REPOWERING DIESEL ENGINE POWERED THERMAL POWER PLANTS THROUGH A POWER AND COOLING COGENERATION CYCLE

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ABSTRACT

This study explored the applicability of the Goswami cycle for repowering internal combustion engine-based power plants, using the Viana Power Plant as a case study. Thermodynamic, exergetic, and economic analyses were performed. The cycle optimized for power generation (GWC-P) outperformed the refrigeration-focused configuration (GWC-R), enhancing net electrical generation and exergetic efficiency. Despite the complexity and implementation costs of the Goswami cycle, GWC-P exhibited potential, yielding up to 5,069 kW of additional power without extra fuel consumption. While GWC-R offered cooling benefits, GWC-P overshadowed it in terms of power generation enhancement. The research highlights the feasibility of the Goswami cycle for repowering, improving efficiency and power capacity in existing power plants.

Keywords: Diesel Engine Thermal Power Plants, Goswami Cycle, Waste Heat Recovery.

Introduction

The growing demand for electrical energy has led the Brazilian thermoelectric power plants, mostly designed and built with low cost and reduced efficiency, to operate more frequently. These plants discard a significant portion of residual heat, which could be harnessed to repower and enhance the efficiency of these facilities without requiring additional fuel consumption. In the context of thermoelectric power plants using internal combustion engines, repowering is generally carried out in two ways: reusing the residual heat for (1) additional power generation in bottoming cycles or (2) cooling the intake air of the engines through absorption chillers, thereby increasing the power generated by the engine itself and reducing specific fuel consumption.

The Goswami cycle presents itself as an important alternative, as it is a cogeneration cycle capable of addressing both mentioned forms of repowering, resulting in an efficiency increase of the plant. In this regard, the contribution of this work lies in the evaluation of the Goswami cycle as a viable alternative for repowering internal combustion engine-based thermoelectric power plants through the recovery of residual heat from these engines. The present study employs the Viana Power Plant as a case study, aiming to present an additional option for repowering the thermoelectric power plants that constitute the Brazilian energy matrix.

Study Case

Located in the state of Espirito Santo, Brazil, the Viana Thermal Power Plant (Viana TPP) is a Brazilian facility equipped with twenty Wärtsilä W20V32 diesel engines capable of generating 9,000 kW of mechanical power each, coupled to generators configured to produce a total of 8,730 kW of electrical power, resulting in a combined installed capacity of 174.6 MW [1]. The plant employs heavy oil OCB-1 as its fuel source, which possesses a lower heating value (LHV) of 40,785 kJ kg^{-1}, a mass flow rate of 0.5 kg s^{-1}, and a molar-based chemical composition of 40.35% carbon (C), 59.22% hydrogen (H), 0.26% oxygen (O), and 0.17% sulfur (S), as specified by [2].

The primary source of available residual heat within the plant comes from the engine exhaust gases. Exhaust gases are only accessible from fifteen engines, as the exhaust gases from the other five are utilized in recovery boilers to produce steam for internal processes. The exhaust gases have a mass

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flow rate of 16.7 kg s^{-1}, a temperature of 345°C, and a molar-based chemical composition comprising 6.36% CO_2, 5.58% H_2O, 75.53% N_2, 11.60% O_2, 0.90% Ar, and 0.03% SO_2, as determined through a complete combustion model by [2].

System Description

The main components of the Goswami cycle (GWC) are: absorber, pump, heat exchanger, recovery boiler, separator/rectifier, turbine, expansion valve, and refrigeration heat exchanger, as illustrated in Fig. 1.

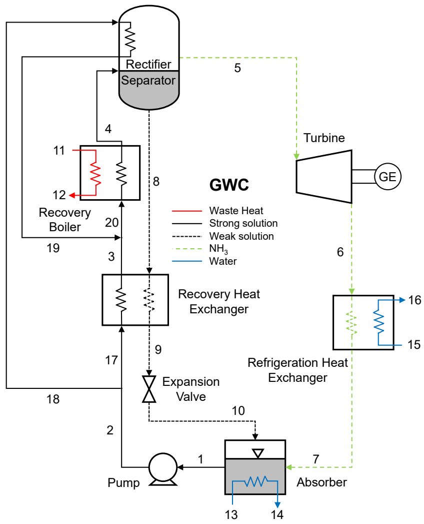

Figure 1: Schematic diagram of the Goswami cycle.

The concentration of ammonia varies throughout the cycle, such that in state 1, the flow of strong solution (ammonia-rich mixture) exits the absorber as saturated liquid at low pressure, is pumped to the high pressure of the cycle (state 2), and preheated through a heat exchanger. Upon entering the recovery boiler, the mixture is partially evaporated, generating a biphasic mixture (state 4): a relatively ammonia-poor liquid and an ammonia-rich vapor. In the rectifier, a cold stream cools the saturated ammonia-rich vapor to condense any remaining water, allowing the concentration of ammonia-rich vapor to increase, generating nearly pure ammonia vapor (state 5). Subsequently, the ammonia vapor expands in the turbine, generating power while throttling the fluid to the system's low pressure (state 6). Later, under certain operational conditions, the temperature of the fluid exiting the turbine in state 6 can be significantly lower than ambient temperature, such that cooling can be achieved by sensible heating of the expander's exhaust flow through a cooling heat exchanger (state 7). The residual liquid in the separator (state 8), also known as weak solution (lower ammonia concentration), is used to preheat the strong solution in the heat exchanger. The heat exchanger preheats the strong solution and also ensures the cooling of the weak solution to saturation conditions to satisfy zero vapor quality requirements. After leaving the heat exchanger, the flow of weak solution (state 9) is throttled to the system's low pressure and sprayed into the absorber (state 10). Finally, the ammonia vapor rejoins the weak solution in the absorber, where it is absorbed, releasing heat to the ambient.

Metodology

In this section, the thermodynamic model developed for thermodynamic and economic analyses is presented.

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Thermodynamic Analysis Methodology

The principles of conservation of mass, the first and second law of thermodynamics are applied to each component of the system, and the developed equations will be described below.

The balances of mass flow and solution concentration for each cycle component are applied, starting from Eqs. 1 and 2.

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

$$\sum \dot{m}_{in} \cdot x_{in} = \sum \dot{m}_{out} \cdot x_{out} \tag{2}$$

where: $\dot{m}$ is the mass flow rate and x is the concentration of ammonia in the solution in kg of NH_3 / kg of solution. The developed mass and energy balances are shown in Tab. 1.

Table 1: Mass and energy balances for the main components of the Goswami cycle.

Component	**Mass Balance**	**Energy Balance**
Pump	$\dot{m}_1 = \dot{m}_2$	$\dot{W}_{pump} = \dot{m}_1(h_2 - h_1)$
Recovery Boiler	$\dot{m}_{20} = \dot{m}_4$	$\dot{Q}_{boiler} = \dot{m}_{20}(h_4 - h_{20})$
Turbine	$\dot{m}_5 = \dot{m}_6$	$\dot{W}_{turbine} = \dot{m}_5(h_5 - h_6)$
Refrigeration Heat Exchanger	$\dot{m}_6 = \dot{m}_7$	$\dot{Q}_{cool} = \dot{m}_7(h_6 - h_7)$
Absorber	$\dot{m}_1 = \dot{m}_7 + \dot{m}_{10}$	$\dot{Q}_{absorber} = \dot{m}_7 h_7 + \dot{m}_{10} h_{10} - \dot{m}_1 h_1$
Net Power	$-$	$\dot{W}_{net} = \dot{W}_{elet,T} - \dot{W}_{elet,P}$

Economic Analysis Methodology

The economic analysis employed in this study is based on the modular cost technique. To conduct the analysis, it is necessary to calculate the heat exchanger areas that constitute the system. For this purpose, the method of logarithmic mean temperature difference (ΔT_{LMTD}) is applied, as described in Eq. 3.

$$\Delta T_{LMTD} = \frac{\Delta T_{hot} - \Delta T_{cold}}{\ln \dfrac{\Delta T_{hot}}{\Delta T_{cold}}} \tag{3}$$

Where ΔT_{hot} is the temperature difference between the hot side streams and ΔT_{cold} is the temperature difference between the cold side streams of the heat exchanger. With this parameter, it's possible to evaluate the heat exchanger area as described in Eq. 4.

$$A_{HE} = \frac{\dot{Q}}{U \Delta T_{LMTD}} \tag{4}$$

Where: $\dot{Q}$ represents the values of thermal energy exchanged between the fluids, obtained through the conducted thermodynamic analysis, and U is the overall heat transfer coefficient, sourced from the literature.

The subsequent steps involve calculating the basic cost and modular cost of the equipment, with the latter contributing to the total cost. The basic cost for the heat exchangers is obtained using Eq. 5, while for the pump and turbine, it is calculated using the same equation, replacing the term A_{HE} for the heat transfer area with the term $\dot{W}$ for power.

$$\log_{10}(C_B^0) = K_1 + K_2 \cdot \log_{10}(A_{HE}) + K_3 \cdot (\log_{10}(A_{HE}))^2 \tag{5}$$

The modular cost of the equipment can be calculated using Eq. 6. For certain equipment, such as the turbine, F_{mod} is obtained from tables, while for others, this factor is computed using coefficients B_1 and B_2, along with factors F_M and F_P, where F_M is the material factor and F_P is the pressure factor.

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$$C_{mod} = C_B^0 \cdot F_{mod} = C_B^0 \cdot (B_1 + B_2 \cdot F_M \cdot F_P) \quad [US\$] \tag{6}$$

To use Eq. 6, it's necessary to calculate the pressure factor F_P as described in Eq. 7.

$$\log_{10}(F_P) = C_1 + C_2 \cdot \log_{10}(P) + C_3 \cdot (\log_{10}(P))^2 \tag{7}$$

Finally, the acquisition costs of the components are updated to account for inflation to the year 2022 using Eq. 8, employing the CEPCI (Chemical Engineering Plant Cost Index).

$$C_{total,2022} = \frac{CEPCI_{2022}}{CEPCI_{ref}} \cdot \sum C_{i,ref} \quad [US\$] \tag{8}$$

With $CEPCI_{2022}$ value of 832.6 [3] and $CEPCI_{ref}$ value of 397 for the year 2001 [4]. The coefficients used to feed the equations described in this section were taken from [4]. The costs associated with auxiliary equipment were considered to be 10% of the modular cost of the system's components.

Model validation and optimization

The simulations for thermodynamic and economic analyses were conducted using the software EES (Engineering Equation Solver) from the company F-Chart under an academic license. The thermodynamic properties of the binary water-ammonia mixture were obtained using the software's internal library. The model was previously validated based on the work of [5], demonstrating good agreement.

The Goswami cycle was optimized using Genetic Algorithms in EES. As a cogeneration cycle, it offers the versatility to be optimized for power generation and refrigeration, resulting in two distinct objective functions. We refer to the cycle optimized to maximize net power generation as GWC-P and the cycle optimized to maximize cooling effect as GWC-R.

Results and discussion

The Goswami cycle optimized for power generation (GWC-P) produced a net electrical power of 337.95 kW, marking a 3.87% increase in engine power. It also yielded a cooling effect of 50.46 kW. In comparison, the cycle optimized for refrigeration (GWC-R) generated 269.4 kW, with a 3.09% increase in engine power and a cooling effect of 186.77 kW.

In terms of efficiency, GWC-P demonstrated thermal and exergetic efficiencies of 11.81% and 26.76%, respectively. GWC-R showed lower efficiencies at 9.41% and 21.61%, respectively. GWC-R exhibited a higher energy utilization factor (15.94%) compared to GWC-P (13.57%), indicating superior energy usage for the generated products. These details can be found in the Tab. A.1 provided in the supplementary material.

According to [2], to cool the engine's intake air by 15°C from an ISO condition of 25°C, 1 atm, and 30% relative humidity, a cooling capacity of 45.4 tons of refrigeration (TR) is required, resulting in an additional power generated by the engine of 48.3 kW. The proposed cooling system, an absorption chiller, requires electricity to power the cooling tower and auxiliary equipment, resulting in an additional net electrical power of 29.2 kW per engine.

Therefore, although the GWC-R cycle can meet the entire required thermal load, when adding the additional power generated by the engine (48.3 kW) to the power generated by the cycle (269.36 kW), the combined additional power (317.66 kW) is lower than that produced by the GWC-P cycle. As a result, using this configuration to aim for additional power generation is not justifiable.

On the other hand, even though the GWC-P cycle doesn't meet the entire necessary thermal load for cooling the intake air, it generates approximately 7 times more additional power than the cooling effect obtained by the absorption cooling system. Additionally, the GWC-P cycle produces chilled water with a substantial cooling capacity, which can be used internally in the power plant for air conditioning or sprayed onto radiators to enhance heat exchange and reduce the power consumption of auxiliary fans.

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From an economic perspective, the GWC-P configuration has a total implementation cost of US$ 2,512,532, while the GWC-R configuration costs US$ 2,532,209. The higher cost of GWC-R is due to the use of higher evaporation pressures in the cycle, aiming to achieve lower temperatures at the turbine exit and a larger refrigeration effect. Hence, the configuration optimized for power generation proves to be the more viable option. Refer to Tab. A.2 in the supplementary material for details on the economic analysis.

Conclusions

This study assessed the use of the Goswami cycle as a repowering alternative for internal combustion engine-based power plants, employing the Viana Power Plant as a case study. Thermodynamic evaluation of cycle configurations was conducted through energy and exergy calculations, while economic analysis relied on literature data to estimate the additional power generation potential and associated costs for each configuration.

The Goswami cycle configuration optimized for refrigeration was not justified, as the configuration optimized for power generation yielded a higher net electrical power than the GWC-R cycle, in addition to cooling the intake air of the engines. Therefore, from the perspective of effectively increasing the power output of the generating unit, the GWC-P configuration proved to be the most feasible among the evaluated Goswami cycle configurations. It could generate up to 5,069 kW of additional power, representing a 2.9% increase in installed capacity without additional fuel consumption. Furthermore, this cycle would enable a global exergetic efficiency of 41.81%, indicating a 1.51% increase compared to the current efficiencies of the power plant. The Goswami cycle is complex and requires various components, leading to a substantial implementation cost. However, it possesses a cooling effect that can result in savings for other equipment in the power plant.

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APPENDIX A

Table A.1: Parameters obtained through thermodynamic analysis of the Goswami cycle.

Parâmetro	GWC-P	GWC-R
Heat source (kW)	2,862.36	2,862.36
Cooling (kW)	50.46	186.77
Net power (kW)	337.95	269.36
Thermal efficiency (%)	11.81	9.41
Energy utilization factor (%)	13.57	15.94
Exergetic efficiency (%)	26.76	21.61

Table A.2: Main parameters obtained for economic analysis.

Parameters	GWC-P	GWC-R
Absorber Cost [US$]	275,245	344,250
Recovery Boiler Cost [US$]	387,785	373,775
Refrigeration Heat Exchanger Cost [US$]	22,695	28,721
Separator/Rectifier Cost [US$]	135,322	118,858
Recovery Heat Exchanger Cost [US$]	198,432	214,582
Pump Cost [US$]	43,344	83,079
Turbine Cost [US$]	1,221,297	1,138,743
Auxiliary Equipment Cost [US$]	228,412	230,201
Total Cost [US$]	2,512,532	2,532,209

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