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Increasing the Efficiency of Compressor Stations of Main Gas Pipelines

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Abstract. *The recovery and utilization of waste heat from gas turbine engines (GTE) currently stand out as one of the most critical and pressing engineering challenges in the energy sector. This issue is particularly vital for the compressor stations of main gas pipelines due to the massive total installed capacity of the driving gas turbine units. Standard waste heat recovery systems, such as waste heat boilers or traditional gas turbine cycles, impose significant aerodynamic backpressure on the exhaust of the primary gas turbine engines. This backpressure reduces the available enthalpy drop across the power turbine, causing a substantial drop in the shaft power output. To compensate for this loss and restore the original compressor performance, operators must increase fuel consumption, which inherently degrades the overall thermodynamic and economic efficiency of the facility. To resolve this technological conflict, inverted-cycle gas turbine units (ICGTU) can be effectively implemented. In these specialized sub-atmospheric configurations, no additional backpressure is generated at the exhaust of the primary engine because the aerodynamic losses are successfully compensated for by the pressure rise inside the specialized utilization compressor. The thermodynamic process in an ICGTU operates uniquely: the hot exhaust gases from the primary engine enter the turbine first, expand below atmospheric pressure, flow through a gas-water cooler, and are subsequently compressed back to ambient atmospheric pressure. This study provides a comprehensive mathematical and thermodynamic analysis to determine the optimal pressure ratio* ($\pi^*_{k.opt}$) *for maximizing specific work and thermal efficiency. The implementation of an ICGTU can contribute an additional 15–25% to the total mechanical shaft power of the station. When integrated with a district or technological water heating system, the overall fuel utilization coefficient reaches an impressive 0.80–0.85, while simultaneously reducing specific hazardous emissions by 15–25% relative to the combined power output.*

Keywords: *Waste Heat Recovery, Combined Gas-Gas Cycle, Inverted-Cycle Gas Turbine, Thermal Efficiency, Gas Pipeline Compressor Stations*

Introduction

The transport of natural gas over long distances through main gas pipeline networks requires immense amounts of energy to overcome hydraulic friction and maintain required pipeline pressures. Gas compressor stations, distributed systematically along these pipelines, rely heavily on heavy-duty gas turbine engines (GTEs) to drive centrifugal gas compressors (Boyce, 2012; Saravanamuttoo et al., 2009). However, a major disadvantage of industrial gas turbines is their relatively low standalone thermal efficiency, which typically ranges from 28% to 36% for older and mid-generation units.

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A massive portion of the fuel energy is lost to the atmosphere in the form of high-temperature exhaust gases (Walsh & Fletcher, 2004). Utilizing this waste heat to generate secondary energy—either for driving auxiliary compressors, supplying internal station utilities, or generating electricity for the external power grid—represents a crucial mechanism for energy conservation and environmental mitigation (American Society of Mechanical Engineers, 2022). The scale of this potential is immense; for instance, the total capacity of gas turbine drives operating within the gas transportation system of the Russian Federation alone exceeds 38,000 MW. Transforming even a fraction of this wasted thermal energy into useful mechanical or electrical work can yield substantial economic benefits. The integration of waste heat recovery (WHR) systems is highly recommended for compressor stations where the existing gas-pumping units have a long remaining operational lifespan and are not scheduled for decommissioning in the near future (ASME, 2022). Furthermore, such retrofitting projects must prove economically viable, meaning that the required capital expenditures (CAPEX) must be low enough to guarantee a short payback period through fuel savings.

Despite the clear benefits, traditional waste heat recovery systems pose serious operational challenges. When conventional heat exchangers, such as waste heat boilers or recuperators, are placed directly into the exhaust duct of a primary gas turbine, they create an inevitable aerodynamic resistance (backpressure) (Horlock, 2002; Wilson & Korakianitis, 2014). This backpressure increases the pressure at the exit of the power turbine, thereby reducing the available expansion ratio and causing a direct drop in the mechanical power delivered to the natural gas compressor. To restore the gas-pumping unit to its nominal capacity, operators are forced to increase the fuel flow rate, which leads to higher thermal stresses and lowers the net efficiency gains.

Historically, the most common method of utilizing exhaust gas heat has been the installation of simple heat exchangers to heat water. This hot water is then distributed for space heating within the compressor station buildings, nearby residential settlements, or adjacent agricultural greenhouses. For electricity generation, steam turbine waste heat recovery units (Combined Cycle Gas Turbines (CCGT)) are frequently employed (Kostyuk & Frolov, 1988). However, because the initial parameters of the steam are strictly bounded by the exhaust temperature of the primary gas turbine, these steam cycles usually operate at modest pressures and temperatures.

Furthermore, water-steam cycles require complex, bulky, and expensive auxiliary equipment, including recovery boilers, steam turbines with dedicated speed governors, water condensers, deaerators, feed pumps, and extensive chemical water treatment facilities. This complexity increases operational and maintenance costs (OPEX), making steam cycles less attractive for remote or harsh geographic locations. As an alternative, Organic Rankine Cycles (ORC) utilizing low-boiling-point working fluids, such as pentane, butane, or toluene, have been introduced (Najjar, 2013). While ORCs offer better thermodynamic matching at lower exhaust temperatures, they still face issues regarding fluid flammability, toxicity, and high equipment costs.

To bypass the limitations of both steam and organic fluid cycles, gas-turbine waste heat recovery loops have gained significant research attention. These loops can be designed to operate either via traditional closed/open Brayton cycles or via specialized inverted-cycle configurations (Polyzakis, 2019; Zhang & Lior, 2006). In a traditional gas turbine waste heat recovery loop, the heat from the primary GTE exhaust is transferred through a heat exchanger wall to a clean working fluid (usually air or nitrogen) circulating in a closed loop. Due to the inevitable heat transfer temperature difference across the exchanger, the maximum temperature of the working fluid in the recovery loop is restricted, typically remaining below 450°C. While an auxiliary combustion chamber can be integrated to burn additional fuel and boost the recovery loop's power output, this approach shares the same drawback as steam boilers: it increases the station's total fuel consumption to maintain or restore nominal operating parameters.

Methods

To completely eliminate the negative impacts of backpressure on the primary engine, the concept of an Inverted-Cycle Gas Turbine Unit (ICGTU) was developed (Najjar, 2013; Zhang & Lior, 2006). The fundamental difference between a traditional gas turbine cycle and an inverted cycle lies in the sequence of thermodynamic processes relative to atmospheric pressure (Boyce, 2012; Horlock, 2002).

In an ICGTU, the hot exhaust gases from the primary gas turbine engine are not passed through a heat exchanger first; instead, they are routed directly into the inlet of the utilization turbine (Polyzakis, 2019). The pressure of the gases entering this turbine is very close to ambient atmospheric pressure, while their temperature matches the exhaust temperature of the primary power turbine. Inside the utilization turbine, the gases expand to a sub-atmospheric pressure level ($p < p_a$). This sub-atmospheric expansion allows the turbine to extract mechanical energy directly from the hot exhaust gas stream without placing any aerodynamic restriction on the primary engine. After expanding through the turbine, the sub-atmospheric, partially cooled gases enter a specialized gas-water heat exchanger (gas cooler). In this cooler, heat is rejected from the gas stream to an external cooling medium (such as water or ambient air), causing the gas temperature to drop significantly while its pressure remains sub-atmospheric. The cold, low-pressure gas is then drawn into the inlet of the utilization compressor. The compressor compresses the gases from the sub-atmospheric pressure back up to standard atmospheric pressure (p_a), after which they are discharged safely through the exhaust stack into the environment.

A schematic diagram and the corresponding thermodynamic T–s (Temperature–Entropy) diagram of this combined system are illustrated in Figure 1.

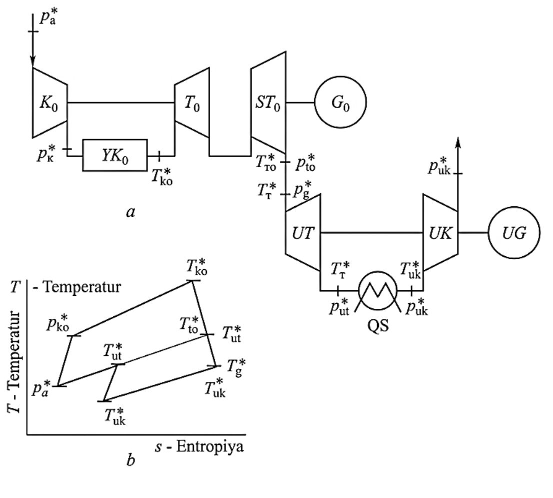

Figure 1

Operational flow of the Inverted-Cycle Gas Turbine Unit (ICGTU) matching the thermodynamic cycle depicted in the original source

In this sub-atmospheric arrangement, the maximum pressure within the utilization loop never exceeds ambient atmospheric pressure. Unlike conventional gas turbine configurations where the compressor forces the working fluid into the system under high pressure, the ICGTU operates via an induced-draft mechanism. The compressor essentially acts as a vacuum pump, pulling the working fluid through the turbine and cooler. Because the static and total pressure at the exit of the primary engine's power turbine remains entirely unperturbed, the primary engine maintains its original design performance and power output. Any minor pressure drops caused by the internal friction of the connecting ducts or the gas cooler are completely overcome and compensated for by the work input of the utilization compressor. To evaluate the mathematical optimization of this cycle, we must analyze the specific useful work (L_e) generated by the utilization loop. In standard Brayton cycles, the specific work reaches a maximum at a specific pressure ratio denoted as π_{kL}^*, whereas the thermal efficiency peaks at a higher pressure ratio $\pi_{k\eta}^*$. However, for an inverted cycle operating on a fixed stream of exhaust gas, the mass flow rate and the available inlet thermal energy are predetermined by the primary engine's operating state. Therefore, the specific work and the total efficiency of the utilization loop peak at the exact same optimal pressure ratio ($\pi_{k.opt}^*$).

The specific useful work of the ICGTU can be formulated using basic aerothermodynamics equations:

$$L_e = C_{pt} \cdot T_t^* \cdot \left(1 - \frac{1}{\pi_T^{*\frac{k_t-1}{k_t}}}\right) \cdot \eta_t \cdot \eta_m - C_{pk} \cdot T_k^* \cdot \left(\pi_k^{*\frac{k_k-1}{k_k}} - 1\right) \cdot \frac{1}{\eta_k}$$

Where:

- C_{pt}, C_{pk} are the mean specific isobaric heat capacities of the gas during expansion (turbine) and compression (compressor), respectively.
- k_t, k_k are the isentropic exponents for the turbine and compressor processes.
- T_t^* is the total stagnation temperature of the gas at the turbine inlet.
- T_k^* is the total stagnation temperature of the gas at the compressor inlet (after the gas cooler).
- π_T^*, π_k^* are the total pressure expansion ratio of the turbine and compression ratio of the compressor.
- η_t, η_k are the isentropic efficiencies of the turbine and compressor.
- η_m is the mechanical efficiency of the shaft assembly.

Since the working fluid throughout the entire ICGTU consists of the same flue gas mixture (operating under high excess air coefficients of $\alpha_\Sigma = 3.0 - 3.5$), and considering that the mean temperature difference between the expansion and compression phases is roughly 300°C to 350°C, we can apply a first-order simplification assuming equal average thermodynamic properties:

$$C_{pt} = C_{pk} = C_p \quad \text{and} \quad k_t = k_k = k$$

The turbine expansion ratio π_T^* is mathematically linked to the compressor pressure ratio π_k^* through the total pressure preservation coefficients of the system:

$$\pi_T^* = \pi_k^* \cdot \sigma_{gir} \cdot \sigma_x$$

Where σ_{gir} is the pressure recovery factor of the inlet transition duct and σ_x is the pressure recovery factor of the gas cooler. Substituting these variables into the specific work equation yields:

$$L_e = C_p \cdot T_t^* \cdot \left[1 - \frac{1}{\left(\pi_k^* \cdot \sigma_{gir} \cdot \sigma_x\right)^{\frac{k-1}{k}}}\right] \cdot \eta_t \cdot \eta_m - C_p \cdot T_k^* \cdot \left(\pi_k^{*\frac{k-1}{k}} - 1\right) \cdot \frac{1}{\eta_k}$$

To find the optimal value of π_k^* we can define a substitution variable $X = \pi_k^{*\frac{k-1}{k}}$. This transforms the equation into a simplified algebraic form:

$$L_e = A - \frac{B}{X} - C \cdot X$$

Where the constant coefficients A, B, and C are defined as follows:

$$A = C_p \cdot T_t^* \cdot \eta_t \cdot \eta_m + \frac{C_p \cdot T_k^*}{\eta_k}$$

$$B = \frac{C_p \cdot T_t^* \cdot \eta_t \cdot \eta_m}{\left(\sigma_{gir} \cdot \sigma_x\right)^{\frac{k-1}{k}}}$$

$$C = \frac{C_p \cdot T_k^*}{\eta_k}$$

Results

To calculate the exact optimal pressure ratio that yields the maximum specific work, we take the first derivative of $L_e$with respect to the variable X and set it to zero:

$$\frac{dL_e}{dX} = \frac{B}{X^2} - C = 0 \Rightarrow X_{opt} = \sqrt{\frac{B}{C}}$$

Reversing the substitution variable $X = \pi_{k.opt}^{*\frac{k-1}{k}}$ provides the final analytical solution for the optimal compressor pressure ratio:

$$\pi_{k.opt}^* = \left(\frac{T_t^*}{T_k^*}\right)^{\frac{k}{2(k-1)}} \cdot \left(\frac{\eta_t \cdot \eta_k \cdot \eta_m}{\left(\sigma_{gir} \cdot \sigma_x\right)^{\frac{k-1}{k}}}\right)^{\frac{k}{2(k-1)}}$$

This derivation shows that the optimal pressure ratio of an ICGTU is primarily dictated by the temperature ratio across the cycle ($\frac{T_t^*}{T_k^*}$), alongside the component efficiencies and aerodynamic pressure losses. For standard industrial gas turbines operating at compressor stations (such as the NK-14ST, GTNR-16, or GTU-25P), the exhaust gas temperatures typically range from 724 K to 790 K. Assuming effective cooling in the gas-water heat exchanger reduces the compressor inlet temperature T_k^* to approximately 310–330 K, the optimal sub-atmospheric pressure $\pi_{k.opt}^*$ typically falls within a narrow range between 1.5 and 2.2.

When operating at this optimized thermodynamic point, the implementation of the inverted cycle yields highly favorable performance metrics:

1. **Power Augmentation:** The secondary sub-atmospheric loop generates substantial shaft power without modifying the primary engine core. This adds an additional 15% to 25% of clean mechanical or electrical power to the station's baseline capacity.
2. **Fuel Utilization Factor:** By fully capturing the waste heat for power generation and utilizing the remaining low-grade thermal energy from the gas cooler to heat water, the overall fuel utilization coefficient (η_{fu}) of the combined plant rises to 0.80–0.85.

3. **Environmental Impact:** Because the total mechanical power output increases significantly while the absolute fuel consumption remains unchanged, the specific emissions of hazardous pollutants (such as NO_x and CO) drop by 15% to 25% relative to the total combined plant output.

Discussion

The analytical models and results presented here confirm that inverted-cycle gas turbine units offer a highly effective alternative to traditional steam-water combined cycles for pipeline compressor stations. The primary advantage is the complete absence of aerodynamic backpressure on the primary engine, allowing it to maintain peak nominal performance under all operating conditions. This completely bypasses the traditional efficiency penalty associated with regular heat recovery exchangers (Najjar, 2013; Polyzakis, 2019; Zhang & Lior, 2006).

However, successfully deploying an ICGTU requires addressing several unique engineering constraints:

- Volumetric Flow Rates: Because the turbine expands the working fluid into sub-atmospheric pressures ($p < 1$ bar), the specific volume of the gas increases drastically. This requires larger turbine blade heights and larger flow cross-sections, which increases the physical size and structural mass of the low-pressure turbomachinery components (Boyce, 2012; Wilson & Korakianitis, 2014).
- Sealing and Structural Rigidity: Operating under vacuum conditions requires robust casing designs to prevent atmospheric air leakage into the sub-atmospheric zones, which would dilute the working fluid and degrade compressor efficiency (Polyzakis, 2019).
- Optimization Limitations: Due to structural, material, weight, and dimensional constraints at remote compressor stations, the actual design pressure ratio is sometimes chosen to be slightly below the theoretical thermodynamic optimum (π_(k.opt)^*). This slight reduction helps minimize the overall footprint and manufacturing costs of the equipment while preserving the majority of the efficiency gains (Horlock, 2002; Wilson & Korakianitis, 2014).

Future research will focus on developing advanced compact heat exchangers to minimize the pressure drop (σ_x) and exploring multi-stage sub-atmospheric compression with intercooling to further boost the net efficiency of the combined gas-gas cycle.

Conclusion

This study demonstrates that the implementation of an Inverted-Cycle Gas Turbine Unit (ICGTU) provides an effective solution for recovering waste heat from gas turbine-driven compressor stations without introducing the aerodynamic backpressure associated with conventional waste heat recovery systems. Unlike traditional steam or recuperative technologies, the proposed sub-atmospheric gas-gas cycle preserves the operating conditions and rated power of the primary gas turbine while converting a significant portion of the exhaust gas energy into additional useful mechanical work.

A mathematical model was developed to determine the optimal compressor pressure ratio for the inverted cycle. The derived analytical expression shows that the optimum pressure ratio is primarily governed by the turbine-to-compressor inlet temperature ratio together with the efficiencies of the turbine, compressor, and mechanical transmission, as well as the pressure recovery characteristics of the utilization system. Under operating conditions typical of industrial gas pipeline compressor stations, the optimal pressure ratio generally lies between 1.5 and 2.2.

The thermodynamic analysis indicates that the proposed system can increase the total shaft power output of compressor stations by approximately 15–25% without increasing the fuel consumption of the primary gas turbine. Furthermore, by combining mechanical power recovery with low-grade heat utilization for water heating, the overall fuel utilization coefficient can reach 0.80–0.85. Because the total useful energy output increases while fuel consumption remains essentially unchanged, the specific emissions of harmful pollutants per unit of useful energy are reduced by approximately 15–25%.

Although practical implementation requires careful consideration of larger volumetric flow rates, vacuum sealing, and the structural design of low-pressure turbomachinery, these engineering challenges do not outweigh the significant thermodynamic and economic advantages of the proposed concept. Consequently, the inverted-cycle gas turbine represents a promising technology for improving the energy efficiency, operational reliability, and environmental performance of existing main gas pipeline compressor stations. Future research should focus on experimental validation of the proposed mathematical model, optimization of compact low-pressure heat exchangers, and investigation of multi-stage sub-atmospheric compression systems with intercooling to further enhance cycle performance.

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