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EQUIPMENT AND COMPONENT ADVANCES IN THE APPLICATION OF SCO2 BRAYTON CYCLES FOR BOTTOMING CYCLES AT GAS COMPRESSOR STATIONS

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Abstract

For large gas turbines, a steam Rankine bottoming cycle is typically integrated to recover additional energy and maximize the overall performance of the facility. There are over 1,300 GT-driven gas compressor stations in North America and most operate as simple cycles that exhaust a significant amount of waste heat that becomes unused thermal energy. Since most compressor stations are remote and lack access to water, steam bottoming cycles are not practical. Some stations integrate organic Rankine cycle (ORC) systems into their facilities since they can operate effectively with air cooling, but that technology has failed to spread commercially due to the high costs and limited performance. Supercritical CO2 (sCO2) bottoming cycles are emerging as an improvement on current ORC systems since they are simpler systems which reduce both materials and installation costs and can deliver significantly improved performance.

An sCO2 bottoming cycle uses CO2 as the working fluid where the minimum cycle pressure is greater than the critical pressure. In an sCO2 Brayton cycle the CO2 is directly heated by the GT engine exhaust, passed through a power turbine to generate power, then flows to an air cooler before being compressed and returned to the heater after preheating in a recuperator. This cycle avoids the need for a thermal oil loop in similar ORC systems, and since the working fluid is supercritical, it cannot freeze. A comparison to ORC shows the potential of sCO2 bottoming cycles to recover about at least 20% more power than ORC. The additional potential power recovery and lower cost equipment results in an improvement to the expected return on investment (ROI). At favorable sites, lower CAPEX & OPEX is expected to be achievable. Testing of the key components of this system has been completed and initial commercialization efforts have begun.

Introduction

Gas turbine technology has earned widespread recognition in power generation and various industrial applications due to its efficiency, reliability and versatility. Most power generation plants initiate the process by combusting natural gas with compressed air and producing hot gas pressure that drives a turbine. This rotating turbine is linked to a generator to create electrical power, and this process is referred to as the Brayton cycle. To further optimize energy utilization, the integration of a steam Rankine bottoming cycle with gas turbines, known as a combined cycle, has become a prevalent practice especially in large gas turbines. This integration allows for the recovery of waste heat from the gas turbines that would otherwise be rejected, thus effectively generating additional power. In a conventional combined cycle, a gas turbine functions as the topping cycle, while the steam Rankine bottoming cycle operates at a lower temperature. The high-temperature exhaust gas from the gas turbine transfers its heat energy to water and steam in a waste heat recovery boiler, which is part of the bottoming cycle. These combined cycles significantly enhance overall efficiency, making power generation more sustainable and effective.

Gas turbines are commonly used in pipeline compressor stations due to their reliability and efficiency in driving the compressors of natural gas for pipeline transportation where maintaining gas pressure and flow over long distances is essential. Thus, optimizing the performance of these stations is very important as they are critical components of the energy infrastructure. With over 1,300 gas turbine (GT)-driven gas compressor stations in the United States alone, many situated in remote locations due to operational and safety considerations, the compact design and high power-to-weight ratio of gas turbines make them ideal for installation in such areas [1].

While the combination of gas turbines and steam Rankine cycles shows great promise in enhancing the energy efficiency of compressor stations, remote locations present unique challenges. These challenges arise primarily due to the lack of access to water for the steam bottoming cycles in the vicinity of remote compressor stations and the lack of adequate electric capacity to deliver power into the utility grid. To address these challenges, some stations have explored alternative solutions, such as integrating organic Rankine cycle (ORC) systems, that can effectively operate with air cooling and provide smaller capacity systems for grid interconnection. However, the commercial adoption of ORC technology has faced obstacles, including high costs and limited performance. Nevertheless, a promising advancement in energy recovery systems has emerged in the form of supercritical CO2 (sCO2) bottoming cycles. These innovative cycles offer significant improvements over current ORC systems, with simpler designs that reduce both materials and installation costs while delivering substantially enhanced performance.

This paper discusses the advancements in the application of sCO2 Brayton cycles as bottoming cycles, specifically in pipeline compressor stations, where they utilize gas turbine waste heat to generate additional power, and compares them to ORC systems.

Organic Rankine Cycle (ORC) Systems

The ORC is a common technology used for heat recovery from lower temperature geothermal heat sources from 100°C to 350°C. It is also used in waste heat recovery (WHR) to capture the heat from energy-intensive industrial processes such as those occurring at steel mills, refineries, furnaces, ovens, kilns, farming processes, and exhaust gases from gas turbines and power generation units, to name a few. This technology is now being employed on heat sources where the volume of heat is not sufficient to support the large size of steam turbine technology (below 25MW) and where the ORC becomes a viable alternative for power generation.

The system uses an organic working fluid in a closed loop where the fluid is pumped to a boiler, vaporized and expanded through a turbine that spins a drive to power a generator. After passing through the turbine, the vaporized gas is then passed through an air-cooled or water-cooled condenser to bring it back to a liquid state. The heating of the organic fluid is performed using a hot thermal oil loop which recovers the heat from the industrial process and then transfers it to the organic fluid in a closed-loop system. Both the heating and cooling sources do not come into direct contact with the working fluid. The working fluid of an ORC system is selected for a particular thermodynamic behavior that suits the cycle. The chosen fluids are usually higher molecular weight hydrocarbons, e.g. cyclopentane, isobutane, pentane, propane or HFCs, that exhibit boiling point temperatures lower than water, but have a high latent heat and density to improve the ability to capture the heat from the source. However, the ORC cycle requires the heat to be added to the working fluid when it is at high pressure and there are significant safety concerns related to the high flammability and the consequences of a high pressure leak within the WHRU so close to the combustion in the gas turbine. While implementing a direct heat-to-working fluid system would increase the efficiency of the cycle, the safety implications have so far been too onerous to take that step.

In order to reduce these safety concerns, a thermal heat transfer fluid is employed in an intermediate closed loop that is operated at low pressures. However, despite the near-atmospheric operating pressures these thermal oils are also flammable and require appropriate system design for mitigation/suppression, as well as maintenance in order to be operated safely at the prevailing cycle temperatures. In contrast, the sCO2 working fluid is naturally non-flammable and so those hazards of ORC systems are eliminated.

Commercial challenges with ORC systems

While the ORC technology provides various benefits due to the technology and its attributes, there are also commercial challenges that have to be addressed when developing projects. Several of these challenges are listed below. The focus here is on employing ORC on WHR applications.

- Gaining acceptance by clients for pursuing new opportunities and technologies that are outside of the core traditional business functions in their operations.
- Selecting client applications with both high load factor and high run times necessary to support positive project economics.
- Installing equipment within the boundaries of a client's current operations.
- Providing guarantees to the client regarding limitation of impacts to existing client operations – both during installation and after commercial operation.
- Potential impacts due to use of both organic materials and thermal oil, both of which are toxic and flammable materials.

- Grid infrastructure limitations due to both remote locations of many sites and also limited grid capacity at these locations.
- Incentives for Waste Heat to Power (WHP) are in place only in certain states. WHP is approved as a full renewable technology only in selected states. WHP is the terminology used for WHR in the language for these incentives.

In addition, ORC WHR systems typically require an intermediate fluid for heat transfer between the heat source and the organic working fluid. This intermediate loop generally uses a thermal oil which requires addressing additional operational considerations, additional equipment to install, and may also require the need for fire protection systems to prevent any possibility of fire in the thermal oil system.

Supercritical CO2 (sCO2) Bottoming Cycles

CO2 in the supercritical state has a higher density than organic materials as used in the ORC process and therefore has better heat transfer characteristics. These characteristics allow for a higher production of usable power from a similar amount of thermal heat input. This benefit can be realized as additional on-site power to offset utility demand requirements or as additional energy that can be sold back into the grid to the local utility or Independent System Operator (ISO).

The sCO2 system utilizes a direct heat recovery process from the heat stream to the working fluid, thereby resulting in an inherently safer and more cost effective means for WHR. SCO2 Bottoming Cycles, like other bottoming cycles, extract as much of the remaining energy as possible from the exhaust system of a gas turbine and use it as the heat input to a cycle that produces electrical power. Figure 1 provides a simplified demonstration of the basic heat transfer cycle. In the sCO2 Bottoming Cycle, the heat exchanger in the exhaust system transfers the recovered energy to a closed-loop, high-pressure system that consists of the following main basic components:

1) A Cooler, that can be air-cooled or water-cooled
2) A Compressor
3) Exhaust Waste Heat Recovery Unit (exhaust-to-sCO2 heat exchanger), or WHRU
4) A Turbine or Expander
5) An Electrical Generator, generally synchronous type

These components are shown in the second diagram in Figure 1 below.

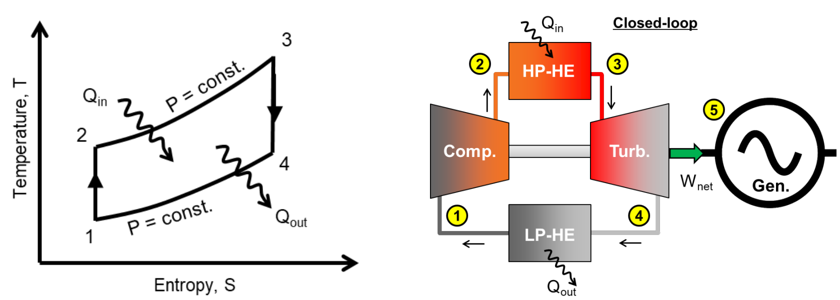

Figure 1. sCO2 Bottoming(Brayton) Cycles and main basic components [2]

In order to enhance the efficiency of the system, i.e. produce more electrical power from the same amount of heat, more complex cycles are used that retain more of the heat within the system and recycle it internally.

Following extensive evaluations that compared system efficiency and output power to system cost and footprint, Hanwha Power Systems (HPS) has selected the optimum configuration for WHR applications to be a Recuperated Pre-Heating Brayton Cycle. This cycle adds a recuperator to the system to recycle a significant amount of heat and also extracts extra heat by diverting some of the compressor discharge flow around the recuperator directly to a Pre-Heater in the gas turbine exhaust.

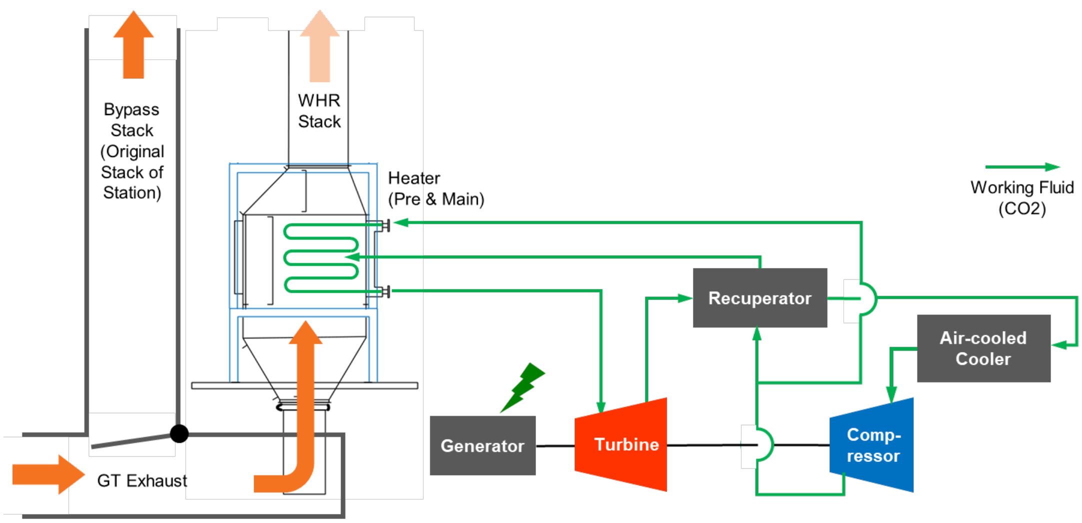

Figure 2. HPS sCO2 WHR Power System Configuration

Advantages of sCO2 bottoming cycles over ORC systems

Several advantages can be realized in an optimized sCO2 WHR cycle over competing systems in the power range of gas turbines present at typical natural gas pipeline compressor stations:

- Simpler system design, eliminating the intermediate thermal fluid and removing a significant potential flammable hazard thereby eliminating the need for costly fire suppression systems.
- No toxic or hazardous fluids are used in the system.
- Dependent on the working fluid chosen, and therefore, the low temperature required for condensation, no full system chiller is required to liquefy the process fluid. However, even if refrigeration equipment is not required for liquefaction, the cooling system is physically much larger because of the lower pressure and density at that point in the cycle.
- Reduced materials and installation costs due to the very high power density of the sCO2 turbomachinery, requiring a significantly smaller footprint and less construction materials.

- Most importantly, improved performance is achieved through eliminating the intermediate thermal fluid and its additional thermal exchange step through direct heating of the sCO2 fluid in the exhaust WHRU. This allows more heat to be recovered from the gas turbine exhaust to the power cycle and requires lower auxiliary load requirements for the system.

Testing of key components

An extensive test program was completed in 2021 at Southwest Research Institute (SwRI) in San Antonio, Texas, on a full-sized, but partial cycle prototype integrally geared compander sponsored by the Department of Energy under the Sunshot Apollo program over a 5-year period. This was further supplemented by HPS-funded extended testing to explore additional control features and off-design performance [11].

Prior to the test program at SwRI, a no load mechanical test was performed using a shop driver at the HPS production facility in Korea before shipping the unit to SwRI.

During the test program at SwRI, the following key operating parameters were demonstrated successfully:

- Full design flow, pressure rise and efficiency of the first stage compressor. A performance curve was also obtained through multiple test points from choke to surge at a range of inlet conditions close to the CO2 critical point.
- Full design discharge pressure of the second stage compressor (designed for only partial flow) at over 270 barA.
- Full turbine inlet pressure of 266 barA and temperature of over 705°C of the first stage expander. The expander wheel was designed for full flow, but with a partial admission nozzle to reduce the flow rate and limit the heat input due to the test loop heater and cooling capacity limits.
- Full speed.
- Dry gas seal operation at each stage's full operating conditions.
- Full radial and axial vibration and axial position monitoring.
- Full bearing temperature monitoring.

Additionally, information was obtained that will allow a more robust control system to be implemented due to factors found where CO2 close to its critical point creates a high level of measurement uncertainty.

The testing also included an extended run of over 12 hours to prove the reliability of the machinery while operating at the design conditions of a commercial Concentrated Solar Power (CSP) contract currently being produced with a Turbine Inlet Temperature of 600°C. Due to the Department of Energy (DoE) program budget limits, only a partial cycle could be replicated, but all the key design parameters were achieved.

Comparison of sCO2 and ORC Bottoming Cycles

Due to the relatively opposite characteristics of the two cycles, sCO2 is normally optimized for medium-to-high temperature heat sources and ORC is normally restricted to low-to-medium temperature heat sources. The basic features of the two cycles are initially compared in Table 1 below, before in-depth comparison in terms of general competitiveness, potential power recovery, cost (CAPEX and OPEX) and expected economics (IRR) is conducted to determine the best cycle for the application.

Table 1. Basic Features Comparison for sCO2 and ORC Cycles

No.	Feature	HPS sCO2	ORC
1	Cycle	Brayton	Rankine
2	Working Fluid	CO2 in supercritical state	Organic Fluid (Cyclo-pentane, etc.)
3	Working Pressure	High (> 200 barA)	Low (< 50 barA)
4	Competitive Heat Source Temperature	Medium-High Temp. (400 ~ 1,000 °C)	Low-Medium Temp. (200 ~ 500 °C)
5	Main Application	- Gas Turbine WHR - Industrial Process WHR - CSP - SMR	- Geothermal / Biomass - Reciprocating Engine & Gas Turbine WHR - Industrial Process WHR

Note. WHR: Waste Heat Recovery, CSP: Concentrated Solar Power, SMR: Small Modular Reactor

General Competitiveness Comparison

One of the main competitive applications of the two cycles is gas turbine WHR in natural gas pipeline compressor stations where water is rarely available in sufficient quantities and remote operation is required, so applying a steam turbine is not appropriate. For the WHR application at a compressor station, a general competitiveness comparison is presented in Table 2 below.

Table 2. General Competitiveness Comparison for Natural Gas Pipeline Gas Turbine WHR

sCO2 Power System	ORC Power System
Higher gross efficiency originating from supercritical working fluid feature *(utilizing gas with a liquid-like density)* especially for high temperature heat sources like GT exhaust	Inherently lower gross efficiency originated from organic working fluid feature especially for a high temperature heat source
No intermediate heat transfer required; heat transfer occurs directly from heat stream to the sCO2 system	Uses intermediate thermal oil heat transfer process which increases auxiliary load requirements, decreases operating efficiency and introduces additional design complexity as well as potential fire hazards.
Higher net power through lower auxiliary power consumption: - No additional power consumption for heat recovery - Relatively small power requirement for air-cooled cooler due to there being no phase change of CO2 in the Brayton cycle application	Lower net power through higher auxiliary power consumption: - Thermal oil pump power for indirect heat recovery - Larger power for air-cooled condenser due to organic fluid phase change by Rankine cycle : Heat rejection normalizing factor (Heat rejection per Net power produced) comparison 1) ORC a) GT WHR, typically 7.0 (W/ Fluid R245fa) : Estimated using the reference data [14] b) Geothermal, typically 11.1 (R134a) [15] 2) sCO2 : GT WHR typically 3.5
Simple and modularized power block	Complicated and spread out power block
Compact footprint and versatile layout options available	Larger footprint, plus requiring additional space for thermal oil intermediate loop
Lower $/kW of CAPEX estimated by comparable equipment cost, but reduced installation cost at high net power	Higher CAPEX in the range of typically $2,500/kW to $3,000/kW (equipment plus engineering & construction)
Lower OPEX estimated from elimination of thermal oil loop, reduced maintenance time and non-toxic working fluid	Higher OPEX resulting from indirect oil heat recovery, complicated system composition and toxic working fluid
Easy accessibility, non-toxic and inexpensive working fluid (CO2)	Highly flammable, fire hazard, toxic and expensive working fluid (cyclo-pentane) requiring a fire suppression system
Very suitable for compressor station WHR and high potential for commercialization & popularization	***Application to pipeline compressor station WHR has been very limited so far***

Potential Power Recovery Comparison

For a natural gas pipeline compressor station gas turbine WHR application, the net power recovery is compared in Figure 3 below. For sCO2, the recovered power is the calculated power based on HPS design experience and reference test programs. For ORC, the values are based on actual compressor station WHR reference projects in commercial operation and/or being constructed now, and based on public release from three ORC system suppliers [12, 13]. As seen in the figure, it is shown clearly that sCO2 power recovery is superior by at least 20% over ORC, indicating very strong potential for commercialization of waste heat recovery from pipeline compressor station gas turbines.

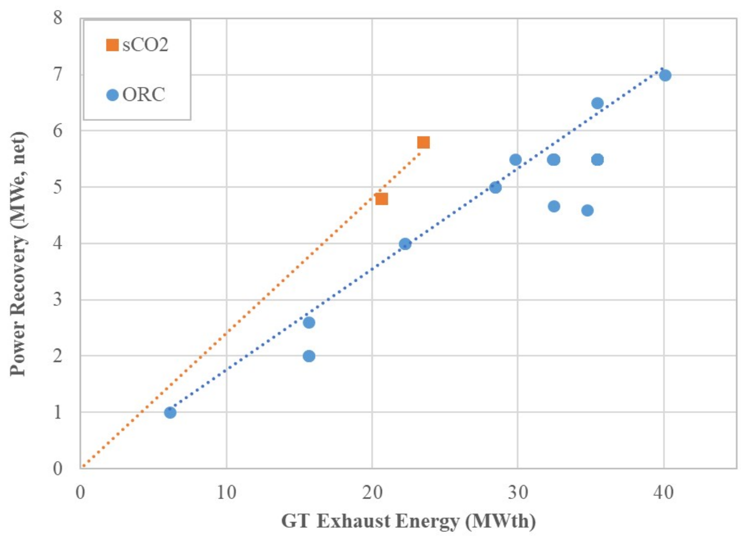

Figure 3. Comparison of power recovery from GT exhaust heat for sCO2 and ORC system

Cost Comparison (CAPEX, OPEX)

Based on the publicly released reference projects of compressor station ORC systems, CAPEX for 5 to 8MW class is estimated as $2,700/kW~$3,000/kW. CAPEX for 5 to 6MW class of sCO2 is targeted as $2,600/kW~$2,800/kW, which is comparable to ORC, even though this is for a first-of-a-kind sCO2 power system, mainly due to the sCO2 power system advantages such as high power density, compact footprint, modular power block construction and direct heat recovery, etc. After full mass production entrance, sCO2 WHR system CAPEX is expected to be less than $2,500/kW.

Regarding OPEX, based on Northern Border Pipeline ORC 5.5MW project in 2007, ORC OPEX is estimated to be about $6.8/MWh based on current data, while the OPEX for an sCO2 WHR system is targeted at about $4.6MWh, which would be 20~30% lower than ORC for a 5 to 6MW class of system [12]. The main reasons for sCO2 OPEX competitiveness are listed below.

- Elimination of thermal oil intermediate loop increasing the power available and reducing the system cost.
- Reduced field maintenance time by easy access to the power block and simplified system
- Non-toxic and non-flammable CO2 working fluid eliminates the need for some safety systems.

Economics Comparison (Expected IRR)

The economic case study for compressor station GT WHR wholesale BOO business, based on Northern Border Pipeline RB211 gas turbine, IRR (Internal Rate of Return) for sCO2 and ORC is compared below [12]. (ORC 5.5MW vs. sCO2 6.8MW net power, with an assumed $50/MWh)

[1] sCO2: 13.0%

[2] ORC: 11.3% as of 2007

(8.5% as of 2023, after considering inflation, assuming a conservative 1.0% yearly escalation)

With this case study, it is estimated that sCO2 economics would be competitively better than ORC, through higher net power output/efficiency and lower CAPEX & OPEX as shown in Figure 4 below.

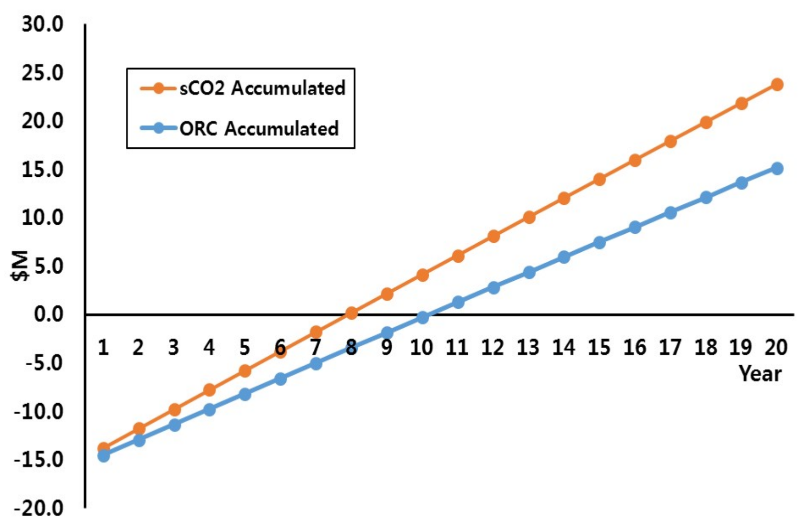

Figure 4. Comparison of Cash Flow for sCO2 and ORC for Economics Case Study (2023 Basis)

Conclusions

The integration of supercritical CO2 (sCO2) bottoming cycles presents a promising economic and technological advancement over the current design of steam and organic Rankine cycle systems for large gas turbines. With the ability to operate utilizing air cooling and eliminating the need for a thermal oil loop, sCO2 systems offer simplified and cost-effective solutions. The utilization of CO2 as the working fluid, operating at pressures above the critical point, ensures that freezing is not a concern.

Comparative analysis with ORC indicates a potential power recovery improvement of at least 20% with sCO2 technology, leading to enhanced return on investment (ROI) and lower capital and operating expenses, especially at favorable sites. Successful testing of key components has paved the way for initial commercialization efforts, underscoring the promising future of sCO2 bottoming cycles for recovering unused thermal energy in gas compressor stations.

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