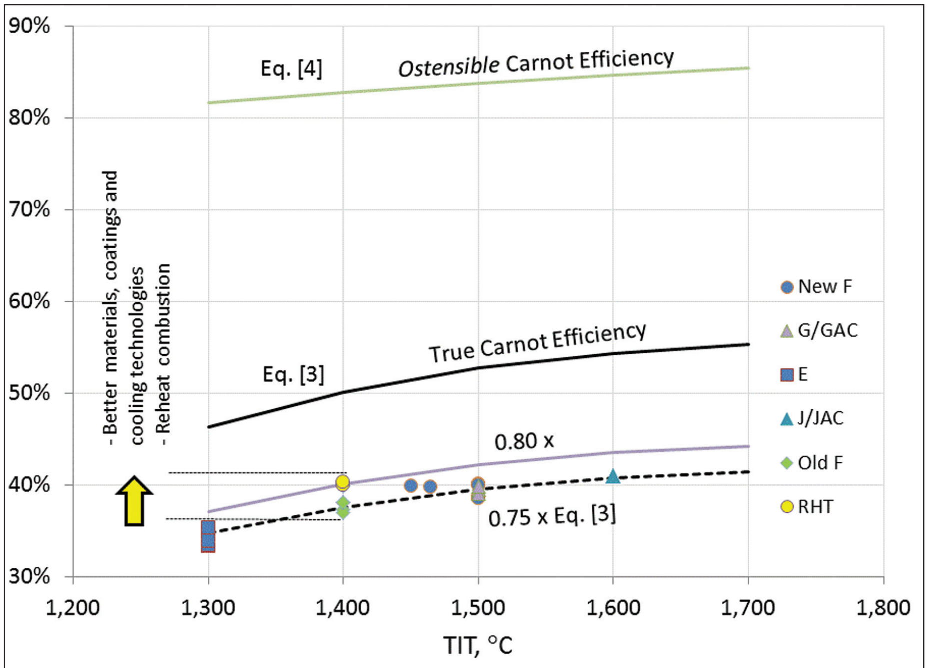

Figure 3: Gas turbine Brayton cycle efficiency. Data points are from trade publications

What's next?

How much more can be squeezed out of this technology for land based electric power generation remains to be seen. As far as the TIT goes, the number on the horizon is 1,700 C. The Carnot factor is unlikely to go much beyond 0.80 – unless ceramic matrix composite (CMC) turbine blades (already tested in a jet engine), wheels and other HGP components become a reality. This will close the gap between TIT and RIT and is by far the most potent game changer.

Material capability hampered the efforts of earliest gas turbine designers, who came up with brilliant solutions, which went largely unnoticed. Norwegian engineer Aegidius Elling's first successful gas turbine concept (patented in 1903) already included water cooling to bring the hot combustion gases from the combustor (adiabatic flame temperature of ~2,000 C) to about 400 C at the turbine inlet. The steam generated during the process was mixed with the gas and expanded in the turbine. In essence, Elling developed a poor man's H-System with an open-loop configuration a century before the real thing was first-fired in an actual power plant.

Around the same time, Hans Holzwarth of Germany built his first "explosion" turbine – a hybrid machine combining constant volume combustion (à la automotive internal combustion engine) with axial expansion in a two-stage velocity-compounded turbine. The great Aurel Stodola himself calculated 25.6% efficiency for the test of one of Holzwarth's later machines in Mühlheim-Ruhr. Holzwarth's work was continued by Brown Boveri Company (BBC) and the work done on his turbine eventually resulted in the first commercial stationary gas turbine for electric power generation in Neuchatel, Switzerland in 1939 (now an ASME historic landmark). This gas turbine (PR of 4.4 and TIT of ~540 C), which preceded Jumo-004 by three years, had a Carnot factor of 0.56 at 17.4% efficiency.

Right after WWII, engineers on both side of the Atlantic went to work to build better gas turbines. German and American engineers followed the turbojet path, the latter with heavier emphasis on military aircraft propulsion systems, whereas Swiss (BBC) stuck to the industrial gas turbine development. The dearth of high-temperature capable materials continued to be the bane of designers and this led to intricate cycle configurations to maximize efficiency with what they have available to them. These included:

- Water-cooled turbine blades (Germany, 1950s) and ceramic stationary blades (quickly dropped, though, due to very short parts life) for 1,000 C TIT. 1,055 C was achieved in the tests but the program eventually folded due to cost issues

- Recuperation (regeneration) was a textbook way to increase efficiency (because it increases METH and decreases METL simultaneously) at modest TIT. It was known to the earliest designers including Elling and was adopted by some postwar designs

- Intercooling and reheat combustion with multi-shaft designs (Switzerland, 1950s) were successfully developed by BBC and commercial installations followed (e.g., Port Mann Station in BC, Canada)

Water cooling was looked at later in the 1980s in the U.S. and dropped again. Eventually, though, steam cooling and reheat combustion made their way into commercial products (the latter much more successfully). Recuperation and intercooling are also available in commercial products, albeit in smaller aeroderivative gas turbines with high PR, where they make the biggest impact in simple cycle configuration.

Armed with this brief history and a few simple formulas, a glimpse into the future is in order. Figure 4 shows GTCC performance data from the trade literature. The state-of-the-art (SOA) line is from the formula

$$\eta_{CC,NET} = \left[c \cdot \eta_{C,BC} + \left(1 - c \cdot \eta_{C,BC}\right) \cdot c' \cdot \eta_{C,RBC}\right] \cdot (1-\alpha) \quad [6]$$

where $\eta_{C,BC}$ and $\eta_{C,RBC}$ are ideal efficiencies from Eqs. [3] and [5], c is the Carnot factor for the GT Brayton *topping* cycle (0.75 for SOA and 0.80 for advanced), c' is the Carnot factor for 3PRH steam Rankine *bottoming* cycle (0.75 for SOA ±0.05 for advanced and *cheap* versions) and α is the plant auxiliary load as fraction of the gross output.

The value used for α is 1.6%, which is appropriate for nominal rating purposes (roughly, a plant with once-through, open-loop steam condenser with access to a natural coolant source such as river, lake etc.). Real installations, say, with air-cooled condenser systems can be much higher than this, e.g. as much as 2.5% of the gross output. (In passing, note that the plant where 60+% efficiency was measured has access to rather cold cooling water from a nearby river.)

Pushing for ever higher TIT, even with suitable materials, TBC and film cooling techniques, becomes increasingly infeasible with today's DLN combustion technology due to stringent NOx regulations. Note that the 1,700 C TIT (super J-class?) systems are envisioned with up to 30% exhaust gas recirculation (EGR). Even with the emissions issue resolved (or ignored via shifting the onus downstream to the SCR), such high TITs are commensurate with high PRs (25 or even higher at 1,700 C) to keep the GT exhaust temperature down. The obvious reason is the design constraints imposed by long last stage blades (especially for the recent generation of 50-Hz machines with nearly 400 MW ratings) with tremendous centrifugal forces acting on them. Even with that problem solved, one should still consider the bottoming cycle limitations – currently the highest possible steam temperature is 600 C (1,112 F). Thus, going too much beyond the current GT exhaust temperature maximum of ~650 C will simply lead to a waste of GT exhaust exergy.

High PRs bring with them their own design issues – primarily due to very high air temperatures at the compressor discharge (note that reheat GTs with 35+ PR have ~1,000 F at the discharge) resulting in costly materials and excessive (chargeable) cooling air extraction. The latter problem is typically solved with cooling air coolers (CAC in Figure 1; typically kettle type evaporators to make steam for the bottoming cycle) with added cost and complexity.

There is not a lot of room left in component efficiencies; 3D airfoil designs enabled by advanced numerical codes and computational resources push them to their entitlement (92.5% polytropic efficiency is one cited ultimate value). Active clearance control, advanced seals for reduced leaks, advanced film cooling schemes are already deployed and it is really difficult to foresee how much more can be squeezed out of them.

At this point, as far as expectations of future GTCC efficiencies are concerned, it is hard to see how the oft-cited 65% barrier can even be approached anytime soon (let alone broken). In the absence of a game-changing development in materials obviating the need for cooling air extraction (or drastically reducing it), Figure 4 pretty much speaks for itself. As a final word, it should be recognized that the two venerable century-old technologies (in concept, that is), namely reheat combustion and closed-loop steam cooling (at the very least for the first stage nozzles), especially in combination, still hold great promise to achieve significant performance levels without forcing the issue in terms of TIT (and in NOx emissions with existing DLN combustion technology). One should also mention the significant improvement potential of constant volume combustion (see the efficiency cited above for Holzwarth turbine nearly a century ago, which was head and shoulders above those for its turbojet brethren for the next two decades); pulse detonation combustion is one way to achieve it in an industrial gas turbine. At the end of the day, in terms of possible non-metallurgical solutions, one can really conclude that there is indeed nothing new under the sun. TI

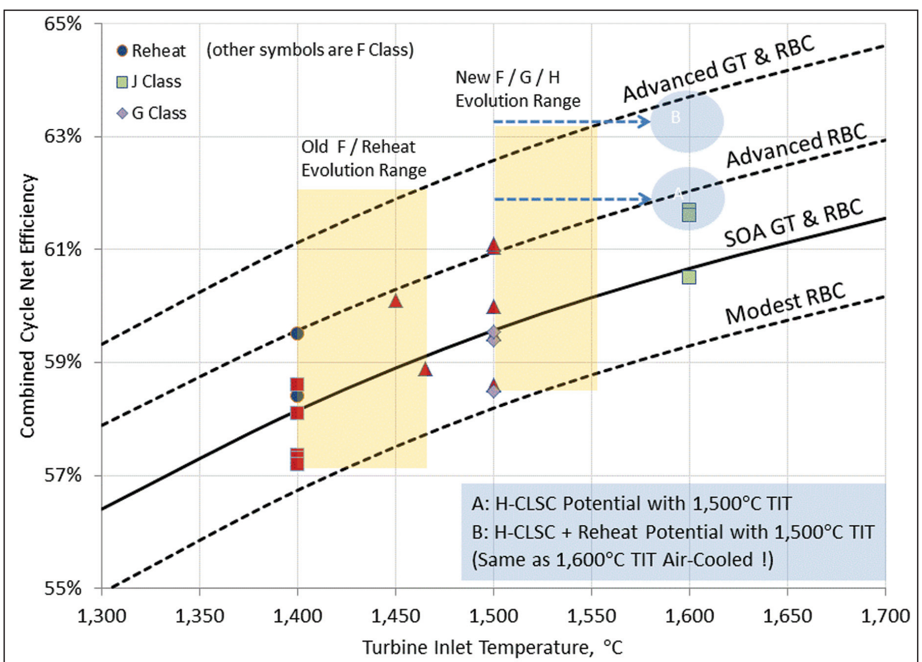

Figure 4: Gas turbine combined cycle (net) efficiency

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