INFRASTRUCTURE

MINING & METALS

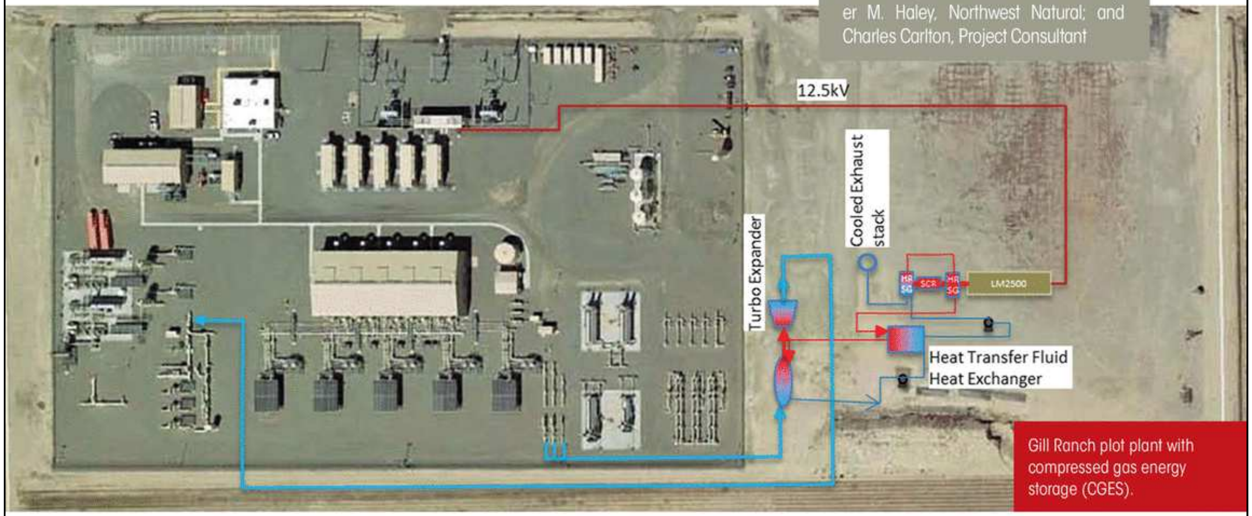
NUCLEAR, SECURITY & ENVIRONMENTAL

OIL, GAS & CHEMICALS

Compressed Gas Energy Storage

2017

Authors:
S. Can Gülen, Bechtel Infrastructure & Power Inc.; Sarah S. Adams and Roger M. Haley, Northwest Natural; and Charles Carlton, Project Consultant



Gill Ranch plot plant with compressed gas energy storage (CGES).

Compressed Gas Energy Storage

BY S. CAN GÜLEN, SARAH S. ADAMS, ROGER M. HALEY AND CHARLES CARLTON

There is a trend across the USA to mandate increasing amounts of energy to be derived from renewable resources. Electric utilities in California, for example, are required to have 33 percent of their retail sales derived from eligible renewable energy resources by 2020. It is speculated that this requirement may later increase to 50 percent.

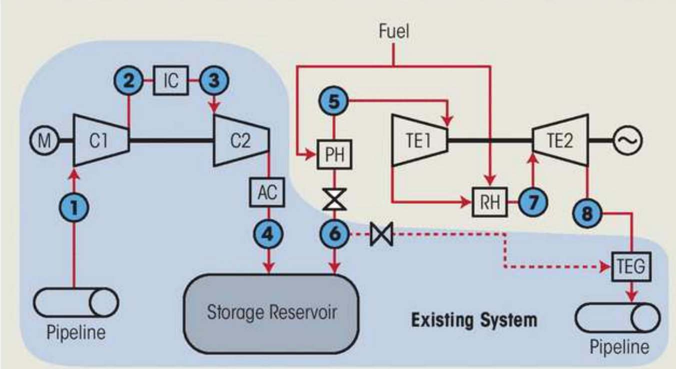
The challenges of integrating major renewable energy resources, such as wind and solar energy, into the electric distribution grid are well-known: intermittency and low predictability, which renders renewable-source generation not dispatchable. In layman's terms, "they are on when they are on" and cannot be brought on line when called upon to satisfy immediate demand. Inevitably, this makes it difficult to match electric power demand and supply, which is very important for both users and producers of electrical energy. Currently, the solution to maintaining a stable grid with a generation portfolio including renewable technologies is to have fossil fuel-powered backup generation (typically aero-derivative gas turbine peakers or fast-start combined cycles).

A more elegant solution to the supply-demand mismatch is energy storage, which is based on the principle of "time shifting". In other words, excess energy from renewable generation during times of off-peak demand is stored for later use at times of peak demand when the renewable generation comes up short. Currently available and commercially proven energy storage technologies are pumped hydro and compressed air energy storage (CAES) for large-scale applications (i.e., hundreds of megawatts or even a gigawatt or more) and lithium-ion batteries for much smaller scale uses. Each technology has its advantages and disadvantages, which have been discussed elsewhere.

Conventional CAES technology uses relatively cheap electric power during times of low demand to compress air into a reservoir where it is stored at high pressure. During times of high demand, when the price of electric power is relatively expensive, the stored air is expanded through a gas turbine with additional fuel burn. In other words, more megawatt-hours of energy is generated than stored. In order to improve fuel efficiency, stored air is preheated in a regenerative heat exchanger by using the gas turbine exhaust. Thus, CAES economics rely on the price spread between electricity prices at peak and off-peak demand hours as well as fuel price to make money.

Although two CAES power plants have already been built and successfully operated for decades (one in Huntorf, Germany, and the other in McIntosh, Alabama), the technology has failed to catch on for

CGES 1



Compressed gas energy storage (CGES) system (IC: Intercooler, AC: Aftercooler, PH: Preheater, RH: Reheater, TEG: Triethylene Glycol Dehydration Unit)

widespread commercial deployment. One reason is the high initial cost for air-storage reservoir excavation and the unique, custom-design turbomachinery.

Cavern excavation cost can be avoided by utilizing depleted natural gas reservoirs. However, it has not been demonstrated that conventional CAES systems can safely use these ready-made "holes in the ground" to store the compressed air. This can be extremely dangerous since the natural gas left in the reservoir, or gas that may still be seeping into the reservoir from natural sources, could form a flammable mixture.

An alternative solution, which addresses these two concerns, is to utilize an existing and operational natural gas storage facility in a simultaneous capacity for energy storage. This is achieved by inserting a natural gas turbo-expander between the storage reservoir and the natural gas pipeline so that each time natural gas is delivered to the latter, saleable electric power is generated and supplied to the grid.

BASIC IDEA

The CGES concept is based on an analogy to the well-known CAES technology described in the introduction. Northwest Natural Gas Company's Gill Ranch Storage facility in Madera, California, has two of the major CAES components already in place, namely the compressor train (comprising five units with variable frequency drives (VFD), inter- and after-coolers – basic fin-fan air coolers) and the storage space (depleted natural gas reservoirs). Thus, the present concept simply comprises the addition of a turbo-expander train to transfer the existing natural gas storage facility into a C<u>G</u>ES power plant. The resulting power plant is analogous to C<u>A</u>ES with the main difference being the storage medium (i.e., natural gas instead of air).

The heart of the CGES concept is a turbo-expander with gas heaters. A two-stage turbo-expander (TE1 and TE2) with a preheater (PH) and reheater (RH) generates power by expanding the gas from the reservoir to the entrance of triethylene glycol (TEG) dehydration unit. The variation between different implementation options is in the manner of gas heating and the temperature to which it is heated. The primary reason for gas heating is to prevent the formation of liquids within the expander. The natural gas from the reservoir is at 2,500± psia and approximately 100°F. If the expansion from the storage to the pipeline pressure (around 900± psia) were done without any preheating, gas temperature at the end of the expansion would be -18°F (very common in refrigeration plants); not desirable in this case due to liquid water and hydrate formation. An additional benefit of gas preheating upstream of each turbo-expander stage is increased power output due to increased *availability*. Without any preheating, the turbo-expander would generate about 8 MWe (320 MMSCFD gas flow); with preheating to 200°F at each stage, the power generation is about 11 MW with about 130°F at the end of the expansion.

There are essentially *three* ways to preheat and reheat natural gas during the expansion process:

1. Stored energy (adiabatic – no fuel burned, no emissions)
2. Waste heat from a process source or a prime mover exhaust
3. Fired heat exchanger (essentially, a furnace with a heater coil – what is shown in Figure 1)

Adiabatic CGES – Option 1

Adiabatic CGES is the "green" option; no gas is burned to accomplish preheating during expansion (i.e., zero emissions). The source for heating is the energy from compressor inter- and after-coolers. Since compression and expansion take place at different points in time, the energy from the former is stored in a *thermal storage system* (TSS) for use during the latter. In particular,

1. Heat energy recovered from the hot gas in compressor gas coolers is transferred to a storage medium and sent to a storage tank ("hot" tank).
2. When needed, heat energy from the "hot" tank in the form of the storage medium is sent to turbo-expander gas heaters to heat the natural gas.
3. Depleted (energetically, that is) storage medium from the gas heaters is sent to another storage tank ("cold" tank).
4. The charge-discharge loop is completed when the storage medium from the "cold" tank is sent to the gas coolers during compression.

The key design question in Option 1 is the selection of storage medium. There are three possibilities:

1. Molten salt (e.g., mixture of near-eutectic sodium nitrate ($NaNO_3$) and potassium nitrate (KNO_3) in 60/40

(% weight) ratio)

2. Heat transfer fluid (HTF) such as *Therminol*
3. Water

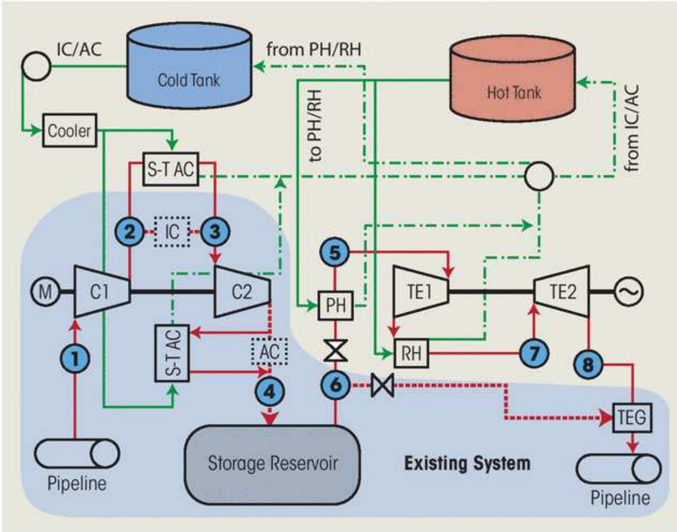
Natural gas temperatures during compression and expansion (between 100°F and ~300°F) preclude the molten salt option due to its high freezing point ~425°F. Furthermore, the same temperature range (100-300°F) makes water at relatively low pressures (100 to 150 psia) a suitable storage medium vis-à-vis HTFs.

The system is described in Figure 2. During compression/storage (charge) phase, original fin-fan gas-to-air coolers (IC and AC) are bypassed. Heat from the compressed gas is transferred to water in shell-and-tube heat exchangers, S-T IC and S-T AC. Heated water is sent to the "hot" storage tank.

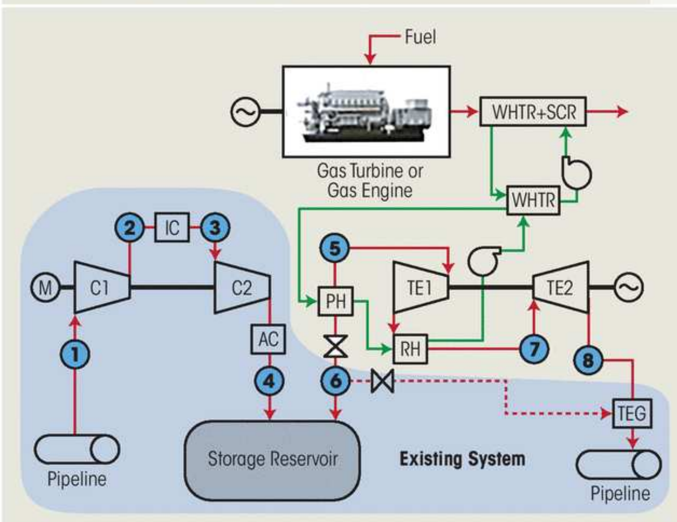
"Cold" water for S-T IC and S-T AC comes from the "cold" storage tank. However, since the temperature of water in that tank (~145°F) is hotter than the cooling requirement (120°F), it is first cooled to about 90°F in a mechanical-draft wet cooling tower (COOLR). The cooling tower requires make-up water to replenish losses due to blowdown and drift. This might be an issue in a desert environment. Other possibilities exist; e.g., fin-fan coolers that can operate at night to cool the water in the tank or even a heat pipe.

During expansion/discharge phase, hot water from the "hot" storage tank (230-235°F) is sent to expander gas heaters (PH and RH) to heat the natural gas to 200°F. "Cold" water from PH and RH (~145°F when mixed) is piped to the "cold" storage tank.

Adiabatic CGES (COOLR: Wet Cooling Tower) 2



CGES with prime mover 3



CGES with Prime Mover – Option 2

Since there is not a source of readily available waste heat (e.g., a chemical process plant) near the Gill Ranch facility, a possible solution is to include a gas-fired prime mover (i.e., a gas turbine or a reciprocating gen-set) with exhaust heat recovery along with the turbo-expander. The system is shown in Figure 3 schematically.

Preheating and reheating (to 200°F) is accomplished by hot water in shell-and-tube heat exchangers (STHX). Circulating water is heated in a separate STHX (WHTR in the diagram) using hot water from the waste heat recovery unit (WHRU) – basically, a combined cycle *heat recovery steam generator* (HRSG) without evaporator and superheaters.

A prime mover (e.g., an aero-derivative gas turbine (GT) or reciprocating engine gen-set) generates electric power and exhaust heat, which is recovered in the WHRU, which also contains the *selective catalytic reduction* (SCR) system to reduce NOx and CO in the exhaust gas to regulated levels.

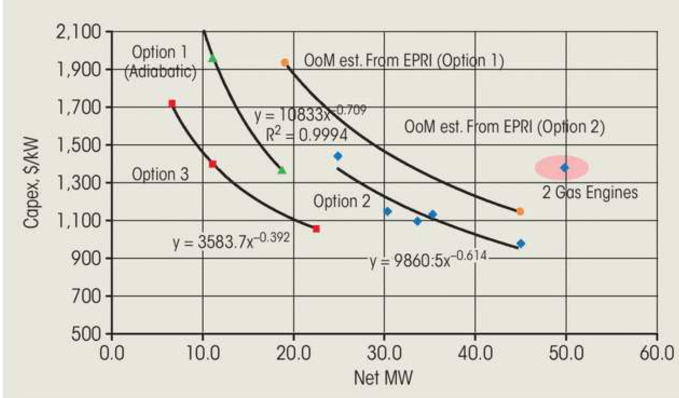
CGES with Line Heaters – Option 3

The third option involves turbo-expander gas preheating and reheating using gas-fired water-bath heat exchangers similar to those already in place to heat the natural gas from the storage reservoir prior to entry to the dehydration unit (see Figure 1 on page 27).

Obviously, from a high-level thermodynamic perspective, Options 2 and 3 are equivalent to each other in functionality; namely, *burning natural gas to heat natural gas*. The difference lies in the *exergetic* use of natural gas burned, which in Option 2 is superior to that in Option 3 via generation of electric power. They also suffer from the same drawback: emission of pollutants such as NOx, CO, CO_2 and UHC.

Note that, water-bath heat exchangers are limited in temperature (not much more than 200°F in the water-glycol bath). For higher gas temperature to enable higher turbo-expander output (discussed in detail below), a special-design fired heater with a heat transfer fluid (HTF) replacing water-glycol can be used.

Capex of CGES options 4



PERFORMANCE METRICS

The following performance metrics are used for the evaluation of the aforementioned CGES options:

Net Power Output – Reservoir Discharge

Net power output during reservoir discharge is arrived at by subtracting the plant auxiliary load from the generator (gross) output of the turbo-expander and prime movers (if present). The generator output is measured at the generator terminals and is inclusive of generator and gearbox losses. The auxiliary losses are:

- Feed and circulation pumps (0.25% of gross output)
- Cooling tower fans
- GT auxiliary (0.5% of GT generator output)
- Turbo-expander auxiliary (0.5% of TE generator output)
- Miscellaneous (0.1% of gross output)
- Transformer (0.5% of gross output)

Primary Energy Efficiency (PEE)

This is the ratio of net plant energy output to the sum of compression energy input (electricity consumed by the motors)

Comparison of CGES options Table 1

		CGES - Lowest LCOE Cases			
		Adiabatic	Gas Turbine	Gas Engine	Fired Heater
Gas Temperature	F	200	200	200	200
Expander Flow	MMSCFD	192	224	192	192
Expander	MW	6.7	7.8	6.7	6.7
Prime Mover	MW	0.0	22.0	18.5	0.0
Total Net	MW	6.6	29.4	24.8	6.6
Expander/Total		1.0	0.3	0.3	1.0
Prime Mover / Exp.		0.0	2.8	2.8	0.0
Generation	MWh	14,781	56,328	55,659	14,781
Capacity Factor		26%	22%	26%	26%
Compressor	MW	-26.3	-26.3	-26.3	-26.3
Exp./Comp. Hours		1.7	1.4	1.7	1.7
Primary Energy Efficiency		41.8%	25.5%	36.2%	31.9%
Capex	MM$	$18.9	$34.8	$35.8	$11.3
Specific Cost	$/kW	$2,860	$1,150	$1,440	$1,710
LCOE	$/MWh	$226	$168	$164	$196


TECHNICAL INSULATION
New ROXULASSIST
online calculation tool
Industrial insulation shaped by experts.
Experience our knowledge with ROXULASSIST®, the online thermotechnical calculation tool
This expert tool puts our extensive consulting skills and 75+ years of experience in the field of industrial insulation at your fingertips. Together, with our sustainable ProRox® stone wool solutions, we can help you deliver the highest available protection against fire, heat, noise and energy loss while also reducing CO_2 emissions. Do the math and calculate the ideal insulation thickness with ROXULASSIST, our free online expert calculation program. Sign up at www.roxulassist.com or call (800) 265-6878 to help your business shape up.
ROXULASSIST is available free of charge at www.roxulassist.com
EXPERT TOOL
ROXUL
www.roxul-rti.com
For info. http://powereng.hotims.com RS# 21

and the electric energy equivalent of the fuel consumed (in LHV) in fired line heaters (Option 3) or prime movers (Option 2). In the absence of fuel consumption during generation phase (or if it is ignored), it is also known as the **roundtrip efficiency** (RTE), which can be considered as a "pure" storage efficiency.

In mathematical terms, PEE is given as (with t_e and t_c as times of expansion and compression, respectively, in hours)

$$PEE = \frac{W_{net}}{W_{comp} + (\bar{\eta}_{sys} \cdot FC_{LHV}) \cdot t_e} \quad (1)$$

$$W_{net} = (\dot{W}_{exp} + \dot{W}_{GT/GE} - \dot{W}_{aux}) \cdot t_e \quad (2)$$

$$W_{comp} = \dot{W}_{comp} \cdot t_e \quad (3)$$

$$FC_{LHV} = \dot{m}_{fuel} \cdot LHV \quad (4)$$

Note that for $t_e = t_c$, Equation 1 becomes

$$PEE = \frac{\dot{W}_{net}}{\dot{W}_{comp} + \bar{\eta}_{sys} \cdot FC_{LHV}} \quad (5)$$

The rationale for Equation 1 is this: The fuel burned in line heaters or prime movers for expander gas preheating and reheating is natural gas, which would otherwise be sent to the pipeline and, ultimately, to a (unknown) power plant for power generation. According to a recent article in *Electric Power & Light* magazine, average heat rate of gas fired power generation in USA is 8,430 Btu/kWh (HHV). In terms of LHV, this is equal to

η = 3,412/8,430 x 1.109 = 44.9%.

As such, it is a reasonably good representative system efficiency (average) to be used in the denominator of the formula on the right-hand-side of Equation 1.

Fuel Efficiency (FE)

Fuel efficiency is the ratio of the net energy generation to the sum of fuel equivalent of total compression energy consumption during reservoir charge and the fuel consumed (in LHV) in fired line heaters (Option 3) or prime movers (Option 2). The average gas-fired generation efficiency defined above (44.9%) can be used to calculate the fuel equivalent of compression power. In other words, the electric power from the grid to drive the charge compressors is assumed to be generated in a generic gas-fired power plant with an efficiency equal to the average of all gas-fired generators. Thus

$$FE = \frac{W_{net}}{\frac{W_{comp}}{\bar{\eta}_{sys}} + FC_{LHV} \cdot t_e} \quad (6)$$

Note that for $t_e = t_c$, Equation 6 becomes

$$FE = \frac{\dot{W}_{net}}{\frac{\dot{W}_{comp}}{\bar{\eta}_{sys}} + FC_{LHV}} \quad (7)$$

CAPEX AND LCOE

Capital expenditure (capex) is the total outlay of money by the plant owner from the moment the NTP (notice to proceed) is signed to the moment the keys to the plant are handed over after all the acceptance and commissioning tests have been successfully performed. Capex calculation approach utilized in this study follows well-known cost estimate and roll-up conventions such as capacity factoring and cost factoring. The cost roll-up used herein for each CGES system option is as follows:

1. Installed Equipment Cost (E + M + L)
   a. Unit Cost (E)
   b. Installation Materials (M)
   c. Installation Labor (L)
2. Total Installed (Direct) Costs
3. Indirect Costs (% of Total Direct)
   a. Engineering
   b. Contingency
   c. Contractor's Mark-Up
   d. Owner's Costs
4. Total Capex (Direct + Indirect)

Equipment unit cost is obtained from direct vendor quotes (e.g., turbo-expander OEM budgetary price quotes), from Thermoflow, Inc.'s **PEACE** (Plant Engineering And Construction Estimator) software program, which is a companion to Thermoflow's **GT PRO** and **Thermoflex** heat balance software, and industry publications such Gas Turbine World (GTW) 2013 Handbook.

Installed cost factor, CFINST is

$$CFINST = \frac{E+M+L}{E} = 1 + \frac{M+L}{E} \quad (8)$$

and its range is 1.5 to 2.0, i.e., 50% to 100% on top of the bare equipment cost, with a mean/median of 1.75.

If the cost of a plant or a particular piece of equipment is known from a previous job or project (or some other source), it can be used to make an order of magnitude estimate via **capacity scaling**:

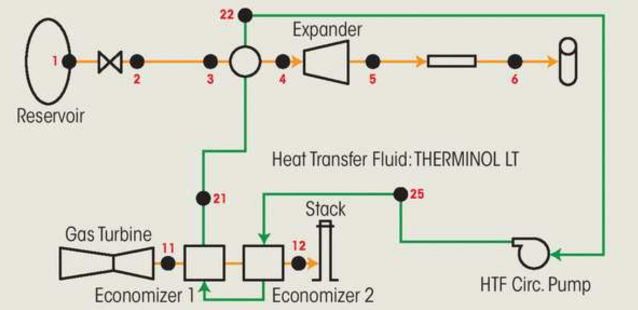
$$C = \left(\frac{Q}{Q_0}\right)^{\alpha} x\, C_0 \quad (4)$$

where C_0 is the known cost of a known piece of equipment or plant with a representative capacity of Q_0 (e.g., MW rating of a power plant) and Q is the capacity of piece of equipment or plant under study. The cost of the latter, C, can thus be referenced to its *scaled* capacity via the *scaling exponent* α. The most commonly used exponent is 0.6 from the well-known "six-tenth rule of thumb". Exponents for different equipment and plant types are available. From the data in Gas Turbine

CGES cost and performance with two different gas turbines with DLE combustors. Note that the capex numbers do not include the re-cylindering cost to reduce energy consumption during reservoir charging. Table 2

Gas Turbine	Output (MW)	Efficiency	Fuel (lb/sec)	TE Output (MW)	Total GT+TE	Capex ($MM)	$/kW	NOx
GT A	21.5	36.5%	2.9	14.8	36.3 MW	$53.70	$1,479	<15ppm
GT B	18.4	33.8%	2.635	14.8	36.3 MW	$49.30	$1,512	<42ppm

Flow diagram and heat and mass balance data for the expansion process 5



	Case		
	Gas Flow	MMSCFD	541.9
	HP Expander Gen.	kW	14,750
	Lp Expander Gen.	kW	
	Auz. Power	kW	489
	HX #1 Duty	Btu/s	30,465
1	Flow	lb/s	282.4
2	Pressure at Reservoir	psig	2,477
3	Temperature at Reservoir	F	95.0
4	Pressure at HX Inlet	psia	2,465
5	Temperature at HX Inlet	F	85.0
6	Pressure Inlet To Turboexpander	psia	2,432
7	Temperature Inlet To Turboexpander	F	238.6
8	Pressure Exit of Turboexpander	psia	857
9	Temperature Exit of Turboexpander	F	116.5
10	Pressure at Pipeline	psig	730
11	Temperature at Pipeline	F	115.5

	Case		
	GT	kW	21,479
	HTF Circ. Pump	kW	122
	GT Fuel Flow	lb/s	2.90
	GT Fuel Temp.	F	365
	Economizer Duty	Btu/s	31,182
11	Flow	lb/s	150.1
11	Temperature	F	1,015
12	Temperature	F	218.8
21	Flow	lb/s	180.5
21	Pressure	F	343.1
21	Temperature	F	449.4
22	Flow	lb/s	180.5
22	Pressure	psi	231.3
22	Temperature	F	125.8
25	Flow	lb/s	180.5
25	Pressure	psi	350.0
25	Temperature	F	127.0
26	Flow	lb/s	
26	Pressure	psi	
26	Temperature	F	
	Total Net Output		

World (GTW) 2013 Handbook, the exponent α is calculated as 0.713 for gas turbine combined cycle power plants.

Capacity scaling method is used to estimate ("order of magnitude") the capex of CGES options 1 and 2 from the cost estimates in a recent EPRI report on CAES [4]. See the section "CAES Performance-Cost" in the Appendix. Accordingly, the estimated capex for Option 2 (~650 MMSCFD gas flow, 45 MWe net) with a gas turbine or gas engine is **~$1,150/kW** (-30% to +50% accuracy). Similarly, the estimated capex for Option 1 is **~$2,000/kW** (-30% to +50% accuracy).

A more detailed but still study/budget level (-20% to +40% accuracy) capex roll-up is used in this study. Some key assumptions are as follows:

- Gas turbine, HRSG, major heat exchanger, pump, CEMS, DCS, electrical unit costs are from PEACE; installation costs are also estimated from PEACE results.
- Turbo-expander unit cost is from OEM quote; 25% installation M+L is assumed.
- Line heaters are estimated from capacity scaled invoice prices.
- Piping cost is set to 10% of bare equipment cost. Note that this is a minimum – in some cases, it can be as high as 80%. In fact, an examination of Gill Ranch numbers show that piping was nearly 25% of the total installed cost, which suggests 35-50% of bare equipment cost.
- As a default, CFINST of 1.40 is used (based on PEACE estimates). (Gill Ranch installed cost data suggests close to 2.0.)
- Gas engine installed cost is assumed to be $800/kW
- Engineering is 8% of TIC (Total Installed Equipment Cost – Direct Costs)
- Contractor's mark-up is 6%

The chart below provides a graphical summary of the capex data.

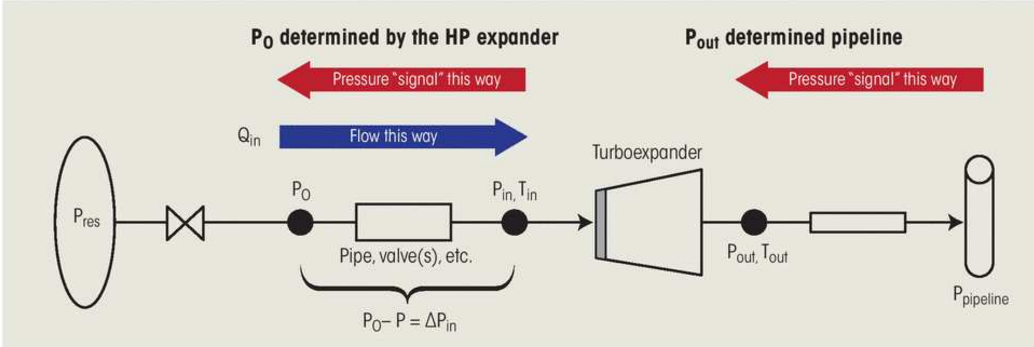
A *Monte Carlo* simulation is done using **Oracle® Crystal Ball** software to gauge the uncertainty in capex estimates. Inputs to the cost roll-up such as equipment costs, installed cost factors, etc. are represented as uniform distribution with minimum and maximum limits, e.g. ±20% for expander cost, piping cost factor 0.10 to 0.35, etc. Based on the M-C simulation results, the capex estimates in this study are considered to have an accuracy of -20% to +40%, with the higher end being more likely. As such, this particular uncertainty band is in accordance with "study" or "budget" estimate classification. Accordingly, a project contingency of 15% is applied to the final roll-up along with 10% for owner's costs.

Levelized Cost of Electricity (LCOE) is a widely used metric for comparing power plant system alternatives. Traditionally, it combines a power generation system's ownership costs (capital and operating) and thermal performance (output and efficiency). This metric is useful when comparing power generation alternatives that use *similar* technologies. The standard formulation of LCOE is the sum of capital, fuel, and operations and maintenance (O&M) costs of plant ownership.

To evaluate CGES options, emissions and power purchased during compression are also included in the mix. One can also add firm capacity and annual energy generation replacement costs to the list (and in pure power generation cases one must). Due to the unique economic drivers for the CGES project, replacement costs are ignored.

Recent information suggests $15/ton is a good value for CO_2 emissions allowance price to use in LCOE calculation. (According to a Bloomberg news article in 2014, California sold 19.5 million carbon allowances at auction for $11.48 each. Futures based on permits that can be used to cover emissions in the same year settled at $12.15 a ton.)

Turbo-expander flow-pressue balance 6



Prices of annual sulfur dioxide (SO_2) and summer seasonal nitrogen oxides (NOx) emissions allowances obtained from the EPA's Clean Air Interstate Rule (CAIR) and the NOx Budget Trading Program (NBP) are seen to have fallen dramatically in recent years. – i.e., from ~$800/ton in 2008 to about $16/ton by 2011 for NOx. This drop is attributed to several factors such as regulatory changes, environmental control systems such as FGD (Flue Gas Desulfurization) and SCR adopted by coal plants and lower coal-fired power generation.

It is impossible to predict the future trends. Thus, a representative $20/ton value is used herein. The same price is applied to CO emissions as just a "place holder" to quantify it in dollar terms. (As it turned out, whether NOx is $800 or $20 per short ton was rather immaterial.)

The following assumptions are used in the LCOE calculations

Capital charge factor (β) of 15%

- Levelization factor of 1.25
- Fuel price of $4/MMBtu(HHV)
- Capex and performance as described above
- Electricity purchase price of ¢1/kWh (assuming that compression is done during the time of excess capacity and low demand – this can even be a *negative* number – i.e., one is *paid* to use electricity!)
- Emissions costs as described above
- O&M costs for GT and gas engine (GE) from Table 1-5 in Ref. [8]
- Option 1 fixed/variable maintenance costs $1/kW and 1 mils/kWh (estimate)
- Option 3 fixed/variable maintenance costs $1/kW and 2 mils/kWh (estimate)
- SCR O&M 1 mils/kWh for GTs, 2 mils/kWh for GEs
- Turbo-expander O&M from the OEM quote

Average annual ambient temperature of 77°F is assumed for the generation runtime (peak hours during the day – especially hot in the summer). A 4% power output lapse is applied to the GT (small rise in heat rate is ignored). Turbo-expander and GE are assumed to be unaffected. This should take care of higher output during much colder times and lower output during much hotter times.

Note that inlet conditioning (e.g., chillers or evaporative coolers) for the GT for hot day power augmentation is not considered. This should be evaluated carefully during the detailed design phase.

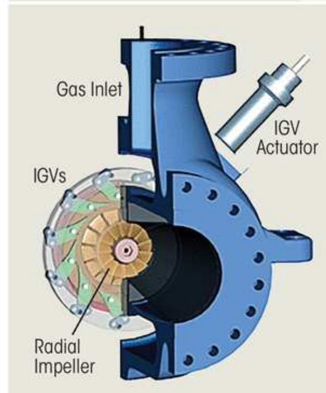
PHASE I ANALYSIS

Three CGES options described above have been evaluated in terms of performance, emissions, capex and LCOE.

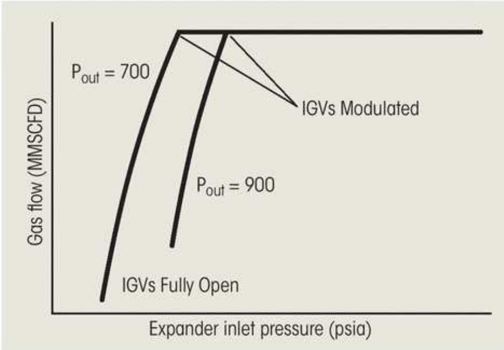
With the current natural gas compressor station in Gill Ranch and 4 hours of charge operation, natural gas turbo-expander generation entitlement is approximately 15,000 MWh.

From a capex and LCOE perspective, the optimal configuration is with the ratio of compression hours to generation hours of 1.7. Ignoring losses, this is the ratio of compressor inter/after cooling duty to expansion pre/reheat duty, which enables the "green" adiabatic CGES option. Note that, due to its high exhaust temperature/energy, with one gas turbine and WHRU, up to 224 MMSCFD gas can be heated to 200°F so that the lowest LCOE ratio is 1.4.

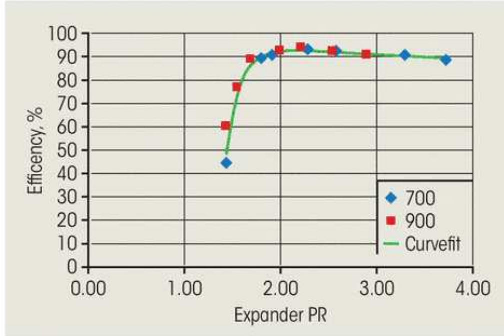
Radial turbo-expander with IGVs (Courtesy: Atlas Copco) 7



Turbo-expander flow-inlet pressure map 8



Turbo-expander efficiency as a function of expander PR 9



Adiabatic CGES (with thermal energy storage using hot pressurized water as storage medium) and CGES with line heaters are both rated at ~6.5 MWe at a capex of $2,860/kW and $1,710/kW, respectively. The PEE of the adiabatic CGES option is 41.8% with an LCOE of ~¢23/kWh. In comparison, adiabatic CAES at the same size would cost ~$3,750/kW with a PEE efficiency of 37.5% ($2,600/kW if air compressor and storage construction costs are ignored).

Combining CGES with a prime mover provides significant improvement in terms of performance, capex and LCOE. Emissions impact on LCOE is negligible and emitted quantities are well below permitting thresholds. In particular, with a 45+% efficient, 18 MW gas engine, additional 40,000 MWh generation is possible at a capex of $1,440/kW and LCOE of ~¢16/kWh with a PEE of 36% (cf. around 40% for CAES – adiabatic and ~20% GT plus expander options).

Key findings from the Phase I analysis are summarized in Table 1 on page 30.

Option 1 (adiabatic) was originally

picked to minimize impacts from an environmental standpoint and yet provide a safe reliable energy storage project that met the utility requirements. The inherent technical problem associated with this process is the ability to capture and hold the necessary hot water from the heat of compression of the compressors. The time lag from when compressor injection occurred and when stored energy is needed during discharge required very large insulated storage tanks. Also the low temperature differential between the temperature of the compressor energy and the needed gas temperature caused the heat exchangers to be very large and very expensive. This resulted in significantly larger specific cost and LCOE vis-à-vis the other two options and Option 1 was eliminated from further consideration.

Economically, Option 2 with a prime mover and exhaust gas heat recovery was the clear winner. The type of the prime mover, i.e., gas turbine or gas engine, became the decision to be made. Note that the natural gas flow rate from the storage reservoir can be increased up to ~650 MMSCFD.

At 542 MMSCFD, the ratio of compressor inter/after cooling duty to expansion pre/reheat duty is 1.0. Thus, at the beginning of the study, this was determined as the maximum-flow case. Since a single gas engine is not capable of providing the heat necessary for the 542 MMSCFD of gas discharged, this option was not considered due to complexity and large capital costs.

Consequently, Option 3, with the gas turbine and exhaust gas heat recovery, was selected for study in Phase II.

PHASE II ANALYSIS

Two small industrial gas turbines from two different OEMs were selected for this phase, which will be referred to as GT A and GT B. The system in Figure 3 on page 28 was rigorously modeled in Thermoflow Inc.'s heat balance simulation software, **Thermoflex**. Due to the very high pressure of natural gas, REFPROP (NIST) property model was used in turbo-expander calculations. Gas turbine performance data was directly obtained from the built-in "engine library" in Thermoflex.

Table 2 on page 32 summarizes the CGES cost and performance comparison with the two gas turbines.

Table 2 CGES cost and performance with two different gas turbines with DLE combustors. Note that the capex numbers do not include the re-cylindering cost to reduce energy consumption during reservoir charging.

Gas turbine A is an aero-derivative gas turbine, which is sized to provide the necessary heat to the turbo-expander and produces approximately 21.5 MW, with the turboexpander producing approximately 14.8 MW. Total gross electrical output of the CGES facility utilizing this gas turbine is approximately 36.2 MW.

The exhaust temperature of gas turbine B, also an aero-derivative GT, is not high enough to bring the gas inlet temperature to the turboexpander to the same level as GT A. As a result, the turbo-expander power output is lower. Gas turbine B would produce 18.4MW, with the turbo-expander producing approximately 14.3 MW. Total net electrical output utilizing GT B is approximately 32.6 MW. While this gas turbine could support most of the flow conditions, it is smaller and the process is less efficient.

The heart of CGES compression-expansion cycle performance simulation is the turbo-expander model. Due to varying storage reservoir and pipeline pressures, the selected equipment is expected to run in a wide range of operating conditions dictated by upstream and/or downstream pressures. Reservoir pressure is a linear function of the inventory and changes from 1,750 to about 2,500 psig between 35% and 100% full. The operating mechanism of the single-stage turboexpander is described in Figure 6 on page 34.

Note that, originally (as depicted in Figures 1-3), a two-stage turbo-expander with a reheat heat exchanger between the high-pressure (HP) and low-pressure (LP) sections was considered. Cost-performance evaluation indicated that increased complexity and cost were not worth the performance gain so that it was decided to go with a single-stage turbo-expander as shown in Figure 6.

From an operation and performance modeling perspective, a turbo-expander can be considered as an "orifice" in the flow-path of the fluid in question. Thus, the system between the reservoir and the pipeline can be imagined to be a long natural gas pipe with many obstacles along the way, e.g., miscellaneous valves, equipment, pipe joints and tees, etc. along with the turbo-expander (TE) itself.

Pipeline pressure dictates the TE discharge pressure. TE inlet pressure is dictated by the gas flow and the "swallowing capability" of the TE, i.e., the inlet flow area. The reservoir pressure is determined by the gas inventory, which will obviously diminish over time when the expansion process is active.

Consequently, the pressure-flow balance between the reservoir and the TE inlet is established by the flow-control valve. A second degree of freedom is provided by the TE inlet guide vanes (IGV). As shown in Figure 7 on page 34, IGVs act as stator nozzle vanes as well as inlet area adjusters.

The maximum variable inlet guide vane area is 125% of the required area at the design (typically, best efficiency) point. At low inlet pressures, the IGVs are fully open, and the flow is the maximum that will pass through at this fully open position. At high inlet pressures, IGVs are closed to keep the flow at the maximum 542 MMSCFD.

From basic gas-dynamic theory, expander inlet flow-pressure relationship is

$$\frac{\dot{m}\sqrt{T_{in}}}{P_{in}} = C_q \cdot A_{in} \cdot f\left(\gamma, PR_{exp}\right) \quad (10)$$

where γ is the specific heat ratio of the gas, PR_{exp} is expander pressure ratio and C_q is a flow coefficient. For a TE with IGVs, $C_q \cdot A_{in}$ can be written as

$$C_q \cdot A_{in} = \frac{Y_{IGV}}{100} \quad (11)$$

where Y_{IGV} is the percent-open position of the IGVs. Thus, using Equations 10 and 11, OEM-provided TE characteristics can be translated into Y_{IGV} by assuming that the design point is 80% open and maximum opening is 100%, i.e., equivalent to 100% and 125% open, respectively. Thus, any desired flow-pressure combination for given P_{in} and P_{out} (i.e., PR_{exp}), Tin (from the inlet heater) and Y_{IGV} can be determined via

$$\frac{\dot{m}\sqrt{T_{in}}}{\frac{Y_{IGV}}{100}P_{in}} = 6.75 - \frac{484.59}{\left(1 + 0.00824 \cdot PR_{exp}\right)^{425.91}} \quad (12)$$

Gas mass flow rate from Equation 12 along with P_{in} and P_{res} determines the flow control valve opening. Expander efficiency is also determined from OEM-provided data via a curve-fit (see Figure 9 on page 35). Equation 12 and Figure 9 curve-fit are implemented in Thermoflex gas expander module to evaluate the operating envelope of the CGES system, which is shown in Figure 10 on this page.

The CGES system in the photo on page 26 includes the following major equipment:

- High-pressure turboexpander with generator
- Preheater (shell-and-tube heat exchanger)
- Gas turbine with generator.
- Heat recovery steam generator (includes the SCR)

Interconnecting gas lines are run from the existing gas withdrawal process to the turbo-expander preheater, and from the outlet of turbo-expander back into the existing gas withdrawal process downstream of the pressure control valve and upstream of the existing dehydrators and then on to the existing pipeline (no new pipelines exits the site).

Existing utilities for air, water, and sewer in Gill Ranch are adequate to be extended to the CGES component.

One of the site's redundant transformers is used for the step-up of the new generators' output to the transmission voltage. Transformer redundancy is maintained as the compressors are not operated at the same time as the turboexpander.

In an effort to further increase the overall efficiency of the storage process, a discretionary investment to re-cylinder all five of the existing compressors is considered. This modification is expected to reduce the compressor load by approximately 22% or 5 MW when utilizing four compressors (one compressor held in reserve) at an additional cost of about \$4.5 million. Thus, with the following assumptions

- Charging for 7.5 hours at 320 MMSCFD
- Discharging for ~4.5 hours at 542 MMSCFD based on the performance data in Figure 5 on page 33, a primary energy

efficiency of 42.4% is expected. Note that, in order to facilitate an "apples-to-apples" comparison to a competing storage technology with no fuel-burn, the roundtrip efficiency of the CGES system with re-cylindered compressors is

Turbo-expander Generation = 14,750 x 4.428 = 65,313 kWe

Compressor Consumption = 15,700 x 7.5 = 117,750 kWe

Roundtrip Efficiency = 65,313 / 117,750 = 55.5 percent

CONCLUSION

Gill Ranch is an existing 20 BCF gas storage operation that includes five electric-drive gas compressor units used to compress natural gas for injection into the existing natural gas storage reservoirs. The compressor station receives electricity from an electric power line that runs from PG&E's Dairyland-Mendota 115 kV transmission line to the substation at the compressor site.

When needed, withdrawal of the gas is extracted from the reservoir at an average flow rate of 542 MMSCFD. During the withdrawal process, gas is heated, pressure is reduced over a set of valves, dehydrated and put in the pipeline at approximately 700 psig.

By incorporating a turboexpander generator train into the expansion part of the cycle, the natural gas storage facility will be transformed into an energy storage facility.

The proposed compressed gas energy storage system will produce electricity upon withdrawal of the high-pressure gas that was previously injected by the electric-drive compressors. The CGES system also includes an aero-derivative gas turbine for a nameplate rating of 35 MWe with a primary energy efficiency of 42.4 percent. Roundtrip efficiency based on turbo-expander only (i.e., excluding gas turbine) is 55.5%.

Noting that Gill Ranch is an active natural gas storage operation, one way to look at the power generation part of the thermal energy storage cycle is as a highly-efficient combined cycle power plant.

This is so because, whether CGES concept is realized or not, compression of gas into the storage reservoir is a given. The efficiency of the GT-TE combined cycle is 58%, which is at the same level as utility-scale F class technology. pe

Gill ranch CGES operating envelope 10