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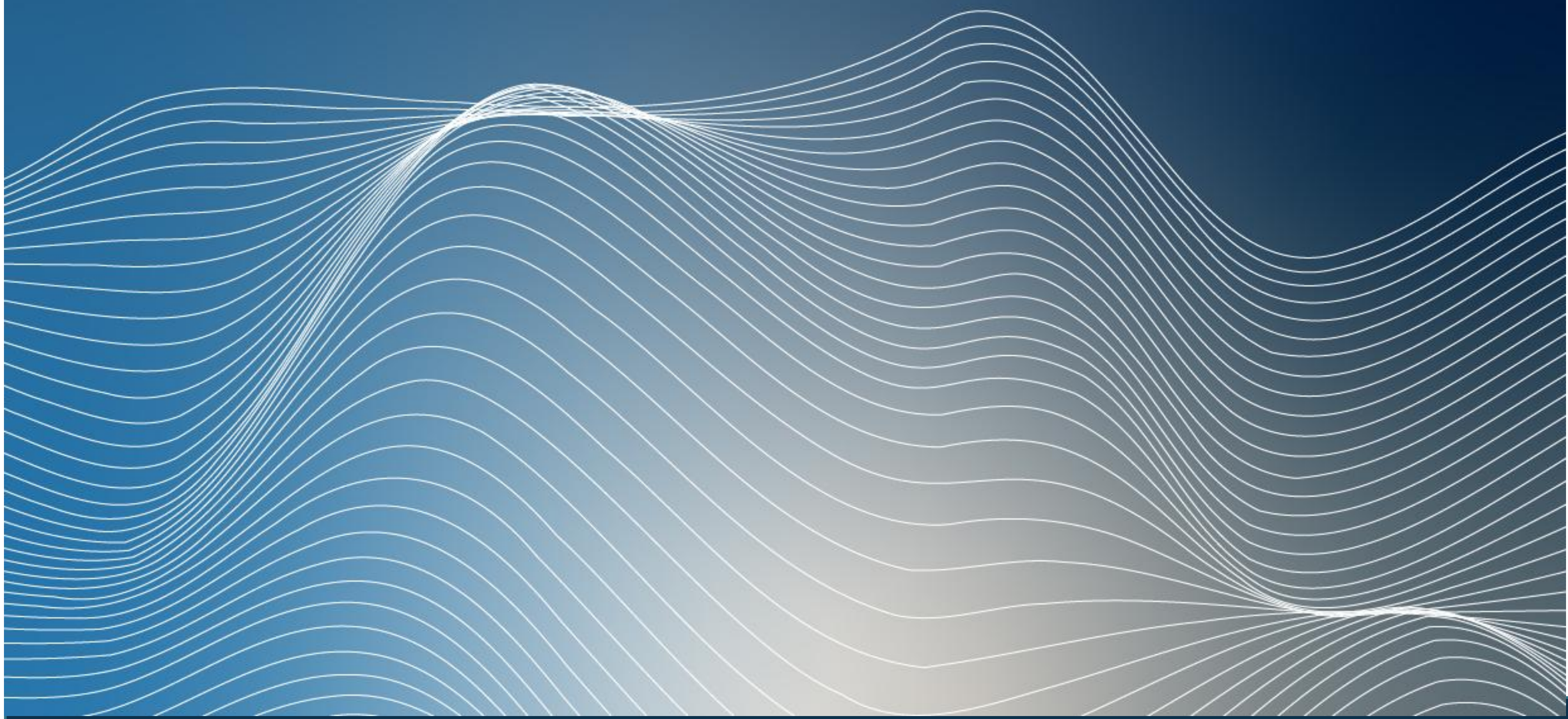
Thermodynamics system design



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List of abbreviations

ABBREVIATION	DESCRIPTION
IC	Intercooler
LPC	Low pressure compressor
HPC	High pressure compressor
HPT	High pressure turbine
LPT	Low pressure turbine
RECU	Recuperator
WHB	Waste heat recovery boiler
LPG	Low pressure shaft generator
HPG	High pressure shaft generator

1. Introduction

The recuperated gas turbine (GT) process with waste heat recovery (WHR) using hydrogen as fuel is simulated to investigate various cycle configurations and to evaluate the feasibility of the proposed cycle. The GT comprises of three compressors, two intercoolers (IC) and two turbines and they are mounted on two different shafts. In addition, the system includes a combustion chamber, a recuperator and a waste heat recovery boiler. Simulations explore different pressure ratios and cooling flow arrangements. Simulations incorporate updated turbomachinery efficiencies and pressure ratios from mean line simulations and account for pressure losses in different components. The WHR system, including an exhaust gas boiler and a steam turbine, is first simulated using a simple model based on the assumption of 50% Carnot efficiency. The model it is later updated using more sophisticated methods. The goal of the study is to achieve a high electric efficiency while maintaining a feasible and cost-effective design. Ensuring affordability of design requires that equal electric power be produced on both shafts.

The analysis shows that a maximum pressure of 12 bar is the most feasible for the current gas turbine design and a high efficiency is obtained with one IC. However, using a single IC results a slight decrease in GT's efficiency, but greater energy recovery is possible through the WHR system. Additionally, cooling the turbine blades reduces the overall cycle efficiency, so minimizing the cooling is crucial for maximizing system efficiency.

2. Thermodynamic cycle

A thermodynamic evaluation of different cycle architectures was carried out to achieve both good efficiency and a mechanically feasible design. Twelve distinct cases were simulated to identify the most optimal and efficient cycle layout for the current project. Based on the initial design, the two spool recuperated GT with a waste heat recovery cycle was proposed. Both spools are designed to produce equal power output and minimize costs. The maximum turbine inlet temperature was set at 1250 ^{O}C which was found to be feasible for cooled blades. The initial WHR cycle power output is estimated using 50% of Carnot efficiency. This is a simplified estimation and a more detailed analysis for the WHR system is performed later. However, this initial estimation provides an accurate enough basis for determining the most promising cycle option. The different cycle options are shown in Table 1 and cycle diagram with cooling is show in Figure 1.

Table 1. Different cycle options.

Cycle option	Max pressure	cooling	other param
Case 1	8 bar	no cooling	
Case 2	10 bar	no cooling	
Case 3	12 bar	no cooling	
Case 4	14 bar	no cooling	

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Cycle option	Max pressure	cooling	other param
Case 5	12 bar	cooling flow taken after first IC and heated flow to recuperator	
Case 6	12 bar	cooling flow taken after first IC and heated flow to WHR boiler	
Case 7	12 bar	cooling flow taken after second IC and heated flow to recuperator	
Case 8	12 bar	cooling flow taken after first IC and heated flow to recuperator	updated turbomachinery pressure ratio and efficiency
Case 9	12 bar	cooling flow taken after first IC and heated flow to recuperator	No second intercooler, updated turbomachinery pressure ratio and efficiency
Case 10	12 bar	cooling flow taken after first IC and heated flow to recuperator, updated cooling flow	No second intercooler, updated turbomachinery pressure ratio and efficiency
Case 11	12 bar	cooling flow taken after first IC and heated flow to recuperator, updated cooling flow	No second intercooler, updated turbomachinery pressure ratio and efficiency, TIT 1350
Case 12	12 bar	cooling flow taken after first IC and heated flow to recuperator,	No second intercooler, updated turbomachinery pressure ratio and efficiency, TIT 1350

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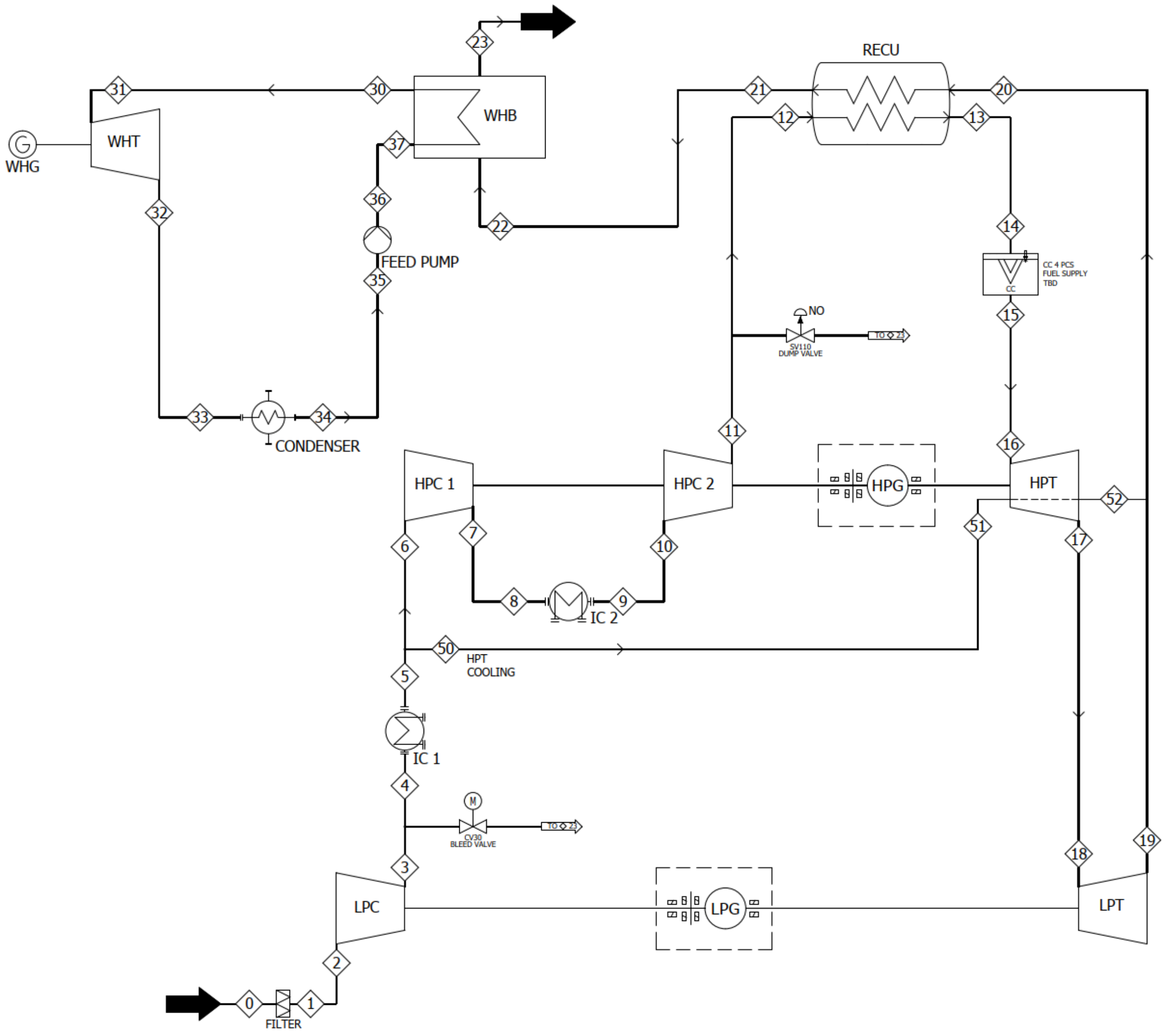

Figure 1. Cycle diagram

2.1. Evaluation of maximum pressure

The cycle configuration is shown in Figure 1. It is a two-spool design with two intercoolers and cooling of turbine blades is adopted. This configuration, with no cooling flow, was used to investigate the feasible maximum pressure. Four different pressure levels (8, 10, 12 and 14 bar) were examined. The net electric efficiencies of the GT and GT + WHR cycle are shown in Figure 2. Constant generator and converter efficiencies 97% were used. It was observed that higher maximum pressure results in increased cycle efficiencies. However, the improvement of efficiency diminishes as pressure increases. The 12 bar case was chosen to be the most suitable option. The simulation was done using constant efficiencies of turbomachinery, but to achieve higher pressure, the rotational speed of the turbomachinery needs to be increased. For rotor and bearing dynamic reasons, the maximum rotational speed was limited to 15 000 rpm and therefore, the optimal design values for turbomachinery could not be chosen. The higher the pressure of the cycle, the more difficult it is to obtain high efficiencies for compressors.

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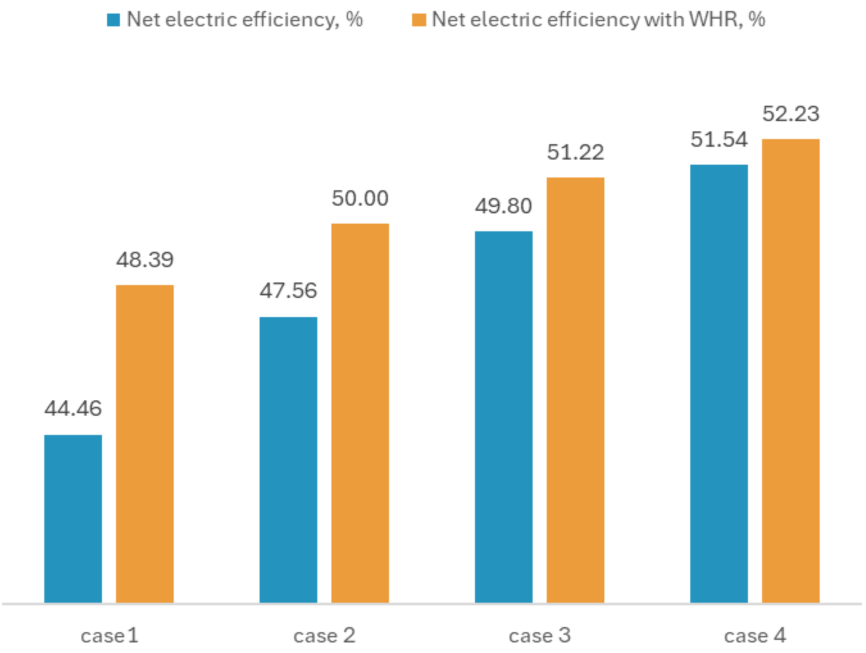

Figure 2. Electric efficiency of GT and GT with WHR with different maximum pressure.

2.2. Evaluation of cooling layout

To achieve high efficiency for the cycle, a high turbine inlet temperature (TIT) was targeted. High TIT cannot be achieved without cooling the blades in the high-pressure turbine. Three different cooling layouts were studied. In all of these cases, cooling is arranged using convection by cooling channels inside the blades.

In the first case (case 5), the cooling flow after the first IC is used for cooling and the heated flow is introduced to the recuperator inlet on the hot side. In the second case (case 6), the cooling flow after the first IC is used for cooling and the heated flow is introduced to the WHR boiler. In the third case (case 7), the cooling flow after the second IC is used for cooling, and the heated flow is introduced to the recuperator inlet on the hot side.

Cooling case 5 is most efficient when considering GT and GT+WHR electric efficiencies. Case 6 will have higher heat power available for WHR, but temperature is lower; therefore, less power is generated by the WHR cycle. The cooling layout presented in case 5 is chosen for the next simulations.

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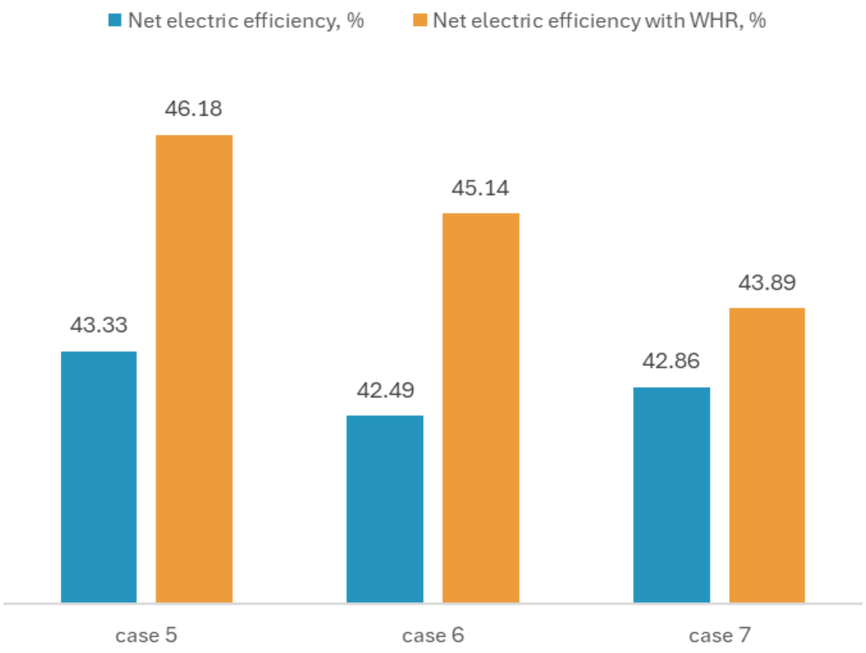

Figure 3 Efficiencies with different cooling layout.

2.3. Updated turbomachinery parameters

The detailed turbomachinery design was carried out based on the cycle simulations presented previously (case 5). Polimi conducted the design and provided updated efficiencies and pressure ratios of turbomachinery (Task 1.4). In addition, the maximum temperature of blades was increased to 1000ºC, which reduced the cooling mass flow from 1.8 kg/s to 0.85 kg/s. The updated values are presented in Table 2. The turbine pressure ratio was adjusted to meet the requirement for the same power of both shafts.

Table 2. Updated turbomachinery design parameters.

	LP compressor	**HP1 compressor**	**HP2 compressor**	**LP turbine**	**HP turbine**
pressure ratio	3.4	2.0	1.76	from process simulation	from process simulation
isentropic efficiency	0.86	0.86	0.85	0.87	0.91

The cycle simulation was updated using the new turbomachinery parameters. The net electric efficiency of GT improved by 3.0% and GT + WHR by 1.7%. These improvements are partly due to the lower cooling flow and the higher allowable blade temperature.

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2.4. Impact of second intercooler

The effect of removing the second intercooler from the cycle was also studied. It is well known that intercooling improves GT efficiency. However, for rotor dynamic reasons, removing the second intercooler was considered, as it would allow for a reduction in shaft length. Without the second intercooler, there is no need for a volute on the HP1 compressor; only a return channel is required, which decreases the necessary axial length. Removing the second intercooler reduced the GT net electric efficiency by 1.1% but improved GT + WHR net electric efficiency by 1.3%. This is because more thermal power is available for the WHR system, and the temperature is higher. Therefore, the process without the second IC is chosen for further studies.

2.5. Improved cooling and higher turbine inlet temperature

Blade cooling was re-evaluated, and it was found that lower cooling flow could be utilized (Task 1.4). Additionally, the evaluation of a higher turbine inlet temperature (1350°C) was conducted. For a TIT of 1250°C, a cooling mass flow of 0.3629 kg/s was used, while for a TIT of 1350°C, a cooling mass flow of 0.6025 kg/s was applied. The results of these simulations are shown in Figure 4.

Improved cooling clearly increased the efficiency of both the GT and GT+WHR for both turbine inlet temperatures. At 1250°C, GT efficiency increased 2.2% and GT + WHR 1.7% (case 9 vs case 10), while at 1350°C, the increases were 1.2% and 0.9% (case 12 vs case 11), respectively. Increasing the TIT to 1350°C without changing the cooling flow (case 9 vs case 12) improved GT efficiency by 2.2% and GT + WHR efficiency by 1.7%. However, when considering improved cooling (case 10 vs case 11), increasing the TIT improved GT efficiency only by 0.1% and GT + WHR by 0.8%.

A higher TIT requires more cooling flow, which diminishes the efficiency improvement. Increasing the TIT typically benefits cycles with higher maximum pressure, but this was not studied. To optimizing cycle efficiency, this aspect could be investigated in future tasks.

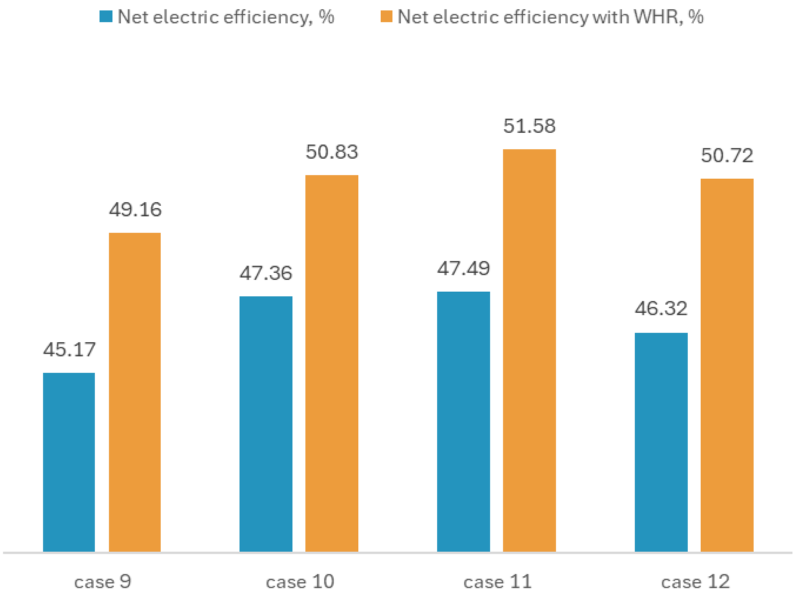

Figure 4. Net electric efficiencies of GT and GT+WHR.

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3. Conclusions

Twelve different cases were simulated to identify the most suitable and efficient cycle layout for the current project. First, the maximum pressure of the cycle was studied and 12 bar was found to be the most suitable pressure, balancing predicted efficiency and operational feasibility. Different blade cooling layouts were studied, and the option of extracting cooling flow after the first IC and directing the heated flow to the hot side inlet of the recuperator was found as the most efficient.

The design of the GT cycle is an iterative process and turbomachinery performances should be updated as the cycle-based design becomes more detailed. The turbomachinery pressure ratio and efficiencies were updated based on the mean line designs developed in Task 1.4. As part of the rotor dynamic design for the high-pressure shaft, also conducted in Task 1.4, it was recommended that the axial length of the shaft of the high-pressure spool be decreased. This led to a study of the impact of removing the second IC from the cycle. It was found that while removing the second IC decreased GT efficiency, the combined GT+WHR efficiency slightly increased due to higher thermal power and higher temperature available for the WHR. Therefore, the cycle without the second IC was chosen for further study.

Based on the cooling analyses from Task 1.4, it was found that reducing the cooling flow improved the cycle efficiency. Additionally, an increase in TIT to 1350°C was studied, and while it improved cycle efficiency, the gain was limited due to the higher cooling flow required.

The study of using film cooling is ongoing and its impact on cycle performance will be further investigated. Furthermore, the turbomachinery performance values will continue to be updated and their impact on cycle performance will be investigated. The generator cooling, along with the forces generated by the turbine, will also affect the design of the GT and WHR process. Therefore, thermodynamic cycle calculations will need to be revised as the project progresses.

With an expected efficiency exceeding 50%, the proposed system demonstrates a significant improvement over conventional internal combustion engines, offering a promising path toward more sustainable and efficient energy solutions.

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