Improving the Simple Gas Turbine Cycle with Compressed Air Energy Storage (CAES)

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Compressed Air Energy Storage (CAES) is considered an attractive and viable concept for balancing electrical load with production, and the basis for its operation has been well established for several decades now. Yet, despite clear benefits and promises, the technology has yet to gain large commercial acceptance: only two CAES plants are currently in use in the world today. This paper examines some of the reasons for this shortcoming, and argues that the development of dedicated CAES-based turbines in direct competition with a mature gas turbine industry is a significant drawback for its large-scale acceptance, and presents a novel way to implement CAES in a way that takes advantage of its perceived drawbacks to improve the simple gas turbine cycle, using three implementation options.

Nomenclature

- CAES = Compressed Air Energy Storage
- CAES-AI = CAES with post-expansion air injection into top cycle gas turbine combustor
- CAES-GT = Gas turbine cycle augmented with CAES pre-cooling, regenerative heating, and direct injection
- CAES-Intercharge = Gas turbine cycle augmented with compressor interstage cooling air injection
- CAES-Supercharger = Gas turbine with supercharged intake from CAES

1. Introduction

Compressed Air Energy Storage, or CAES, is a means of storing excess energy produced at a time of low demand, by using it to compress a gas (usually air), storing it in large caves typically underwater or underground, and using the stored gas at a later time, when demand justifies its use, to produce electricity. While this concept dates back to the 1940s, the first commercial CAES system was only deployed in 1978, in Huntorf, Germany. A second plant was built in McIntosh, Alabama, in 1991. A review of the literature does not reveal any new additional CAES plants commissioned in the world in the past 25 years, though much current discussion surrounds a renewal of interest in the concept.

In both the Huntorf and the McIntosh plant, air is compressed from ambient condition to a discharge state of about 70 bars and 150°C, using multiple stage electric compressors with water intercooling. With after coolers, the air temperature is further reduced (at constant pressure) to about 50°C, when it can be safely injected into storage reservoirs, salt caverns in the two existing examples, where the air ultimately reaches thermal equilibrium with its surroundings, around 15-20°C at McIntosh. This particular method of storing air energy, known as diabatic CAES, highlights one of the fundamental problem of the technology: the thermal heat of compression is entirely lost to the environment, and must therefore somehow be returned to the gas, usually through a combustion process, prior to being expanded through a turbine. Since the expansion turbine is completely decoupled from the compressor, it can generate nearly three times the power of a corresponding simple cycle gas turbine operating at the same conditions (pressure ratio and temperature, see Fig. 1). Thus, if the compressed air can be procured at a time when the electric rates are low, compared to when the turbine is running, the system may be profitable, despite the heat losses.
Figure 1: Schematic TS-diagram of the simple gas turbine cycle. The diverging nature of the isobars (curves 2-3 and 1-5) means that the energy required to compress the air (line 1-2) is provided using only part of the expansion through the turbine, between the same two isobars: line 3-4 powers the gas generator compressor, and line 4-5 is available to power a power turbine and a generator. In a CAES system, heating the compressed gas to state point 3 means that the entire expansion along line 3-5 is available for power generation, or about three times more than in the gas turbine cycle.

Adiabatic CAES, on the other hand, is a significant theoretical improvement on the CAES concept, as it eliminates the need for a fuel system to reheat the stored air prior to expansion in the turbine. It purports to store the heat of compression in a high heat capacity material, to return it later to the gas prior to expansion. Much higher cycle efficiencies are clearly possible using this method, though the engineering challenges are such that no such plant exists today.

Diabatic CAES is thus the only system that can be considered market-ready at the moment. However, even there, specialty hardware (primarily the power turbine, which must operate at pressure ratios many times larger than current gas turbine systems) developed specifically for CAES is a main factor that slowed the market penetration of CAES because of their inherent high cost. Another reason is the very fundamental thermodynamic problem that characterizes diabatic CAES: high-pressure stored air in this system means low temperature and therefore low thermal energy.

II. Current Developments in CAES

Over the past 10 years, the large-scale adoption of wind power by many utility companies has spurred renewed interest in CAES technology development. Wind power production cannot be easily tailored to energy demand, and therefore all means of energy storage are seeing this renewed interest. Two new CAES approaches in recent years are CAES-AI (Air Injection) and CAES-IC (Inlet Chilling)² (see Fig. 5). These “second generation” CAES technologies seek to marry the CAES power production step with an existing simple cycle gas turbine, and use the (significant) exhaust heat energy of the gas turbine as the primary heat source for power production through the CAES expansion turbine, thus turning CAES into a bottoming cycle, similar to the Brayton-Rankine combined cycles used in many power generation plants today. CAES-AI seeks to reinject the CAES expanded air into the combustor of a simple cycle gas turbine, whose exhaust is used to provide the energy necessary to solve the CAES low thermal energy problem, via a recuperator heat exchanger. Clearly, this limits the level of expansion available in the dedicated CAES turbine, as the turbine outlet pressure must now be at least as high as the primary gas turbine compressor outlet pressure, typically around 25-30 bars, though that air provides a mass boost to the gas turbine cycle, without the penalty of compression. CAES-IC on the other hand, allows the full expansion of the CAES gas to atmospheric level through the expansion turbine. The gas (air) is then injected directly into the primary turbine compressor inlet. Since the level of reheat through the recuperator is controllable, the temperature of the CAES air after expansion may be dropped below atmospheric inlet air to the gas turbine, to provide a chilling effect at compressor inlet. The efficiency
of these new CAES-enhanced cycle is reportedly above 70%, similar to adiabatic CAES. Still, the improvement provided by these modified CAES concepts only go part-way in mitigating the basic shortcomings of CAES in the first place: low air temperature requiring reheat, and dedicated expansion turbines.

III. The Simple Gas Turbine Cycle vs. CAES

The gas turbine industrial power plant, on the other hand, has been around much longer than CAES-dedicated expansion turbines, and represents a comparatively mature and competitively-priced technology, indispensable for power production today. With nearly a century of development, it has enjoyed many significant engineering improvements to boost its cycle efficiency: regenerative heating, steam injection (STIG), combined cycle generation, just to name a few, have made the gas turbine power generation cycle overall thermal efficiency routinely exceed the 50% mark. Thus, a power generation system that can directly leverage this engineering legacy, instead of requiring new engineering development ventures, immediately benefits from a significant cost advantage, and is inherently less risky, and presents the question: can a CAES cycle be developed, where the CAES expansion turbine is eliminated altogether?

A. A Problem with the Simple Gas Turbine Cycle

Fig. 1 shows that the work required by a gas turbine compressor is always dependent on the temperature points separating its inlet and outlet pressure. This thermodynamic effect, in fact, has motivated an entire industry to develop and promote pre-coolers and intercoolers in industrial gas turbines operations, with the result of significantly improving cycle efficiency: the simple gas turbine cycle suffers from high temperature (proportional to high thermal energy) through its compressor stages, which must be made up by the turbine, before any useful work can be done through a power turbine. This is the exact opposite of the problem facing diabatic CAES.

B. Two Problems, Many Solutions

While modern CAES research and development continues to focus on the long-term goal of adiabatic or isothermal implementations of the concept (with the clear benefit of completely eliminating the use of any fuel), this path has proven to be so challenging from a technical and financial standpoint, that it cripples new commercial CAES ventures, depriving utilities of the many CAES advantages realizable today, and ultimately delaying progress. The approach presented in this work is more pragmatic, leveraging mature technologies readily available today. A primary goal is to simply eliminate, inasmuch as possible, the dedicated expansion turbine traditionally used in CAES. With pressure ratios nearing 100 (e.g., in the Huntorf setup), these machines fall outside the range of most utility-scale gas turbine generator and thus requires expensive, long-term development programs, with all their associated risks, in addition to all the other risks associated with CAES.

Three new combined-cycle methods of energy production making use of diabatic CAES low temperature stored air supply are proposed and presented here, each eliminating the need for a dedicated expansion turbine, and leveraging instead the technology of existing gas turbine power plants; a fourth cycle, combining both a gas turbine and a dedicated air turbine (as in traditional CAES), optimizes further the thermodynamic efficiency of the cycle beyond what each system can achieve individually. In all cases, an important goal is to minimize the re-engineering of the gas turbine, while maximizing the overall cycle efficiency. Each system is described below:

1) CAES1, or CAES-Supercharger: A “boosted” gas turbine cycle, where the cold CAES stored air is throttled to a desired level, above atmospheric pressure (isenthalpic expansion; temperature remains essentially unchanged from the storage) and feeds the entire inlet of the gas turbine compressor through a sealed shroud, providing uniform pre-cooling as well as supercharge, see Fig. 2. The performance of the cycle is improved both by supercharging and by pre-cooling the inlet. Boost level is essentially limited by the reversed thrust load specifications on the gas generator spool(s), and maximum allowable pressure at the combustor and turbine inlet, for a fixed pressure ratio. The mass flow balance of the compressor/turbine gas generator is unaffected, but the pressure ratio across the compressor may now be kept significantly lower than that of the gas generator turbine/power turbine assembly, increasing the ratio of power available for power production. The performance of the gas turbine also becomes unaffected by variations in atmospheric temperature.
Figure 2. CAES-Supercharger. CAES storage air is throttled and ducted directly to the compressor (C), then flows to the combustion chamber (CC), and to the turbine (T). No ambient air enters the gas turbine. The CAES reservoir pre-cools in the intake, compared to ambient conditions.

Figure 3. CAES-Intercharge. The gas turbine operates with ambient air, but each compressor stage receives air from the CAES storage for intercooling and increased mass flow.

Figure 4. CAES-GT. CAES flow path is shown with dash lines; ambient air flow path is shown with solid lines. Note the two heat exchangers (HX), for precooling and for regenerative heating.
2) **CAES2, or CAES-Intercharge:** This cycle consists of an inter-stage compressor injection process, where the CAES air supply is throttled to a level suitable for pressure-matching of each compressor stage, for injection into that stage, see Fig. 3. Each stage thus benefits from intercooling by mixing with the incoming CAES supply, and the mass flow rate increases through each subsequent compressor stage. The mass flow rate through the last compressor stage matches the flow through the turbine, but the total work required by all compressor stages is reduced. There is no thrust loading impact on the gas generator spools, in contrast to the CAES-boost cycle.

3) **CAES3, or CAES-GT:** This cycle maximizes the benefits of CAES on the simple cycle gas turbine, with three distinct features:
   a. Pre-cools the gas turbine inlets, taking advantage of the CAES low temperature reservoir;
   b. Uses the turbine exhausts for regenerative heating – minimizes fuel energy required in the combustor;
   c. Injects the heated air directly into the gas turbine combustor – leverages CAES pressure and bypasses the gas turbine compressor.

The cycle is illustrated in Fig. 4. From an overall exergy perspective, this cycle is optimized when the CAES reservoir pressure is near that of the gas turbine compressor output, but the cycle efficiency of the gas turbine is maximized in all cases.

In order to present the relative advantage of each cycle configuration, a case study based on their implementation with a General Electric LM5000 aero-derivative power plant, with real-life operating conditions is presented.

C. **Combined Wind-Brayton Cycles**

At the time of the original CAES plants implementation, the main goal of the systems consisted of taking advantage of daytime electrical tariff variations that exist in certain markets. For example, if in a given market a CAES compressor could run at nighttime, when the cost of energy is very low, then, during daytime when the electrical rate justifies its use, the compressed air could drive a dedicated expansion turbine to produce power at a higher tariff. Today, a main driver of new CAES projects is the increasing market penetration of renewable energy production, primarily wind. It is believed that beyond 30% of grid power production produced by wind and solar plants, the utility companies’ ability to modulate power production so that load and contingency reserves are met will work only “if the wind/solar forecasts are perfect” – an untenable long term business situation, made worse as the grid make-up exceeds the 30% threshold. California is committed to providing 30% of its electrical power needs through renewable sources by 2020, and 50% of its need by 2030, so the need for immediate energy storage solutions is clear. Finding suitable locations for large scale CAES (similar to Mcintosh) on the U.S. mainland should not be a problem, as it is reported that “appropriate geology (porous media, hardrock or salt) is available in 85% of the United States” 10. While the proposed cycles above are suitable for mainland CAES applications, with underground temperatures typically around 10-20°C, the pre-cooling and intercooling advantages described above are markedly improved with offshore wind farms, where ocean temperatures at storage depth are closer to 5°C for most of the west coast of North America. Offshore windfarms, expected to represent a significantly growth sector in utilities in the coming decades in the U.S., offer certain unique advantages to diabatic CAES, not readily available to similar land-based facilities. For example:

- simple, well regulated high pressure storage underwater, either in caverns or man-made storage reservoirs;
- man-made reservoirs can be as simple as anchored canopies (single axis load path, simpler than other pressure vessels);
- large thermal reservoir, for inter-cooling offshore compressors;
- natural cooling of pipe conduits to storage reservoir, eliminating the needs for additional heat exchanger systems (versus Huntorf and Mcintosh type plants).

For the sake of the analysis of the systems presented in this paper, a reservoir such as this is assumed to exist. The reservoir is assumed to be at least 300m deep (roughly 30 bars minimum), with a temperature of 5°C. This storage reservoir is ducted to a gas turbine power generating plant, located on land at an economically viable location, with ambient temperature significantly above that of the CAES reservoir.
Figure 5. CAES-AI (top, figure a) and CAES-IC (fig. b). Both may be considered “bottoming cycles”, where the thermal energy required for stored air power generation is supplied by “recuperators” operating from the exhaust of a simple GT cycle (from Ref. 2).
D. Gas Turbine Case Study

One method of reducing the cost of power plants is to leverage the original development cost by modifying existing gas turbines designed for other purposes, instead of designing dedicated gas turbines for power production. Aero-derivative power plants is an example of this practice, and the popular General Electric CF6-series of engines (used, for example, on the Airbus A300, Douglas DC-10, Boeing B747 and many others) gave way to the widely used industrial gas turbine LM5000, see Fig. 6, along with the LM2500 and LM6000 of the same family. This power plant was selected in this study because of its widespread use: hundreds deployed around the world, accumulating millions of hours of documented performance. Other factors include its long history of engineering modifications and upgrades (see, for example, Ref. 8), and relatively low cost: a recent market search showed complete power plants available for less than $2M(USD)$\textsuperscript{13}. By comparison, the McIntosh CAES plant cost a reported $65M(1991$ USD$)$\textsuperscript{14}, and Dresser-Rand (the equipment manufacturer for the McIntosh plant) recently presented plans to supply equipment for a new traditional diabatic CAES plant (317MW) in western Texas, the first new proposed CAES plant in 25 years, for an estimated cost of $200M(USD)$\textsuperscript{15} (the project has been indefinitely postponed, due to high cost). The need for cheaper alternatives to traditional CAES cycles thus cannot be overstated, and a traditional, proven industrial gas turbine, combined in a new cycle with CAES, may provide such an alternative.

The basic performance parameters of the LM5000 operating in a simple Brayton at a standard day of 15°C and 1 atm, are summarized in Table 1. The reported performance associated with a turbine modification to include steam injection in the turbine stages of the gas turbine (STIG) is also included, as a comparison, to illustrate the design flexibility and history of re-engineering of the LM5000.
### Table 1: Baseline operating and performance parameters for LM5000 (Based on Refs. 5 and 8)

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<td>37</td>
<td>33</td>
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<td>Steam Injection</td>
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<td>50</td>
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<td></td>
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<td>+ 32,700 (LP steam)</td>
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### IV. Modeling and Analysis

The procedure used to evaluate the performance of the various CAES-combined cycles proposed here started with the development of a thermodynamic model for the simple gas turbine cycle that matched the actual machine. The purpose of the model is to determine the relative advantage or disadvantage of different cycle configurations, versus the baseline simple cycle gas turbine, so that matching the exact working gas composition (e.g., products of combustion, water injection used for NOx abatement, etc.) is less important than to find a setup that matches the baseline performance, and then keep the basic parameters constant throughout each iteration. “Ideal gas air” and its properties, as defined by Lemmon et al. was chosen as the working fluid in all cases. For the purposes of all analyses presented here ambient conditions surrounding the power generation plant were defined as $T_\infty = 20^\circ C$ and $P_\infty = 1$ bar. The modeling environment used to perform the analysis consisted of the Engineering Equations Solver (EES) developed by F-Chart. This environment provides an ideal platform for this type of research, as the cycle characteristics can be programmed directly along with associated thermodynamic gas property lookup, and solved implicitly with minimal algorithm development, in a simple modular approach. If individual component performance maps became available, they could easily be integrated into the programming environment to account for off-design performance. Since they were not available at the time of this analysis, both the compressor and turbine were modeled with constant isentropic efficiencies of 90% and 85%, respectively, representative operating points for industrial gas turbines such as these. The combustor was assumed isobaric.

For all of the proposed CAES-modified cycles, the analysis assumes that an infinite reservoir of air at 5°C and arbitrary pressure is available to the gas turbine. The work required to produce this reservoir is not included in the analysis (and thus the “cycle efficiencies” presented only refer to the gas turbine cycle itself).

#### A. Baseline Cycle

The TS-diagram for the baseline performance of the LM5000 under baseline conditions is shown in Fig. 7. With the mass flow rate shown in Table 1, this cycle matches the reported performance of the machine, with a thermal efficiency of 37% and an output of 33MW. State points 1 and 2 represent the inlet and outlet of the compressor, respectively; state points 3 and 4 represent the inlet and outlet of the turbine, respectively. The temperature at state 3 is an important design point for a gas turbine (the turbine inlet temperature, TIT). For the LM5000, that design point is 1150°C; this value remains constant for the analysis presented here.
B. CAES-Brayton Combined Cycle 1: CAES-Supercharger

This first proposed modification of the simple gas turbine cycle, shown schematically in Fig. 2, uses the cool, pressurized CAES air for two specific benefits:

1. Naturally pre-cooled inlet, reducing compressor work at all ambient temperatures above that of the reservoir;
2. (Significantly) reduced pressure ratio across the compressor compared to that of the turbine, increasing the net work output to the generator.

This “CAES-Supercharger” cycle, arguably the simplest and least expensive to implement in theory, includes the limitation that it subjects the entire low pressure gas turbine spool to a forward thrust for which it was not originally designed, and which may make it unsuitable for many industrial gas turbines, or at least require potentially re-engineering of the shaft support – something that we explicitly attempted to avoid. The inlet cowlings are now also pressurized, and thus requires consideration. However, given that the CF6 turbine was originally designed to operate with a rearward thrust on the main spool of around 185,000 N (41,500lb.), and that some residual thrust necessarily exists on the gas generator spool (with a free power turbine at least), reversed thrust may provide some cancellation benefit, and reduce bearing friction. Without specific mechanical data on the gas turbine spools design (not available to the author at the time of writing), it is not possible to evaluate how much reversed thrust is acceptable on the LM5000. For the sake of illustration, the cycle is run with a supercharge of 5 bars at the inlet (meaning that the CAES reservoir is throttled down to this pressure). The resulting cycle is shown in Fig. 8.

Despite the clear exergy losses associated with throttling our reservoir to such a low pressure (relative to the reservoir), this proposed change improves the LM5000 performance dramatically: the gas turbine cycle efficiency improves to 55.9%, and the net power output available to the generator increases to 73MW, more than twice that of the original machine, and nearly 50% higher than the STIG modification in use today, without the long lag time penalties associated with all combined cycles and steam injector cycles that require boilers. We note, however, that in this scenario, there is no ambient make-up air. The entire gas turbine mass flow must be supplied by the CAES reservoir, similar to a traditional diabatic CAES setup such as the McIntosh plant.

Figure 7: The baseline LM5000, operating in a simple cycle. The compressor and turbine efficiencies are 90% and 85%, respectively; TIT is set at the LM5000 design point (1150°C). The cycle thus defined matches the manufacturer’s baseline specifications for thermal efficiency and power output.
C. CAES-Brayton Combined Cycle 2: CAES- Intercharge

A second proposed cycle builds on the CAES-Supercharger:
1. avoid pressurizing the intake,
2. reduce exergy loss, and
3. reduce the CAES air mass flow rate by using ambient air intake (thus enabling operation for the same duration as CAES-Supercharger, with a smaller CAES reservoir).

This new cycle, dubbed CAES-Intercharge, consists of injecting CAES air directly in each stage of the compressor, boosting the post-stage mass flow and intercooling the stage prior to the next one, see Fig. 9. The net work across the compressor may still be reduced, while the overall mass flow rate has increased (as opposed to CAES-AI, where the low temperature of the injected flow does not assist the compressor, and creates an additional fuel energy load to reach the same TIT, compared to an equivalent non-injected cycle, Fig. 5 a). The net cycle benefit is similar to that of CAES-Supercharger, with a reduced CAES mass flow rate.

Figure 8. LM5000 cycle with CAES supercharge. The new cycle is overlaid onto the original LM5000 cycle, and shows the reduced compressor work.
D. CAES-Brayton Combined Cycle 3: CAES-GT

The CAES-GT is a further improvement on CAES-Intercharge, with a single injection port at the compressor outlet, see Fig. 4. Two other components required, heat exchangers, do not directly impact the operation of the gas turbine, but directly improve its performance. The CAES reservoir air is passed through an intake heat exchanger and act as a pre-cooler to the compressor, which in return pre-heats the CAES air flow; a second heat exchanger, across the turbine exhaust, further heats up the CAES air flow, which is then injected (at a pressure slightly above compressor outlet) directly into the combustor. The entire system exergy is maximized if the CAES reservoir pressure matches closely the compressor outlet pressure, around 30 bars for a system built around the LM5000.

Note that this cycle allows the gas turbine to operate in standalone mode at any time as well, and benefits from any level of airflow from the CAES reservoir. Start times and shutdown times remain identical as the simple gas turbine cycle.

We illustrate the performance of this system by assuming that two heat exchangers are available on the LM5000, each with effectiveness of 80%. Our baseline operating point assumes that the CAES air input equals that of the simple gas turbine operation (as a basis for comparison with CAES-Supercharger). Assuming an unmodified gas turbine operating at the same pressure ratio and TIT as the simple cycle, this means that the flow speed through each turbine stage doubles (affecting swirl angles through each stage), and thus this scenario may well be beyond the range of operation of this turbine. Still, with the same mass flow rate from the reservoir, it presents the clearest basis for comparison with CAES-Supercharger.

The resulting cycle is presented in Fig. 10. This 1-to-1 match between compressor mass flow and CAES mass flow produces a cycle that generates 136MW of net power, or nearly double the output of the CAES-Supercharger and four times the baseline turbine, with an efficiency of 65.9%, nearly twice that of the original system, for the same CAES intake flow as CAES-Supercharger. Again, it is not suggested that an unmodified industrial gas turbine be able to operate so far beyond its regular design point, but its performance will be improved regardless of the CAES mass fraction used. To illustrate this, the performance of the LM5000 on the CAES-GT cycle, at various levels of CAES fractional flowrates, is summarized in Table 2. The efficiency of the gas turbine thus matches that of the STIG modified LM5000 with a CAES mass flow rate of 10%; the power output at that point is below that of the STIG turbine, and suggests that the effects of this modification on the operation of the gas turbine are acceptable.
Figure 10. CAES3, or CAES-GT. This cycles optimizes the match between a CAES reservoir and the target turbine. The compressor inlet is precooled and $T_1$ is now at 8°C, for an ambient temperature of 20°C; the compressor outlet ($T_2=498°C$) is mixed with the CAES air (now at $T=370°C$, after regenerative heating with the exhaust at $T_4=457°C$) in a 1-to-1 ratio for this illustrative case. The result is a doubling of the mass flow rate of the simple cycle into the combustor, and an average temperature before combustion of $T_{mix}=432°C$.

<table>
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<tr>
<th>CAES Mass Fraction</th>
<th>Thermal Cycle Efficiency (GT only) [%]</th>
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<tr>
<td>5%</td>
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E. Improving CAES-AI with CAES-Intercharge: Combining the Gas Turbine with a Bottoming Cycle Expansion Turbine

Finally, the CAES-Intercharge cycle motivates at least one last cycle improvement, one that builds on the new generation of diabatic CAES cycles (i.e., with expansion turbines), where CAES storage reservoir pressures exceed the compressor delivery point of a simple cycle gas turbine. So this cycle requires both a gas turbine and a standard CAES expansion turbine, and builds upon the advantages of CAES-AI of Ref. 2.

The output of the CAES expansion turbine in the bottom cycle (state point 12 in Fig. 5 a), is maintained at a pressure higher than the gas turbine compressor outlet, though cooled below the CAES reservoir by this expansion (within the limits of water and ice entrainment). It now pre-cools the gas turbine inlet air, and proceeds exactly as described in CAES-Intercharge, with controlled throttle and injection at various stages of the main gas turbine compressor. This combined cycle provides the advantages of the CAES-Intercharge cycle, with improved pre-cooling (limited essentially by water and ice entrainment at operations below the CAES reservoir dew-point temperature); it allows for CAES reservoir pressures that are significantly higher than the other cycles presented here, without incurring an exergy penalty, since a dedicated CAES expansion turbine used the full CAES reservoir pressure, instead of throttling to the injection pressures, and it reduces the thermal penalty at the combustor, in contrast to CAES-AI.

V. Conclusion

The simple gas turbine cycle continues to represent one of the best means of power generation at a time when integration with renewable energy sources becomes paramount, because of its flexibility and rapid ramp up and cool down. Combining this platform with energy storage in the form of CAES, for the purpose of boosting the efficiency of the gas turbine cycle, is one way to rapidly implement CAES on a larger scale without large development cost, and to cope with an ever increasing fraction of renewable energy in the mix. Three new cycles, making use of CAES to enhance the efficiency and output of a standard gas turbine engine, show how it is possible to dramatically improve its performance. A pragmatic approach to power production, allowing diabatic CAES to contribute today, is one method to rapidly reduce carbon emissions at a relatively lower cost than traditional CAES.

References


