MASTER

Technical-economic analysis of a 1MWe Rankine Compression Gasturbine cycle for electrical peakshaving

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Master Thesis

Technical-economic analysis of a 1MWe Rankine Compression Gasturbine cycle for electrical peakshaving

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Abstract

Throughout the years, the energy landscape has changed tremendously. Inclusion of more weather dependent renewable energy sources and electrification of industries have led to challenges in balancing the demand and supply of the electricity grid. Traditionally, steam turbine and generator cycles fulfill a dominant role in the supply of electricity. Although cycles are excellent for base load electricity generation, these face difficulties when used to deliver electrical peak loads. As an alternative to these, the rankine compression gas turbine (RCG) cycle is proposed.

Unlike a steam turbine and generator cycle, the RCG cycle makes use of an additional air cycle and free power turbine, giving it the capability to vary their electrical power output in a matter of seconds. The RCG-cycle is in an early stage of development and requires an additional technical and economical feasibility study to identify the technical and economical benefits with respect to a conventional steam turbine and generator.

This study has led to the development of two type of RCG-cycles. These cycles make use of the same type of steam turbine and compressor, but vary in the used type of power turbine. The first configuration makes use of four automotive turbocharger turbines and the second configuration is equipped with a hot gas expansion turbine. The operational conditions of the power turbine (pressure ratio, turbine inlet temperature) affects the technical and economical performance of the cycle. Hence for both configurations, two different operational conditions have been evaluated which leads to a total of four different RCG options. Consequently, these four cycles have been compared to a conventional steam turbine and generator cycle.

The comparison is done by use of two use cases. The first use case is based on a tar-granulate plant that is driven by waste heat. The second use case includes a metal fuels combustion plant that generates heat by the combustion of iron particles. Major distinction between these two use cases lays within the fuel costs. In the tar-granulate case, the fuel costs are negligible since waste-heat is used. This doesn't hold for the metal fuels combustion case, that uses iron as a fuel source. The economical performance of these cycles is based on the Dutch unbalanced market. A market that is, in contrast to the conventional wholesale electricity market, characterized by highly volatile real time electricity prices.

During the selection of the most suitable option per use case, the interest of two primary shareholders have been taken into account. On one hand there is the plant owner that focuses primarily on financial performance and on the other hand there is the society that desires peak shaving capability and low carbon pollution. Since these are conflicting criteria, a multi criteria decision making analysis is used to provide the most suitable option, either one of the four RCG cycles or the steam turbine and generator cycle. In the decision making analysis three different perspectives have been used. These either lay more emphasis on the interest of the plant owner, the society, or provide an equally balanced perspective of interests. In both use cases, the RCG cycle with the four automotive turbocharger turbines has been proposed as the best solution.

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1 Introduction

From the 1900's, Rankine cycles, as displayed in figure 1, are commonly used for electricity generation in (centralized) power plants. Solid fuels such as biomass, coal or lignite are combusted in a furnace to generate heat. Through the use of a boiler, heat is transferred to water which undergoes a phase change to steam. The steam is expanded in a steam turbine where the steam pressure is reduced and mechanical work is generated on the steam turbine axis. When an electrical generator is coupled to the steam turbine axis, the mechanical power of the turbine is used to generate electricity. At the outlet of the steam turbine, the remaining steam goes through a condenser that condensates the steam back to water. The heat from the condenser is often used for district heating purposes and hence the rankine cycle is commonly applied for combined heat and power (CHP) applications [32].

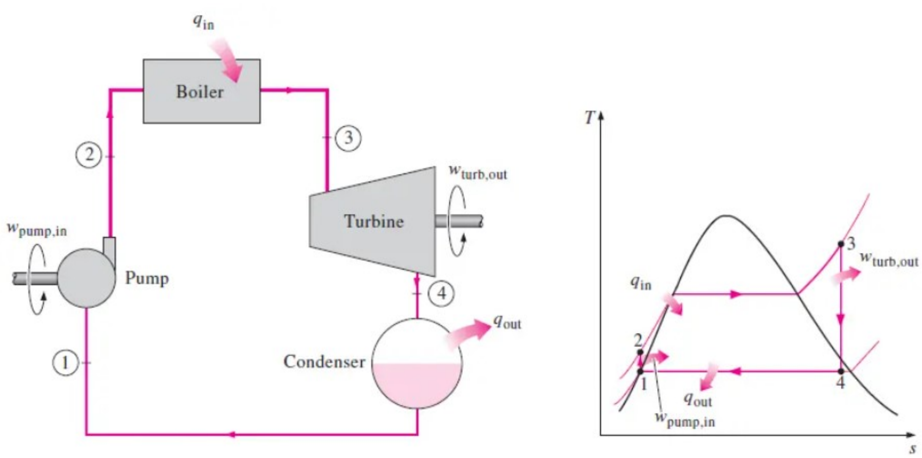


Figure 1: Layout of conventional rankine cycle and pressure enthalphy diagram, adopted from [32]

The majority of steam turbine and generators are applied in centralized power systems for solely base load operation. These plants make use of large steam turbines that can vary in their power output between 1.5 to 4% of the nominal power output per minute. This discrepancy in variation of the nominal power output can be attributed to the type of furnace fuel, feed time and used boiler. Besides that, maximum variation in power output depends on the inertia of the steam turbine.

Over the past decades the energy landscape has changed tremendously. Inclusion of more weather-dependent renewable energy sources in the electricity grid has led to high variations in the electricity supply. In the same period, electrification of industries has led to a higher consumption of electricity. The potential mismatch in supply and demand can lead to problems in the transmission of electricity.

To fulfill demand during electricity peaks, so-called "peak power plants" or "load following plants" are implemented. Traditionally, gas turbine engines, diesel gen-sets and hydroelectric power plants are used. Additionally, energy storage solutions such as chemical energy storage (Lithium-ion, Lead Acid, Redox flow) and compressed air energy storage are under development and may play a role in the future energy landscape during periods of peak electricity demand [21].

The majority of electricity suppliers trade on a forward market (e.g. Merit order dispatch, economic dispatch), commonly referred to as the "wholesale electricity market". These markets are characterized by long lasting contracts. In these markets, consumers and generating parties are obligated to deliver electricity for a fixed fee. To fulfill in the demand of peak electricity in a shorter period of time, electricity transmission operators (TSO's) have introduced short term cost-driven market models where prices are based on the total electricity supply and demand. An example of such a market is the unbalanced market, where electricity prices vary in real time. An example of the electricity market prices of this market is shown in figure 2.

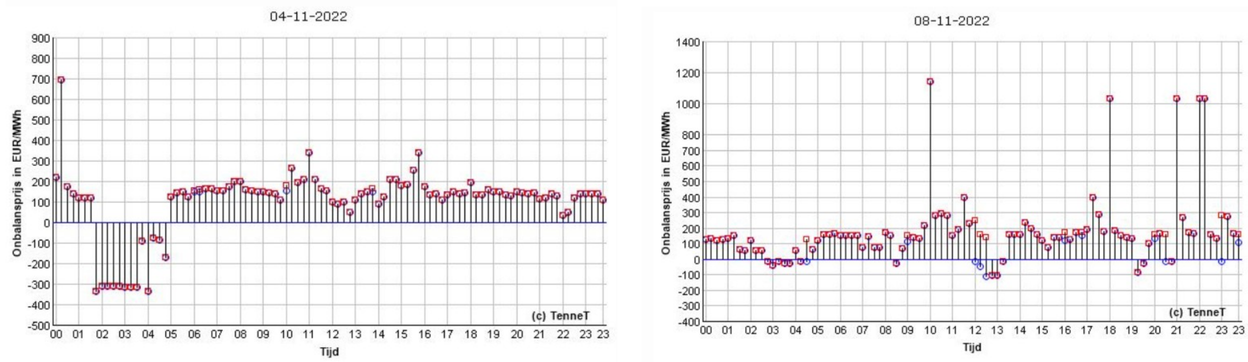


Figure 2: Unbalance market energy prices per MWh with 15 minutes interval at two typical days in November from transmission system operator (TSO) Tennet [31].

1.1 Rankine compression gas-turbine

In 2006, a patent was filled [14] for a new type of fast responding combined steam -and gas turbine cycle: The Rankine Compression Gas-turbine (RCG). Unlike a STG-cyle it is able to vary (electrical) power output in a matter of seconds through the use of an additional air cycle. A schematic overview of the lay-out is displayed in figure 3.

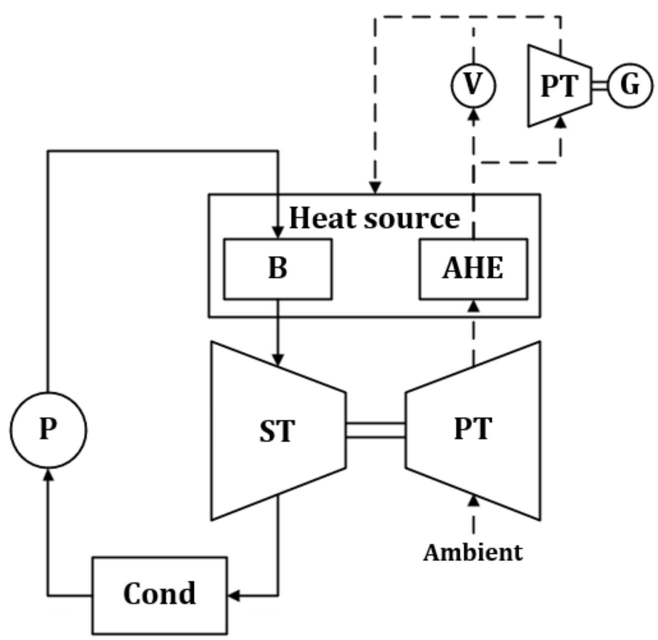


Figure 3: Layout of the RCG-cycle with steam turbine (ST), compressor (C), power turbine (PT), condensor (Cond), boiler (B) and air heat exchanger (AHE) and by-pass valve (V). The flow of water/steam is displayed by a solid line and the flow of air is displayed by a dashed line.

The RCG-cycle makes use of an ordinary steam turbine which in the same principle as a conventional rankine cycle, is driven by steam from a boiler. In a conventional rankine cycle, a steam turbine is used to drive an electrical generator. In the RCG cycle, a steam turbine is used to drive a compressor. The compressor compresses ambient air and guides the air through a heat exchanger which is heated by a heat source. The heated compressed ambient air is thereafter expanded through a power turbine. The power turbine is attached to a generator which converts mechanical work into electrical energy. The amount of air that is expanded in the power turbine, and thus the amount of electricity that is generated by the combined cycle, can be varied by use of a by-pass valve that is placed parallel to the power turbine.

The heat that is present after expansion can be fed back into the heat source by desire to minimize heat loss. Both the steam turbine and compressor combination, and the power turbine and generator will remain at their nominal angular velocity during zero electrical power output. However, the turbo-machinery still consumes a small amount of energy because a marginal amount of torque is required to overcome friction losses. The response time of the system is equal to the response time of the bypass valve. Via this principle 0-100% electrical power output can be delivered in a matter of seconds.

The RCG-cycle can be powered by any type of heat source including waste heat or gasified/solid fuels. The heat sources that power the boiler and air-air heat exchanger do not necessarily have to be taken from the same heat source. Besides a "stand-alone" version, where the RCG-cycle consist out of a boiler and air-air heat exchanger, it is also possible to present the RCG-cycle as an add-on to an existing boiler and furnace system. In this configuration, the RCG-cycle does not include a boiler, pump and condensor but acquires their steam directly from an external boiler.

1.2 Previous work and current status of the technology

A study for the development of a 1MWe RCG-unit was performed in 2008 [27]. In this study a 1MWe RCG-cycle was designed from of the components which proved to be technologically feasible. In addition, the economical feasibility has been assessed for 2.5 and 10MWe with a rough estimate of the initial investment costs varying between €450-720k per MWe. Costs per kWe per year have been established for a use case where the RCG-cycle is fueled by natural gas. Besides a technical and economical analysis, the study included the development of a 5kWe lab-prototype, located at Eindhoven University of Technology. Consequently, the 5kWe RCG-cycle prototype was installed in a wood milling factory near Eindhoven [7] as displayed in figure 4.

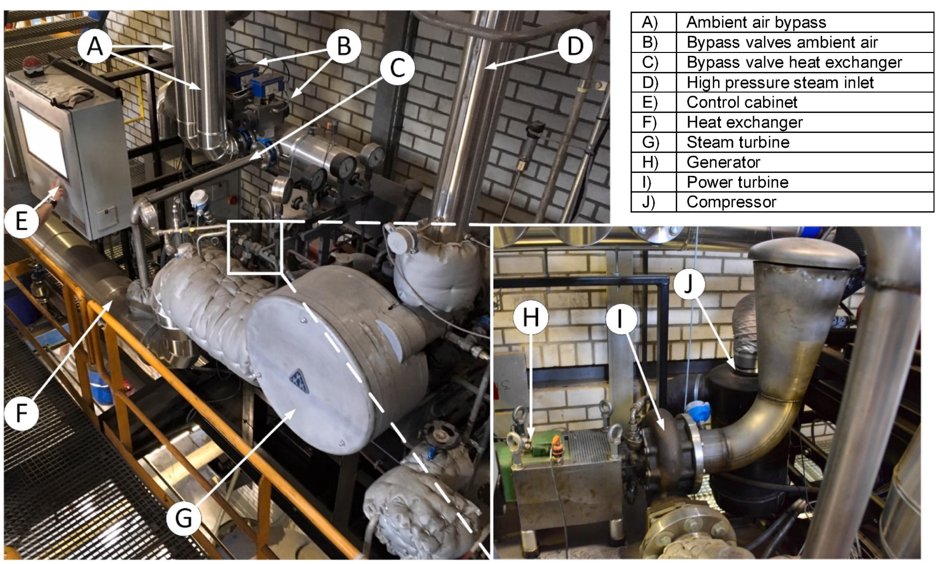


Figure 4: 5kWe RCG prototype at Hout industrie schijndel. Figure is adopted from [7].

The system was configured as an add-on to an existing biomass furnace and boiler system. In this configuration, a Siemens kk&k impulse steam turbine is driven by high pressure super-heated steam (28bar/400C) which is generated in the factory's boiler. Expanded (low pressure) steam at the outlet of the steam turbine is applied to factory's low pressure steam connection (1.4bar). Between the outlet of the steam turbine and the factory's low pressure steam connection, a tube cross flow heat exchanger was installed. The steam turbine was used to drive a centrifugal turbo compressor. The compressed ambient air was guided through the heat exchanger and hence the heat that was present at the steam turbine outlet was used as a heat source for the pressurized air cycle. A by-pass valve located parallel to the heat exchanger was installed so the amount of heat captured from the outlet of the steam turbine could vary.

The heated pressurized ambient air was consequently expanded over a radial power turbine. At the axis of the power turbine a high speed induction generator was mounted for the generation of electricity. The cycle was capable of varying electrical output between 1 to 5kWe within 4 seconds, without interrupting the factory's steam supply. The main limitation in total response time was attributed to the response time of the heat exchanger by-pass valve. Through the use of this first industrial prototype, the unique selling points of the RCG cycle were demonstrated in an industrial environment.

At the moment, a 40kWe prototype is designed and fabricated, which will be installed in the same factory. In this configuration the steam turbine, compressor and power turbine will be re-used. The 5kWe induction generator will be replaced with a 40kWe version. In contrast to the 5kWe cycle, the 40kWe version will acquire its heat from the hot flue gasses of the factory's furnace. The 40kWe RCG-cycle, as an add-on the an existing furnace and boiler system can demonstrate the RCG-cycle as a potential commercial product.

1.3 Goal

Heat Power B.V., the company that is the owner of the RCG patent, is keen on commercializing the RCG technology. The last feasibility study, performed in 2009, focused on a RCG cycle with a power output of 1MWe based on the combustion of natural gas. Meanwhile, innovations in the turbo-machinery market may have led to different component options and variations in the cost price of components. These developments may alter the economical feasibility of the RCG-cycle. In addition, more appealing use cases arised which are worth investigating.

1.4 Research objectives

The main research objectives in this research are:

- **Overview of different component options**
  Although an RCG-cycle basically consist of the same category of turbo-machinery components (compressor, steam turbine, power turbine), every category of turbo-machinery has a variety of different options that can be selected. At first, a literature study is performed to provide an overview of different options per turbo-machinery component and asses their technical limitations and cost prices. From the acquired options, a selection of the most promising component options is made.

- **Propose RCG configurations from commercially available components**
  Although a literature study provides a good indication of the cost price and technical performance of components, a RCG-cycle can only be constructed from components that are available on the market. By use of a market study and meetings with suppliers, more detailed information about the technical performance and cost prices of different turbo-machinery components has been established. Technical performance is gather from data-sheets while invoices can lead to more accurate cost price estimations. From the proposed components, a variety of different RCG cycles will be constructed that vary in both technical and economical performance.

- **Industrial applications**
  The technical and economical performance of an RCG-cycle highly depends on the application. Since the RCG-cycle can be powered by a variety of different heat sources it is applicable to different industries. This research accesses the technical and economical feasibility of an RCG cycle for two different use cases:

  1. Tar-granulate recycling plant, fueled by waste heat.
  2. Metal fuels combustion plant, fueled by iron fuel.

- **Multi-criteria analysis**
  The decision making process of the most suitable RCG option is subjected to a variety of different performance indicators. Since multiple shareholders are involved that on their turn may have conflicting interests, a multi-criteria analysis is used to determine the most suitable cycle per use case.

2 Thermodynamics

In this chapter, the thermodynamic relations of the rankine compression gasturbine (RCG) cycle and a conventional steam turbine and generator cycle are introduced. These thermodynamic relations have been adopted from [26]. Two different steam cycles configurations have been taken into consideration. In the first steam cycle, the steam turbine is implemented as an add-on to an existing boiler and furnace system. The second steam cycle is displayed as a stand-alone close loop rankine cycle. Both cycles are displayed in figure 5

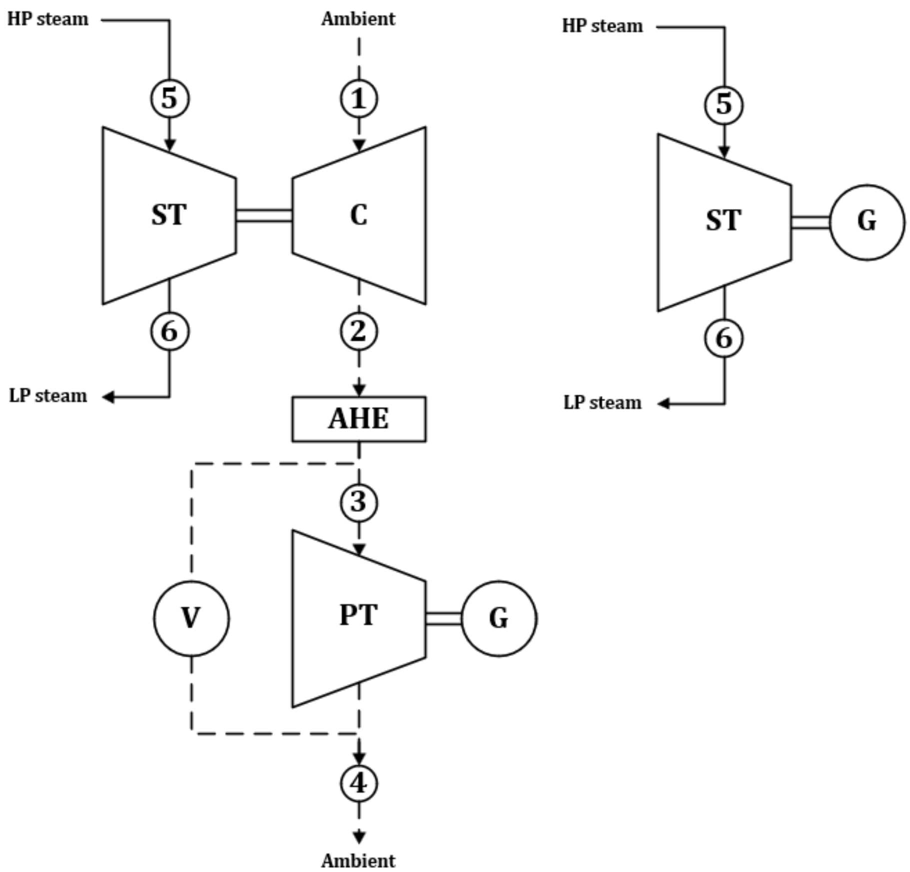


Figure 5: Steam turbine and air cycle of an rankine compression gas turbine cycle (RCG) on the left and a conventional steam turbine and generator (ST) displayed on the right. The rankine cycle contains a steam turbine (ST), compressor (C), air heat exchanger (AHE), power turbine (PT), generator (G) and by-pass valve (V). The conventional steam turbine-cycle has solely a steam turbine (ST) and generator (G). In this diagram, the low -and high pressure steam lines are solid and the air line is represented as a dotted line.

2.1 Air cycle

Between states 1-2, ambient air is compressed by use of compressor. The compressor is directly coupled to a steam turbine. Assuming negligible losses due to the shaft connection, the work of the compressor can be described by:

$$P_c = P_{st} \tag{1}$$

The work that is required by the compressor can be written as the temperature difference between the inlet -and outlet of the compressed air $(T_2 - T_1)$, the mass flow through the compressor $(\dot{m_{air}})$, the mean specific heat $(\bar{Cp}_{1,2})$ and the compressors isentropic efficiency (n_c), as is stated in equation 2. The work that is delivered by the power turbine after expansion of the heated ambient air between state 3-4 is given by equation 3. It is written as a function of the mean specific heat of the expanded air ($\bar{Cp}_{3,4}$)), the temperature difference of the air ($T_4 - T_3$) and turbines isentropic efficiency (n_{pt}).

$$P_c = \frac{\dot{m_{air}} \cdot \bar{Cp}_{7,8} \cdot (T_8 - T_7)}{n_c} \tag{2}$$

$$P_{pt} = n_{pt} \cdot \dot{m_{air}} \cdot \bar{Cp}_{9,10} \cdot (T_{10} - T_9) \tag{3}$$

The air in the cycle is considered to be an ideal gas, free of moist. The relation between inlet and outlet pressure and temperature of ideal isentropic compression/expansion are a function of the specific heat ratio, $k = \frac{Cp}{Cv}$ as can be noticed from equation 4.

$$\frac{T_{i+1}}{T_i} = (\frac{P_{i+1}}{P_i})^{\frac{k}{k-1}} \tag{4}$$

By integrating equation 4 in equation 2 and 3, the work that is required from the compressor and the work that is delivered by the power turbine can be written as equations 5 and 6.

$$P_c = \frac{\dot{m_{air}} \cdot \bar{Cp}_{1,2} \cdot T_1((\frac{P_2}{P_1})^{\frac{k-1}{k}} - 1)}{n_C} \tag{5}$$

$$P_{pt} = \dot{m_{air}} \cdot \bar{Cp}_{3,4} \cdot T_3 \cdot n_{pt} \cdot (1 - (\frac{P_4}{P_3})^{\frac{k-1}{k}}) \tag{6}$$

The outlet temperature at the outlet of the compressor T_2 and the temperature at the outlet of the power turbine T_4 can also be derived by integration equation 4 into equation 2 and 2. These are displayed in equation 7 and equation 8.

$$T_2 = T_1 \cdot \frac{(1 + (\frac{P_2}{P_1}^{\frac{k-1}{k}} - 1))}{n_c} \tag{7}$$

$$T_4 = T_3 \cdot (1 + n_{pt} \cdot (1 - (\frac{P_4}{P_3})^{\frac{k-1}{k}})) \tag{8}$$

A electrical generator is directly coupled to the power turbine for the generation of electricity. In the steam turbine - generator cycle, an electrical generator is directly coupled to a steam turbine. The electrical power output of the generator is limited by the electrical generators efficiency (n_g). The required turbine power of both cycles can therefore be calculated by use of equations 9 and 10 .

$$P_{pt} = \frac{P_e}{n_g} \tag{9}$$

$$P_{st} = \frac{P_e}{n_g} \tag{10}$$

2.2 Steam cycle

In figure 6 two steam cycles are displayed. On the left, a conventional closed-loop rankine cycle is displayed. On the right, an open-loop steam cycle is displayed. In this report, when the RCG is referred to as "stand-alone", it makes use of the steam cycle that is displayed on the left. When the cycle is executed as an add-on, the steam cycle is configured as an open-loop cycle, displayed on the right.

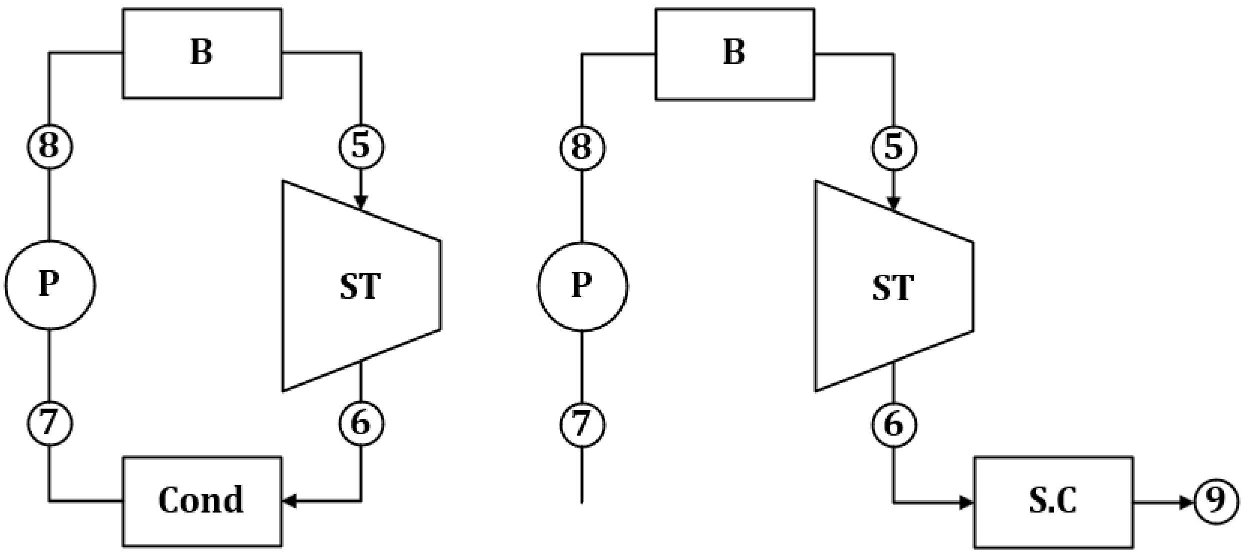


Figure 6: On the left, a closed loop steam cycle and on the right, an open-loop steam cycle. Both steam cycles contain a pump (P), boiler (B), steam turbine (ST) and condenser. In the closed-loop cycle, steam after expansion is condensed between state 6-7, while in the open-loop cycle, heat that is present at the outlet of the steam turbine is fed into an external plant where the steam is consumed (SC) for an industrial process between state 6-9.

2.2.1 Closed-loop steam cycle

The rankine cycle is a closed loop thermodynamic cycle of a constant pressure heat engine that is generates both heat and work. The fluid in the rankine cycle is subjected to four thermodynamic processes.

Between state 7-8, the liquid undergoes isentropic compression by use of a pump. The amount of work that is delivered by the pump (W_{pump}) depends on the mass flow of the fluid ($\dot{m_{fl}}$) and the enthalphy difference before (h_7) and after compression (h_8) as is given by equation 11.

$$W_{pump} = \dot{m_{fl}} \cdot (h8 - h7) \tag{11}$$

Between state 8-5, liquid is subjected to isobaric heat addition in the boiler. At first the liquid becomes saturated, consequently evaporates and lastly becomes superheated vapor. The amount of heat that is required to generate superheated vapor from compressed liquid is calculated by use of the mass flow of the fluid ($\dot{m_{fl}}$) and the fluids enthalphy difference between the inlet (h_8) -and outlet (h_5) of the boiler, as is stated by equation 12.

$$Q_b = \dot{m_{fl}} \cdot (h_8 - h_5) \tag{12}$$

Between state 5-6, the steam undergoes isentropic expansion in the steam turbine. It the closed loop steam cycle, it is common practice to ensure that the vapor fraction at the outlet of the steam turbine is not lower than 0.9 since the water droplets that are present at lower vapor fractions can lead to damage of the turbine blades [18]. Since it is assumed that the steam turbine undergoes isentropic expansion, it is subjected to an isentropic efficiency. The actual enthalphy at the steam turbine outlet can be calculated by use of the following equation:

$$h6_{actual} = h5 - n_{st} \cdot (h5 - h6s) \tag{13}$$

The amount of work that is delivered by the steam turbine can be calculated by use of the mass flow of the superheated vapor and the actual enthalphy difference between the inlet and the outlet of the steam turbine:

$$W_{st} = \dot{m_{fl}} \cdot (h6_{actual} - h1) \tag{14}$$

Between state 6-7, the saturated liquid and vapor substance undergoes isobaric heat rejection. In this process, the substance is cooled up to enters the liquid phase. The amount of heat that is extracted from the saturated liquid and vapor depends on the enthalphy difference between the outlet of the steam turbine (h_6) and the pump inlet (h_7) i.c.w. the mass flow ($\dot{m_{fl}}$) of the fluid, as is given by equation 15.

$$Q_{cond} = \dot{m_{fl}} \cdot (h6_{actual} - h1) \tag{15}$$

2.2.2 Open-loop steam cycle

When the RCG is configured as an add-on to an existing plant, equations 11 till 14 remain valid. The major difference is that the steam at the outlet of the steam turbine is not condensed but will be consumed by an external plant. Industrial low pressure steam consumers typically require dry steam. Therefore is assumed that the vapor fraction at the outlet of the steam turbine is equal to 1. The amount of heat that is present at the outlet of the steam turbine can be calculated by use of the enthalphy difference between the steam at the steam turbine outlet (h_6) and the enthalphy of the steam after consumption (h_9).

$$Q_{Steam,Industry} = \dot{m_{fl}} \cdot (h_9 - h_6) \tag{16}$$

2.3 Fuel

The amount of heat that is required to power the RCG cycle is a combination of the heat that is consumed in the steam turbine (Q_{st}), the amount of heat that is used in the air cycle (Q_{ahe}) and the amount of heat that is recovered at the outlet of the power turbine (Q_{pto}) as can be noticed from equation 17. Due to the absence of an air cycle in the conventional steam turbine and generator cycle, the required heat solely depends on the steam turbine for that configuration, as is given by equation 18.

$$Q_{fuel_{RCG}} = Q_{st} + Q_{ahe} - Q_{pto} \tag{17}$$

$$Q_{fuel_{STG}} = Q_{st} \tag{18}$$

When the stack temperate (the temperature at the outlet of the combustion cycle) is not equal to the temperature of the air at the inlet of the combustor, the relation between the amount of energy that is extracted from the fuel by the combustion process (ΔE) is affected by a combustion efficiency (n_{comb}). The amount of fuel that is needed for the required heat can be calculated by use of equation 19.

$$Q_{fuel} = \frac{\Delta E}{n_{comb}} \tag{19}$$

The heat that is required to power the steam turbine depends on the steam turbines power output and isentropic efficiency:

$$Q_{st} = \frac{P_{st}}{n_{st}} \tag{20}$$

The amount of heat that is required by the air heat exchanger is a function of the mass flow of air (m_{air}), the temperature at the outlet of the compressor (T_2), the turbine inlet temperature (T_3) and the mean specific heat ratio ($C\bar{p}_{2,3}$):

$$Q_{AHE} = \dot{m_{air}} \cdot C\bar{p}_{2,3} \cdot (T_3 - T_2) \tag{21}$$

And lastly, the amount of heat that is present at the outlet of the power turbine is a function of the mean specific heat ratio $\bar{C_p}$, the mass flow of air and the difference between the turbine outlet temperature T_8 and the reference temperature T_0.

$$Q_{gto} = \dot{m_{air}} \cdot \bar{C}p \cdot (T_4 - T_0) \tag{22}$$

The amount of heat that is required from the boiler is related to the heat that is consumed by the steam turbine (Q_{st}) and the amount of heat that is left at the outlet of the steam turbine (Q_{sto}), as is displayed by equation 23.

$$Q_B = Q_{st} + Q_{sto} \tag{23}$$

The amount of heat that is present in the outlet of the steam turbine depends on the mass flow of the fluid ($\dot{m_{fl}}$) and varies per steam-cycle. For the closed-loop cycle, it is related to the enthalphy difference at the outlet of the steam turbine (h_6) and before compression (h_7). In the open-loop cycle it is related to the enthalphy difference between the outlet of the steam turbine (h_6) and the outlet enthalphy of the fluid after consumption (h_9). For the closed loop cycle these relations are given by equation 24 while for the open-loop cycle these are displayed by equation 25.

$$Q_{sto} = \dot{m_{fl}} \cdot (h_7 - h_6) \tag{24}$$

$$Q_{sto} = \dot{m_{fl}} \cdot (h_9 - h_6) \tag{25}$$

The heat to electricity efficiency of the RCG-cycle is equal to the amount of consumed heat w.r.t. the amount of generated electricity, as is given by equation 26.

$$n_{H2E} = \frac{P_{gen} - W_{pump}}{Q_{fuel}} \tag{26}$$

3 Technical and economical analysis of RCG components

An RCG cycle is constructed from a variety of turbo-machinery components such as a steam turbine, compressor, power turbine and electrical generator. This chapter displays the outcome of a literature study that reviews the technical performance and estimated cost prices of different turbo-machinery options.

3.1 Steam turbine

A steam turbine is used to convert heat from the steam into mechanical work. This is done in two consecutive ways. At first, the pressure of the steam is reduced, which results in an increase of the steam velocity and thus increases the kinetic energy of the steam. Then, the kinetic energy from the steam flow is transferred into mechanical work as the steam collides onto the turbine wheel. Based on the method of steam expansion [2], steam turbines can be classified into three types: Impulse steam turbines (Stand alone Curtis), reaction steam turbines (Stand alone Rateau) and combined impulse and reaction steam turbines (Rateau and reaction). Steam turbines are often selected based on the power output and angular velocity of the driven component. For the mentioned type of steam turbine options, their application range in terms of angular velocity and power output is displayed in figure 7.

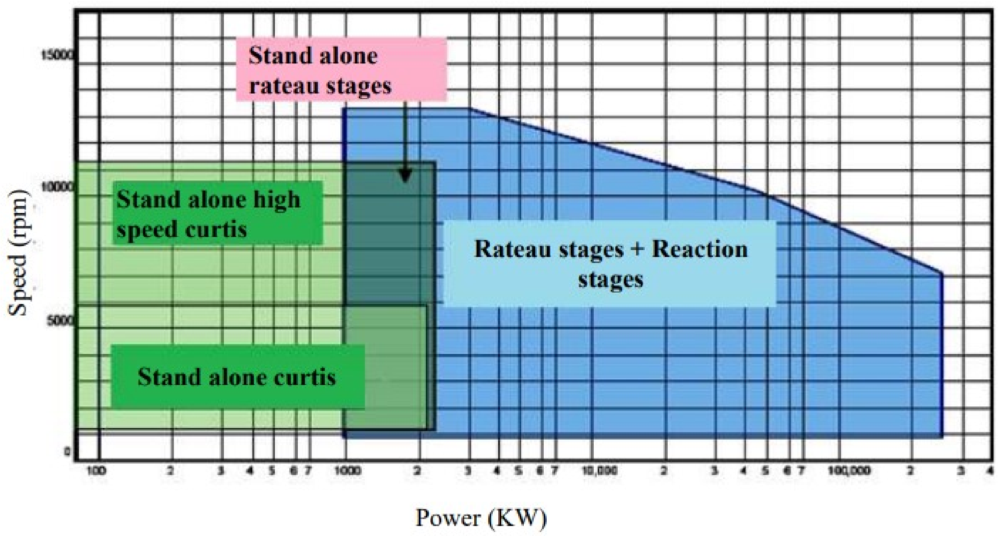


Figure 7: Power and speed operating range of an impulse (Curtis), reaction (Rateau) and combined impulse and reaction (Rateau + reaction) steam turbine. Figure has been adopted from [18].

In an impulse steam turbine, steam enters the turbine via a set of stationary nozzles. As the steam travels through the nozzle, the steam is subjected to a pressure drop that results in a higher velocity of the steam flow. The steam blows on top of a set of bucket shaped blades that are attached to a turbine wheel and as a result, the blades and turbine wheel start to spin. A reaction turbine makes use of a set of stationary fix guide vanes and rotating runner blades. Steam enters the turbine via the guided inlet vanes and leaves the turbine via the runner blade. During this process, the steam manoeuvres between the guided inlet vanes which leads to a pressure drop. The steam is at maximum velocity before it reaches the runner blades.

Due to the geometry of the vanes, the pressure drop per stage is significantly higher for an impulse steam turbine compared to a reaction turbine. As a result, a reaction turbine requires 70-80% more stages for the same power output which leads to a much longer an heaver steam turbine. Due to the construction of the blades and the required extra stages, a reaction turbine is characterized by higher initial investment costs and higher maintenance costs compared to an impulse steam turbine. On the other hand, reaction steam turbines are able to achieve efficiencies up to 80%, while the efficiency of the impulse steam turbine is limited to around 70% [2].

From figure 7 can be noticed that reaction turbines are commonly used for powers up to 2MW. Although these turbines can be used for lower power outputs, the use of a reaction turbine would result in a much more expensive steam turbine. Despite the lower efficiency, the impulse steam turbine is selected as the most suitable turbine for the 1MWe RCG cycle. These turbines are available in low speed (6krpm) and high speed (12krpm) configuration. The choice between these versions depend on the required angular velocity of the compressor.

Impulse steam turbines are available as back pressure and condensing variations. Impulse steam turbines are often used in combined heat and power (CHP) applications, where a condenser is mounted at the outlet of the steam turbine to acquire remaining heat. Condensing steam turbines are, as the name suggests, able to condense the steam to vapor fractions below 0.9 and therefore the enthalphy of the vapor can be reduced which results into a higher power output for the same steam flow.

3.1.1 Steam turbine costs

The cost price of impulse and back pressure steam turbine for powers between 100kW and 1MW are displayed in figure 8. From this overview can be noticed that the back pressure steam turbine are a significantly cheaper option than the condensing steam turbine. The choice between an condensing and impulse steam turbine is commonly made based on the required fuel and heat. In applications where only electricity is required and fuel costs are high, it can be lucrative to install a condensing steam turbine since the majority of energy is transferred into work. When fuel prices are low or when besides electricity also heat is required, a back pressure steam turbine is most convention. In this research, a back pressure steam turbine is taken into consideration due to it's low initial investment cost.

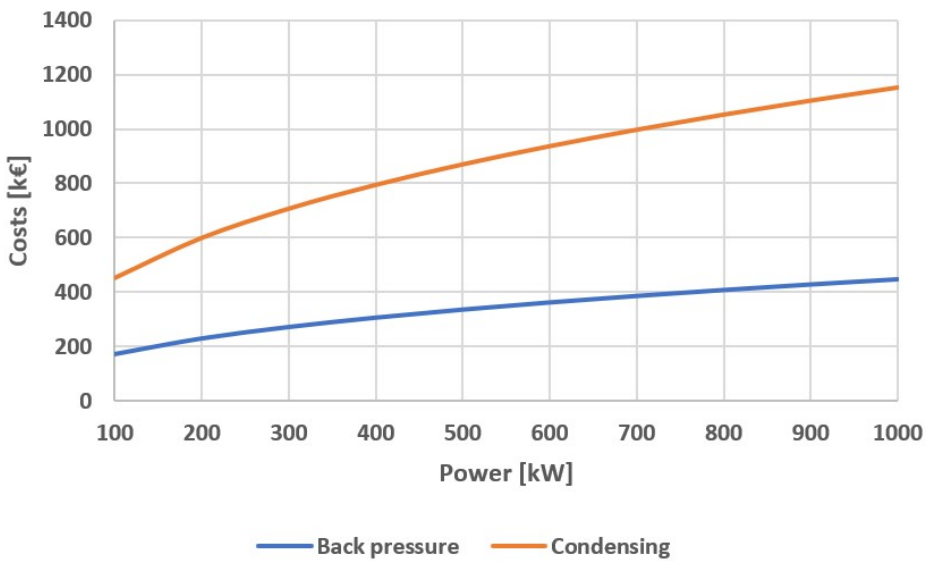


Figure 8: Estimated equipment cost price of an impulse steam turbine per kW mechanical output power. Cost prices have been addopted from [36] and corrected for inflation.

3.2 Compressor

A compressor uses mechanical power to extract and increase the pressure of a gas. compressors are mainly selected based on two technical parameters: Their maximum pressure ratio and volumetric flow. In figure 9, the working area of different compressor options is displayed, based on these two parameters. In general, compressors can be classified in two categories; positive displacement and dynamic compressors. While positive displacement compressors typically generate a constant volumetric flow, dynamic response compressors operate with a constant outlet pressure [15].

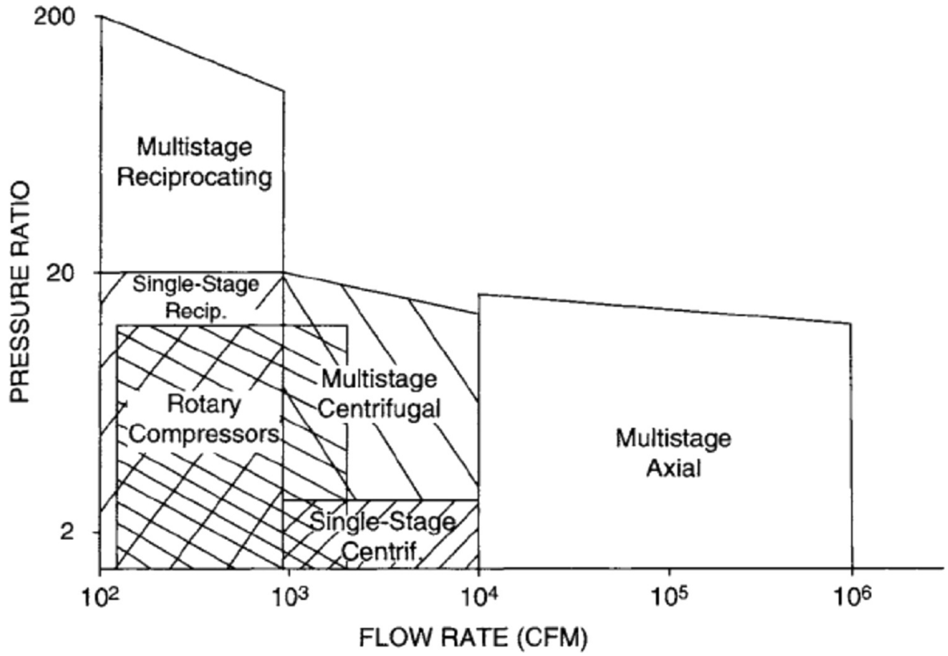


Figure 9: Application area of different compressors based on discharge pressure and capacity, adopted from [13]

3.2.1 Positive displacement compressors

In positive displacement compressors, air can be compressed by use of pistons (reciprocating compressor) or a set of rotors (rotary screw compressor). In a reciprocating compressor, air is sucked into a compression chamber, that is then closed from the atmosphere. A piston is used to decrease the volume of the compression chamber, hence increasing the air pressure. When the maximum pressure ratio is reached, the compression chamber is opened and compressed air is released. The use of pistons in reciprocating compressors enables high pressure ratio's, but limited capacity due to the volume of the compression chambers. In rotary screw compressors, male and female rotors turn in opposite directing. In this principle, air is sucked into the compressor which is on their turn compressed between the rotors. Hence, the volumetric flow of the screw compressor is proportional to the rotational speed of the drive. The principle enables rotary screw compressors to process higher flows than reciprocating compressors, but limits the pressure ratio. Small positive displacement compressors are often integrated in compressed air generation systems. Applied in larger industrial applications, positive displacement compressors are used in oil refinery, chemical industries and natural gas plants. There these are commonly used for the liquefaction of gas. Positive displacement compressors are known to produce a pulsating airflow which is not desired for turbine expansion. And although these compressors are known to be cheap, these are not further taken into consideration [15].

3.2.2 Dynamic compressors

In dynamic compressors, air is drawn from the ambient in a compression impeller at high velocities, which increases the kinetic energy of the air. When the air reaches maximum velocity, it is then discharged through a diffuser where the kinetic energy is transferred into static pressure. Figure 9 displays three types of dynamic compressor options. Multistage centrifugal and axial compressors are commonly applied in applications that require high pressure ratio's and are executed with multiple stages. Since relatively low pressures are required for the RCG cycle, these compressors are considered to be expensive options. Therefore, the single centrifugal compressor is taken into further consideration. Single stage centrifugal compressors can reach angular velocities as high as 32krpm, but can be executed with an internal gearbox when driven by and electrical motor. [30].

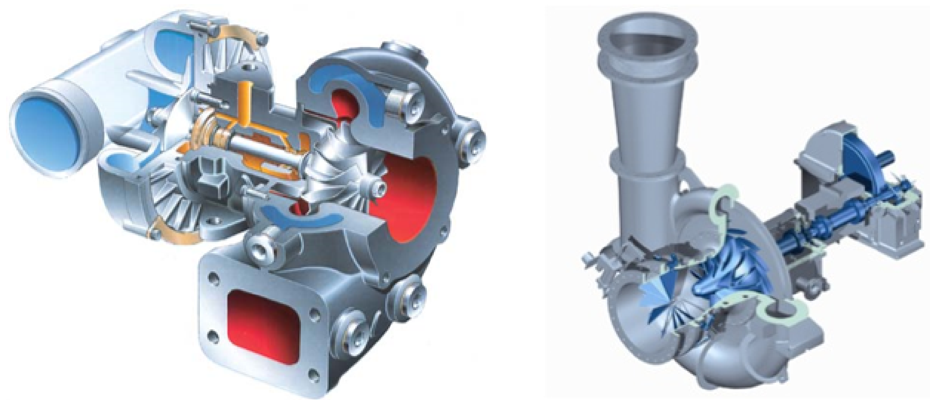


Figure 10: Automotive turbo charger (l) and single stage centrifugal compressor (r). Figures on the left adopted from [29] and figure on the right acquired from [1]

Besides the single centrifugal compressor, a turbo charger compressor is a promising compressor option, mainly due their low initial investment costs. Turbo charger compressors are sold as a set containing both compressor and expansion turbine and are used to enhance the performance of diesel or petrol engines. Since both turbine and compressor are coupled by an axis, the compressor part can be uncoupled from the turbine. Due to their compact size, these type of compressors are characterized by their high angular velocity and limited volumetric inlet flows. An overview of the main characteristics of the compressor options is displayed in table 1.

Type	Flow [m3/s]	r [-]	w [krpm]	n_{max}[%]
Single stage centrifugal	0.25 - 25	3	1.8 / 32	85
Turbocharger compressor	0.25 - 1.5	5	30 - 70	75

Table 1: Overview of compressor characteristics. Single stage centrifugal performance parameters have been adopted from [16] and turbocharger compressor parameters have been gather from [10]

Turbocharger compressors are equipped with static diffuser vanes. Due to this, the relation between efficiency, volumetric flow and pressure ratio is fixed. These relations are displayed in compressor maps, an example of such a map is displayed in figure 11. For different angular velocities, the compressor map shows a fixed relation between the pressure ratio and volumetric flow. The corresponding isentropic efficiency is given in so called efficiency islands. The compressor can't be operated outside their surge limit since this leads to aerodynamic instability that introduces a oscillating or reversed airflow can lead to permanent damage. Turbocharger compressors are available in a variety of sizes which all have different efficiency maps. Centrifugal compressors are often equipped with guided inlet vanes, which controls the angle of the diffuser vanes. The main benefit of this principle is that the maximum efficiency can be achieved at the entire specified pressure ratio and volumetric flow range of the compressor.

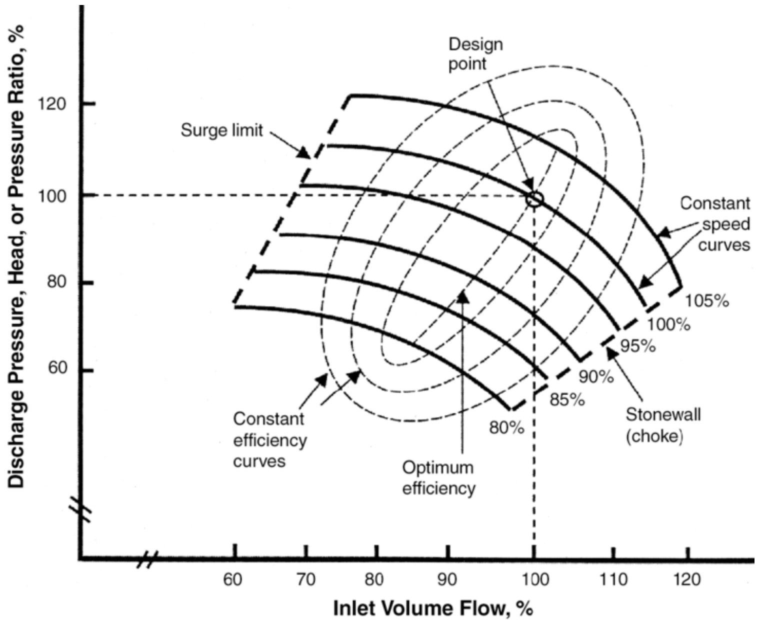


Figure 11: Stable operating region of dynamic compressors, adopted from [1].

3.2.3 Compressor costs

The cost price of the single stage centrifugal compressor and the automotive compressors are displayed in figure 12. The cost price of the automotive turbocharger compressor is size dependent and a commercial variant with a mass flow of 1.5 kg/s is available for €10k.

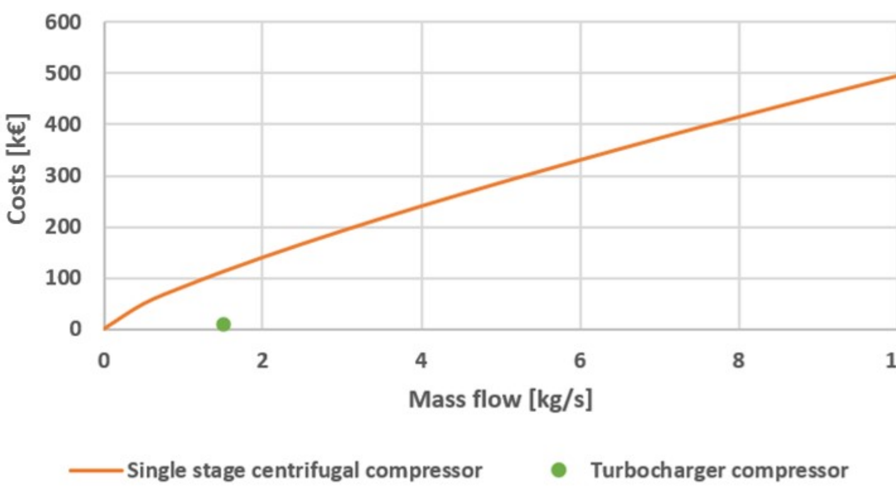


Figure 12: Compressor cost price as a function of the mass flow of the single stage centrifugal compressor (stainless steel), driven by an electrical motor. Cost of the turbocharger compressor (€10k) is based on the GTX5544, that is publicly available. Single stage centrifugal compressor costs have been adopted from [34] and corrected for inflation.

3.3 Power turbine

Power turbines are used to generate shaft power by the expansion of a gas. Their working principle is contrary to a compressor, nevertheless, there are lots of similarities in design/construction. The choice and application of a power turbine mainly depends on the expanded fluid, power output, turbine inlet temperature and pressure ratio. An overview of the most appealing power turbine options and their main performance parameters are listed in table 2.

Type	TIT [C]	r [-]	Power	n	$\omega[krpm]$
Turbocharger turbine	1050	4	400	77	70
Hot gas expander	500	12	MW's	85	6.7
FCC expander	760	3.5	MW's	85	6.7

Table 2: Main characteristics of the evaluated power turbines. The characteristics of the turbocharger turbine have been addopted from [10], the hot gas expander characteristics are based on a hot gas expansion turbine series from GE [[11] and the FCC expander characteristics have been addopted from Baker Hughes FCC expander serie [3].

During the compressor section was concluded that the compressor part of turbochargers are promissing, this also holds for the turbine part. These turbines are able to withstand gasses with a temperature of 1050°C and are available at low costs. Due to their small diameter, these turbines operate at high angular velocities. These turbines are equipped with fixed rotor vanes and hence the expandable mass flow and isentropic efficiency is related to the temperature and pressure of the air and the angular velocity of the turbine wheel. It is common practice to display these relations in turbine maps, an example is given in figure 13.

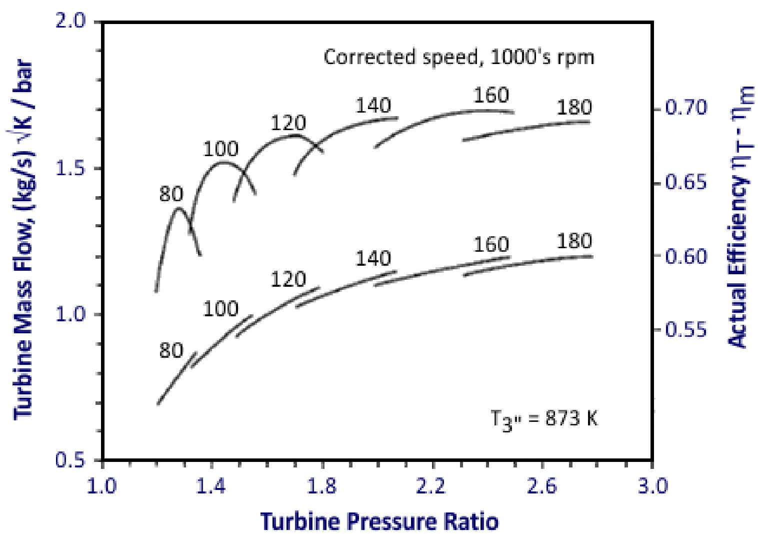


Figure 13: Turbocharger turbine map with turbine mass flow and turbine efficiency, adopted from [35]

Sole power turbines have been widely adopted for the generation of electricity from pressurized (waste) heat streams. These are often available as a package including turbine, gearbox and electrical generator. An important application area is hot flue gas reconvery systems, with temperatures typically up to 500C. A special application of flue gas recovery is fluid catalyctic cracking (FCC), a petroleum refinery process that converts crude oil into gasoline. The vanes of these turbines have a special coating and therefore these are optimized for higher turbine inlet temperatures (760 °C) and harsh industrial environmental conditions. Both type of turbines are available with guided inlet vanes and therefore these are able to expand gasses on their entire specified pressure - mass flow range, while maintaining an efficiency of 85%.

3.3.1 Power turbine costs

Estimated equipment costs prices of the hot gas expansion turbine, the FCC expander and a single turbocharger turbine are displayed in figure 14 per kW mechanical power output.

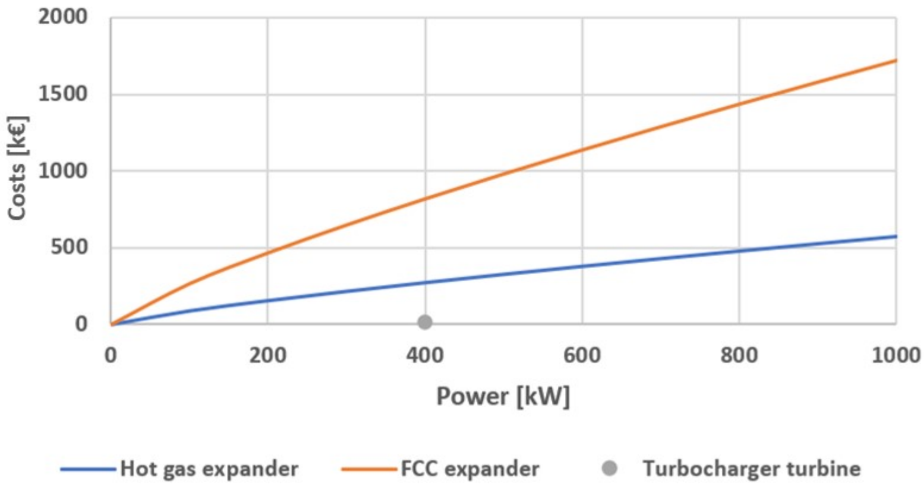


Figure 14: Cost of FCC turbine, hot gas turbine and automotive turbocharger as a function of their mechanical power output. The costs of the hot gas -and FCC expander have been adopted from [33] and corrected for inflation. The cost price of the automotive turbine from manufacturer Garret has a cost price of €10k and was established publicly available. A power output of 400kW is based on the maximum power output at 500C with a pressure ratio of 4.

3.4 Electrical generator

Electrical generators are broadly used for the generation of electricity from mechanical shaft power in combined heat and power, gas turbine engines, hydro-power, maritime applications etc. They consist out of a fixed stator and rotor that, as the name suggests, is able to rotate. A diagram of the electrical generator is displayed in figure 15. The stator contains copper winding's that are subjected to an external alternating (AC) voltage witch results into a switching magnetic field. By applying a force on the magnetic rotor between the electromagnetic field, an electrical current is generated in the coils of the stator.

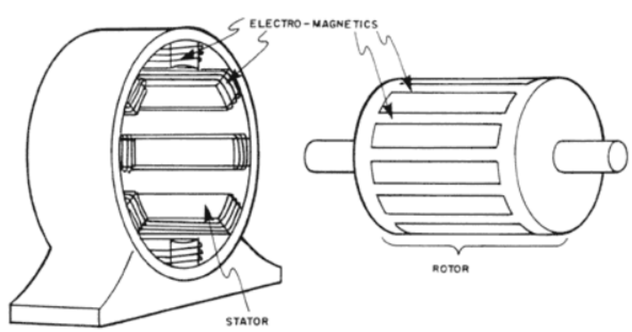


Figure 15: Electrical machine layout with elektromagnetic stator and rotor. Figure is adopted from [13].

The way the rotor is magnetized defines the two dominant types of the electrical generator: the induction generator (IG) and permanent magnet generator (PMG). The rotor core of induction generators consist out of stacked iron plates that are magnetized by use of an external DC voltage. In permanent magnet generators, the rotor core doesn't require and additional voltage as the rotor made from magnetic materials (SmCo, NdFeb).

One of the advantages of an induction generator is that it is relatively simple and inexpensive to produce. Because no permanent magnets are used, the components needed to build an induction generator are often cheaper than those for a permanent magnet generator. Another advantage of an induction generator is that it is often better suited for load variations. Permanent magnet generators are known to be slightly more efficient, since no energy is lost in the stator. This is because the energy needed to create the magnetic field is supplied by the permanent magnets in a permanent magnet generator, rather than by an external voltage in the stator coils.

Most conventional commercial generators have a maximum angular velocity that is equal to 1500 rpm. These type of generators are available at a variety of power outputs, from several kW's up to hundreds of MW's. When these type of generators are used for power generation in applications with high speed turbines such as gas-turbines, often a gearbox is used to match the velocity of the generator.

In the past couple of decades, industry has made an effort in the design and research of high speed electrical generators (HSEG) [12] [13], generators that can be coupled to turbines for power generation without the need of an additional gearbox. The majority of these electrical generators are permanent magnet generators. While the conventional generators produce electrical power with a frequency of 50Hz, the frequency of HSEG is related to the angular velocity of the generator. In order to facilitate the electricity to grid, power electronics are required that introduce additional losses. Literature suggest that both the conventional and induction generator are able to achieve an efficiency of about 95%. Since the additional power electronics have more or less the same efficiency, the total efficiency of the high speed generator is estimated to be equal to 90% [12].

3.4.1 Electrical generator cost price

Since most of these generators are project based, it is hard to obtain/estimate the cost prices of these high speed generators. In the work of Ouwerkerk [27], the cost price of a 250kWe (30krpm) electrical generator was established from an inquiry. Electrical generator manufacturer Marathon published a catalogue with prices of conventional induction generators [25] These, and the for inflation corrected cost price of the high speed generator are presented in figure 16. From the conventional generator prices can be concluded that these are available for roughly €50 per kWe.

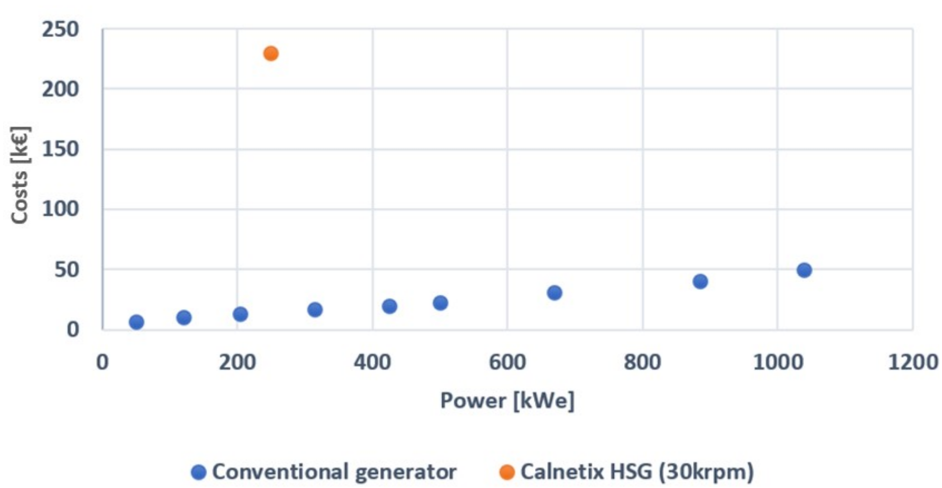


Figure 16: Cost price overview conventional generators adopted from manufacturer Marathon [25] and 250kWe (30krpm) high speed generator from Calnetix.

4 Component selection and RCG configurations

The literature study in chapter 4 has resulted in one or more options per turbo-machinery component. The technical performance/limitations and cost estimations are based on literature. However, these specifications are "not set in stone" and in designing an RCG cycle, one is dependent on the components available in the market. For each component option, the market has been looked at for available components and the associated technical specifications. Discussions with suppliers and requested quotes have led to more accurate cost estimates.

In the RCG-cycle, the steam turbine - compressor and the power turbine - generator are physically coupled. When the angular velocity of these components are not equal an additional transmission is required. Therefore these drive trains have been developed individually.

4.1 Power turbine - generator drive train

For the power turbine - generator drive train, two turbine options have been taken into account: A hot gas expansion turbine and the turbocharger turbine. Since the hot gas expansion turbine is available as a set including transmission and conventional generator, the development of this drive train does not require additional engineering. The automotive turbines require selection of a generator and transmission.

4.1.1 Hot gas expansion turbine

GE offers a serie of hot gas expansion turbine, with their smallest power output starting at 1.6MWe (Frame 20). A major benefit is that, due to the use of guided inlet vanes, an efficiency of 85% can established on their entire volumetric flow - pressure range. As can be noticed from the right hand side of figure 17, the HGAT have a maximum expandable volumtric inlet flow that varies per size. For the 1.6MWe version, the maximum expandable volumetric inlet flow is equal to 100.500 NM3/hr and the maximum turbine inlet temperature is limited to 500C.

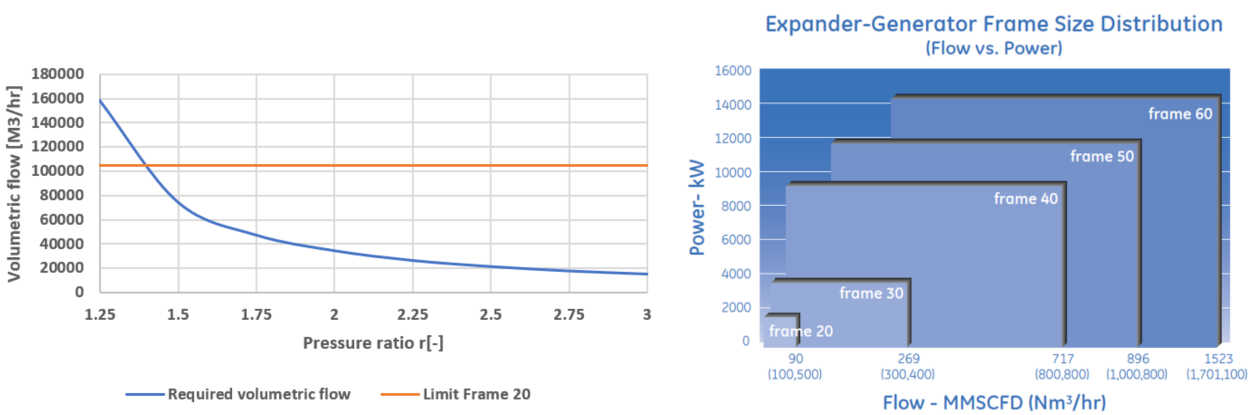


Figure 17: On the left, the maximum expandable inlet flow of the frame 20 and the calculated volumetric inlet flow for varying pressure ratio's with a assumed TIT of 500C, a turbine efficiency of 85% and a power output of 1MWe. On the right, the power output and volumetric outlet flow of the GE expander-generator combination. Figure on the right is adopted from [[11].

The required mass flow of air that the HGAT has to expand for a given power output depends on the maximum turbine inlet temperature, turbine efficiency and pressure ratio. Since the pressure and temperature dependent density of air is known, the corresponding volumetric flow can be calculated by use of the following relation:

$$\dot{V} = \frac{\dot{m}}{\rho(r, TIT_{max})} \tag{27}$$

For the maximum turbine inlet temperature and a power output of 1MWe, the volumetric air flow is calculated for varying pressure ratio's and displayed in on the left hand side of figure 17. As an addition to that, the maximum volumetric expandable flow of the Frame 20 version is displayed. From this figure can be noticed that the minimum allowable pressure ratio is between 1.25 and 1.5. Therefore is assumed that the HGAT can be operated between 1.5 and the maximum allowable pressure ratio of 12. Although an effort has been made, no enquiry was received from this supplier and hence a cost price of €650k has been assumed for the 1.6MWe HGAT.

4.1.2 Turbocharger turbine (TT)

In order to construct a turbocharger turbine - generator drive train, at first the technical performance of three different automotive turbocharger options has been compared. Thereafter, options for coupling to a generator are compared. These include direct coupling to a high speed generator and the use of a transmission and conventional generator.

An important parameter in the selection of the ATT's is the expandable mass flow, which determines the maximum power output of the turbine. As stated in chapter 3, the expandable mass flow of the turbine is related to the pressure ratio, air temperature and the angular velocity of the turbine, commonly displayed in turbine maps. A variety of ATT's have been evaluated from manufacturers Garret, Howden and Borgwarner. All evaluated turbines have similar turbine efficiencies (75%), maximum turbine inlet temperatures (1050C) and pressure ratio's (4). The major difference between the ATT's is the wheel diameter and the size of the turbine housing. A big wheel diameter is desired since this a bigger wheel diameter results into lower angular velocities which is favourable for the coupling to a generator. Additionally, a big turbine housing is desired since this increases the amount of maximum expandable air which results into a bigger maximum power output per pressure ratio. The following ATT's have been taken into consideration:

- BorgWarner K44
- Garret G3000-57
- Holset HX82

From all manufacturers, a turbine map was available that displays the corrected mass flow for varying pressure ratio's. Only the K44 turbine map contained the corresponding isentropic efficiency profiles. From the Holset HX82, only corrected mass flow data was present for pressure ratio's up to 2. Therefore a conservative assumption has been made that beyond a pressure ratio of 2, the corrected mass flow of the HX82 turbine remains constant. The map of the BorgWarner K44 and Holset HX82 are unfortunately not displayable due to copyright restrictions, however, the expandable mass flow of all turbines have been sampled and are displayed in figure 18. From this figure can be noticed that the HX82 turbine is able to expand a higher mass flow of air per pressure ratio and hence, this turbine is selected for the turbocharger drive train.

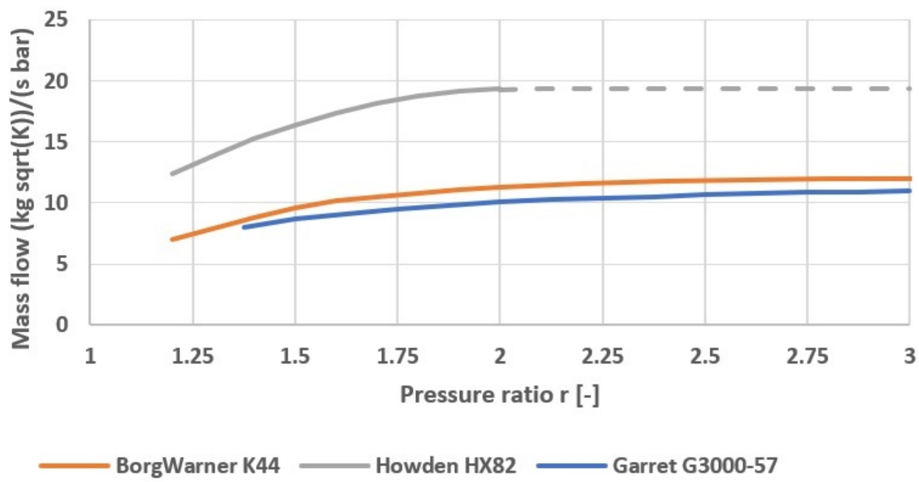


Figure 18: Corrected mass flow of the BorgWarner K44, Howden HX82 and Garret G3000-57 turbine for varying pressure ratio's.

The turbine efficiency of the HX82 turbine is related to the turbines angular velocity and pressure ratio. Since the angular velocity profile of the HX82 turbine is not known, a turbine efficiency estimation method has been used, in more detail described in Appendix A. The efficiency of the HX82 turbine has been computed for angular velocities of 30, 50 and 70krpm, which corresponds to the angular velocities of the evaluated transmission and high speed generator options. The turbine efficiency for varying pressure ratio's and angular velocities are displayed in figure 19.

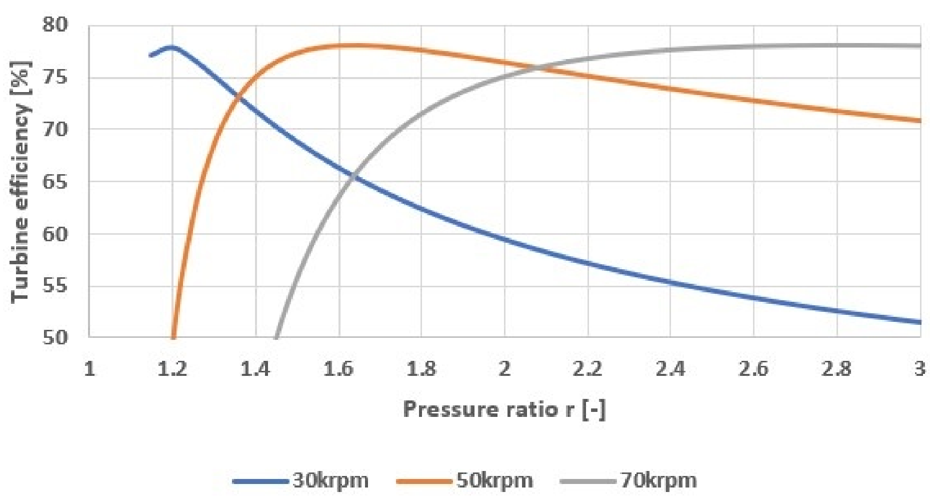


Figure 19: Turbine efficiency of the HX82 turbine for varying pressure ratio's at 30, 50 and 70krpm.

During the selection of the transmission and high speed generator options, it is desired to have an efficiency of at least 75%. From figure 19 can be noticed that for an angular velocity of 30krpm the pressure ratio is limited to 1.3, at 50krpm the pressure ratio can vary between 1.4 and 2.2 and for an angular velocity of 70krpm, the pressure ratio should be higher than 2.

Direct coupling to high speed generator (HSG)

From the work of Ouwerkerk [26] and a quotation from 2017, the cost price of two HSG options has been established; a 175kWe/50krpm and a 250kWe/30krpm version. Since it is desired to ensure that the turbine efficiency is equal to 75%, the maximum pressure ratio for both HSG options is limited. In chapter 2 was stated by equation 6 was that the maximum power output of the turbine depends on the pressure ratio, expandable mass flow and turbine inlet temperature:

$$P_t = \dot{m} \cdot n_t \cdot C_p \cdot TIT \cdot [1 - (\frac{1}{r})^{(\frac{k-1}{k})}] \tag{28}$$

From this equation can be noticed that the maximum power output depends on the (temperature and pressure depended) expansable mass flow and the used turbine inlet temperature. For a turbine inlet temperature of 500C and 1000C, which is considered to be the range of applicable temperatures of the RCG's heat source, the electrical power output and the cost price of the combined HX82 turbine and high speed generator have been calculated, these are listed in table 3.

From this table can be noticed that the 250kWe HSG is a poor choice, since it is only able to reach a fraction of the HSG's maximum power output, while the cost price of the generator is significantly higher than the 175kWe version. When the 175kWe HSG option is implemetend, this would result into an investment cost of €800k since 6 turbines are required to generate an electrical power output of 1MWe.

		TIT = 500C		TIT = 1000C		
Option	ω [krpm]	r [-]	P_{gen}	r [-]	P_{gen}	Costs [k€]
Calnetix 175kWe	50	2.2	168	2	175	132
Calnetix 250kWe	30	1.3	25	1.3	33	268

Table 3: Calnetix HSG and coupling to Holset HX82 turbine. Cost prices include turbine costs of €10k.

Transmission and conventional generator

The most cost effective transmission option of the turbine and conventional generator is expected to be belt/train transmission. Unfortunately, belt transmission from Megadyne and continental for higher power outputs (>100kW) are limited to velocities up to 10krpm. Due to this low angular velocity, the HX82 turbine is able to achieve a maximum turbine efficiency of 45% at a pressure ratio of 1.25. As a result, more than 100 turbines would be required when the TIT is equal to 500C while for a TIT of 1000C, 80 turbines are necessary to generate a power output of 1MWe. Considered that a single induction generator is used, this would result into a challenging transmission and a cost price varying between €863k and €1.1M. When all turbines would be mounted to a single generator, this introduces an additional cost of €3k per turbine. Eventually, this would lead to a total cost price of €1M to €1.3M.

More promising is the use of a gearbox. Thereby, two gearboxes have been considered; a planetary gearbox and a helical gearbox. The helical gearbox has a transmission ratio that is limited to 6.5. Gearbox manufacturer Vortron has off-the-shelf helical gearboxes that are equipped with two transmission axis. Considered that the angular velocity of the generator is equal to 1500rpm, this leads to a maximum input velocity of 30krpm. These gearboxes are available for different power outputs with the smallest version starting at 250kW. The cost price of this option has been established from an inquiry and is listed in table 4.

As an addition to that, a 4-1 planetary gearbox from the work of Ouwerkerk [27] has been proposed. A schematic of the planetary gearbox is displayed in figure 20. On this gearbox, four turbines can be connected that can deliver a maximum power output of 250kW each, which leads to a total maximum output power of 1MW. The input velocity of the gearbox is limited to 70krpm considered that a 1500rpm conventional induction generator is connected. The cost price of this option is corrected for current prices and is listed in table 4.

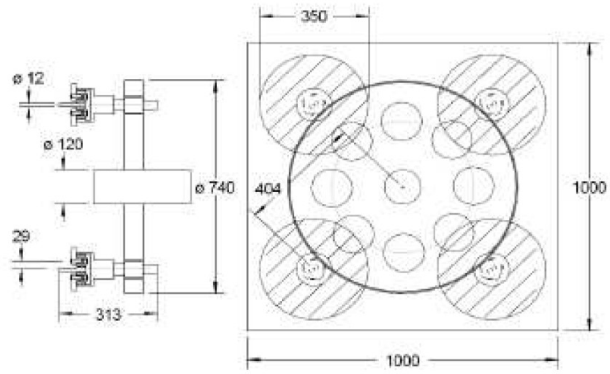


Figure 20: Schematic of elliptical 4-1 gearbox, adopted from [26].

Since the maximum power output depends on the turbine inlet temperature and pressure ratio, the maximum electrical power output of the turbine, transmission and generator combination has been computed by the same methodology as has been done for the high speed generator options. The electrical power output for a TIT of 500C and 1000C are displayed in table 4. Thereby is assumed that the generators are able to achieve an efficiency of 95%.

		TIT = 500C		TIT = 1000C		
Option	ω [krpm]	r [-]	P_{gen}	r [-]	P_{gen}	Costs [k€]
4-1 planetary GB	70	2.6	248	2.25	250	230
Helical GB	30	1.3	27	1.3	35	112

Table 4: Planetary (4-1) and helical gearbox combinations with HX82 turbine for TIT of 500C and 1000C. Cost prices estimations include turbine and induction generator.

From the evaluation of the gearboxes can be concluded that the 4-1 planetary gearbox is most promising since it is able to achieve to highest power output and is favourable in terms of cost price. Since the initial cost price of this type of gearbox is lower than the HSG options, this combination has been proposed as a turbine - generator drive train for the automotive turbocharger turbines.

4.1.3 Power turbine - generator drive train comparison

Eventually, this leads to two possible configurations for the power turbine - generator drive train. A schematic of the possible configuration is displayed in figure 21. On the left, the figure displays the hot gas expansion turbine drive train, while on the right, the turbocharger turbine variant is shown.

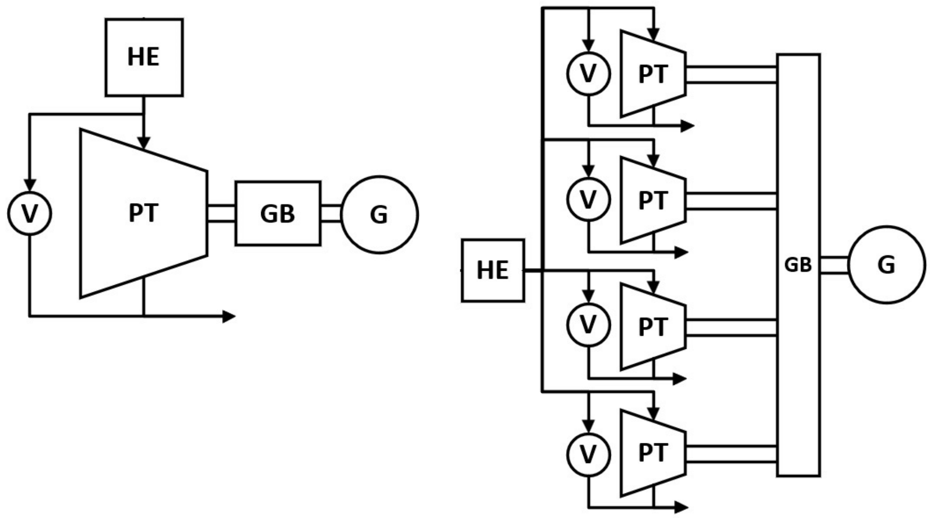


Figure 21: Schematic of the hot gas expansion turbine (HGAT) on the left and automotive turbocharger turbine (ATT) drive train configuration on the right.

An overview of the main technical characteristics are displayed in table 5. Main differences between both configurations are the turbine efficiency, initial cost price and turbine's lifetime. Since the efficiency of the HGAT drive train is higher, this means that the turbine requires a lower mass flow of air for the same power output. Due to this, the use of the HGAT can result into lower investment costs of the compressor and steam turbine. Although the HGAT requires higher initial investment costs, their lifetime is tremendously higher than the turbocharger variant.

Drive train	$TIT_{max}[C]$	r [-]	nT [%]	Cost	Lifetime
Hot air expander including helical gearbox and induction generator	500	12	85	650	>35
Turbocharger turbine including 4-1 planetary gearbox and induction generator	1050	4	75	220	3 (turb)

Table 5: Investment costs and main characteristics of the hot gas expansion turbine and turbo charger turbine drive train.

4.2 Steam turbine - compressor drive train

For the steam turbine - compressor drive train two compressors have been taken into consideration; the turbo-charger compressor and the single stage centrifugal compressor.

4.2.1 Steam turbine

Howden, a manufacturer of impulse back-pressure steam turbines offers three different turbines with power outputs below 1MW. Howden estimated the cost price of their turbines at around €1k per kW. These cost prices are more accurate than the obtained cost from the literature study, however these costs included an additional generator, oil circuit and control circuit. Therefore the estimations from the literature study have been used. The steam turbine can handle steam temperatures up to 400C and pressures up to 40 barg. They are available with angular velocities between 0.5 to 23krpm and have an estimated isentropic efficiency of 70%.

Type	P [kW]	Cost price [k€]
SST-040	300	235
SST-050	750	400
SST-060	1000	420

Table 6: Steam turbine options from Howden with power outputs and estimated cost price

4.2.2 Turbocharger compressor

The turbo-charger compressors (TC) from manufacturer Garret are an appealing option as these are characterized by their low initial investment costs (up to €10k) and good availability. Due to their fixed rotor vanes, the volumetric airflow and efficiency relate to the compressors pressure ratio and angular velocity. These relations are displayed in compressor maps, that differ for varying inducer diameters. The compressor maps of the biggest compressor from Garret, the GTX5544r-102mm is displayed in figure 22.

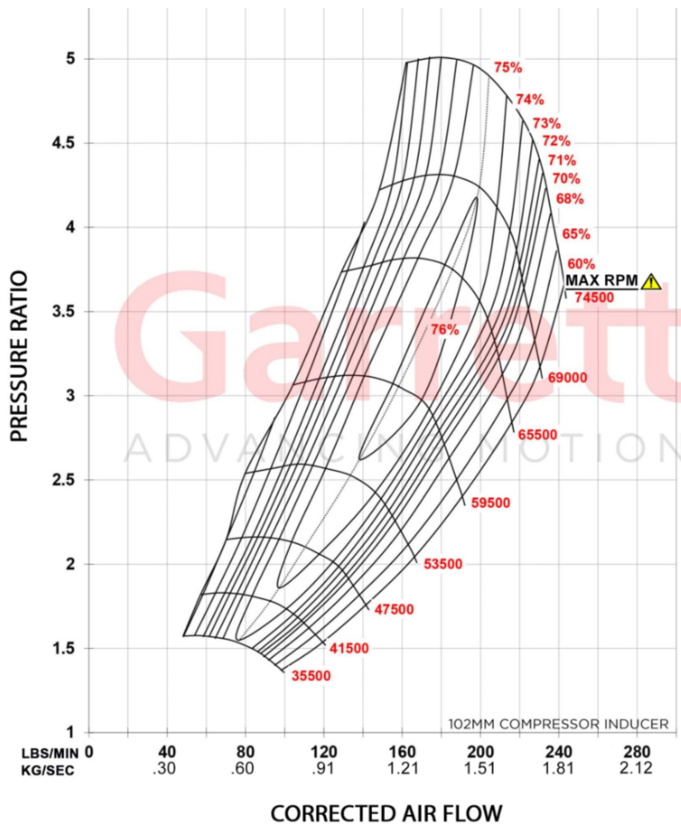


Figure 22: GTX5544r compressor map with 102mm impeller diameter. Figure has been adopted from [10].

The angular velocity of these compressors are much higher than the angular velocity of the steam turbine and hence an additional transmission is required. Previously was concluded that the 4-1 planetary gearbox was the most cost effective solution for the power turbines and the same holds for compressor. As can be noticed from figure 22, the compressor can be operated between 35 to 70krpm. It is assumed that, since the gearbox was available at 70krpm, a version up to 35krpm is available for the same cost price.

During the selection of the compressor, it is desired that stay at the right hand side of the dotted line in the compressor map. When the compressor is operated in the region left from the dotted line, it can be operated near the surge region when a sudden pressure drop occurs. This can cause heavy vibrations and lead to permanent damage.

4.2.3 Single stage centrifugal compressor (SSCC)

Howden has a serie of SSCC's that differ in volumetric inlet flows. These compressors are executed with variable vanes which guarantees an efficiency of 85% on the entire pressure and volumetric flow range. They are executed with an internal gearbox and normally driven by an electrical motor. Since the input velocity is equal to 1500rpm, this compressor can be directly driven by the steam turbine. The compressors have a maximum pressure ratio of 3 and are available with mass flows between 0.27 and 16 kg/s. The cost price of these compressors, estimated by Howden, vary for different sizes. The cost price of the single stage centrifugal compressor and the maximum mass flow are displayed in table 7.

Type	mass flow [kg/s]	Price [k€]
KA2	1.06	100
KA5	1.94	150
KA10	3.33	200
KA22	6.11	250
KA44	10.56	300
KA66	16.11	400

Table 7: Howden HV series compressors. Cost price has been estimated by Howden and mass flows have been obtained from HV-turbo compressor datasheet [16].

4.2.4 Steam turbine compressor drive train comparison

Eventually, this leads to two possible configurations for the steam turbine - compressor drive train. A schematic of the possible configurations are displayed in figure 23. On the left, the figure displays the planetary gearbox option with turbocharger compressor, while on the right, the single stage centrifugal compressor variant is shown. The main technical and economical characteristics of both drive trains is listed in table 8.

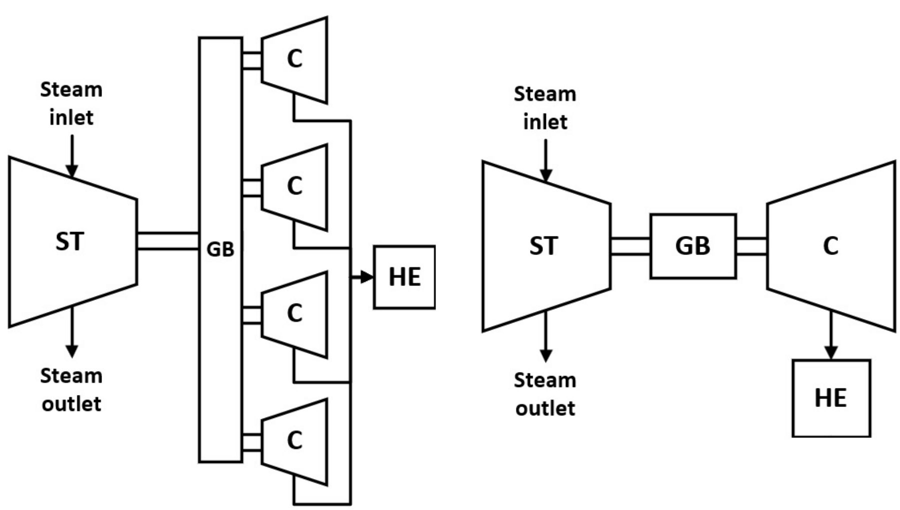


Figure 23: Schematic of the turbocharger compressor with planetary gearbox drive train on the left and single stage centrifugal compressor drive train configuration is shown on the right

From figure 22 can be noticed that the maximum mass flow of the turbocharger compressor varies for different pressure ratio's and compressor efficiencies. When 4 turbocharger compressors are used at a pressure ratio of 3, this compressor drive train can achieve a mass flow between 5-7 kg/s with efficiencies varying between 60-75% while the total cost for this configuration is €220k. On the other hand, a KA22 and KA44 compressor can achieve mass flows of 6.11 and 10.56 kg/s for the same pressure ratio. The cost of this solution is respectively €250k and €300k, while it is able to obtain a compressor efficiency of 85%. Thereby can be noticed that the lifetime of the turbocharger compressor is limited to around 3 years, while the single stage centrifugal compressor is guaranteed to function for more than 25 years. Hence, only the single stage centrifugal compressor is taken into consideration for the development of the RCG cycles.

Drive train	m [kg/s]	$r_{max}[-]$	nC [%]	Cost [k€]	Lifetime [y]
Impulse steam turbine and single stage centrifugal compressor	1 - 16	3	85	100 - 400	>25
Impulse steam turbine with 4-1 planetary gearbox and turbocharger compressors	7	4.5	60 - 75	220	3 (comp)

Table 8: Main technical and economical characteristics of the hot gas expansion turbine and turbo charger turbine drive train.

4.3 RCG options

Since the turbocharger compressor option is not taken into consideration, the evaluated drive train options have led to two different RCG configurations:

- **RCG configuration 1:** Single stage centrifugal compressor with four turbocharger turbines.
- **RCG configuration 2:** Single stage centrifugal compressor with hot gas expansion turbine.

Both configurations are displayed in figure 24 With RCG configuration 1 on the left, and RCG configuration 2 on the right.

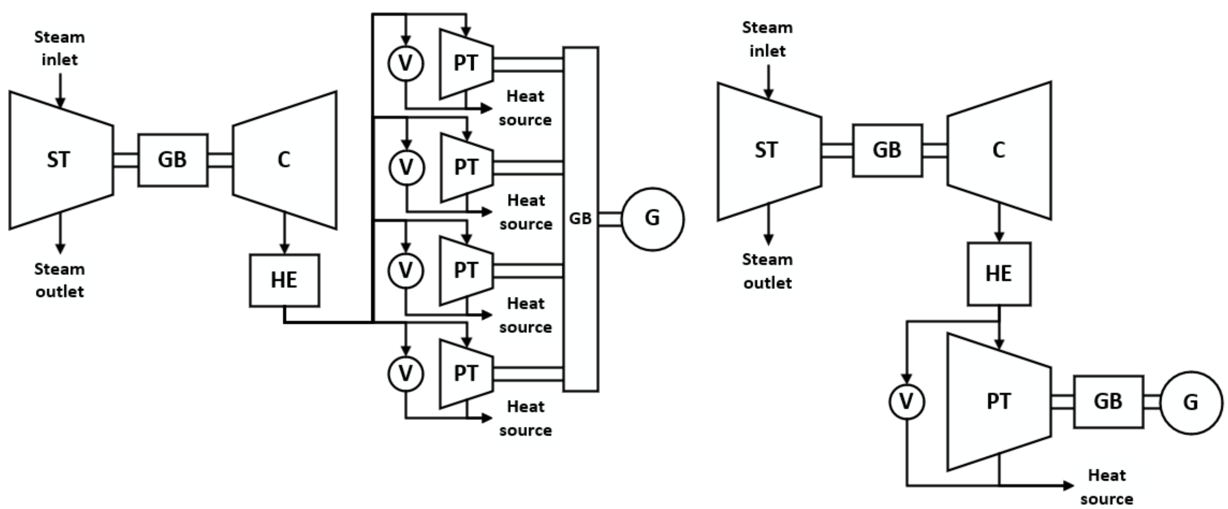


Figure 24: Both configurations containing an impulse steam turbine and a single stage centrifugal compressor. RCG configuration 1 on the left contains a 4-1 planetary gearbox, four turbocharger turbines and a conventional generator. RCG configuration 2, on the right, is eqquiped with a hot gas expansion turbine including planetary gearbox and conventional generator.

The hot gas expansion turbine of RCG configuration 1 has a maximum turbine inlet temperature of 500C. Due to it's variable inlet vanes, the pressure ratio can be varied from 1.5 up to 3 bar, as this is the maximum pressure ratio the single stage centrifugal compressor can deliver. The pressure ratio of the turbocharger turbine that is used in RCG configuration 2 is pressure depended. However, the turbine inlet temperature of the configuration can be altered between the lowest considered temperature of the heat source, 500C, and the maximum allowable turbine inlet temperature of 1000C. Eventually, this leads to 4 different RCG options that will be considered in this report. Their technical characteristics are listed in table 9.

Option	Configuration	TIT [C]	r [-]
A	1	500	2.62
B	1	1000	2.28
C	2	500	1.5
D	2	500	3

Table 9: RCG option A till D, based configuration 1 and 2 with varying turbine inlet tempertures and pressure ratio's.

The thermodynamic equations from chapter 2 have been translated to a thermodynamic model. For all configurations, the properties of the thermodynamic states and the power and heat of the air and steam cycles have been calculated. These are listed in appendix B.

4.4 Component selection

The selection of power turbine - generator options are clear for options A till D. What is left is the selection of the compressor and steam turbine. The selection of the compressor depends on the required mass flow of air that on their turn depends on the operational conditions of power turbine. The selection of the steam turbine depends on the required compressor power, which depends on the required mass flow of air and the pressure ratio of the power turbine.

4.4.1 Compressor selection RCG configuration 1: option A and B

From figure 18 was concluded that the expandable mass low of the HX82 turbine is temperature and pressure dependent, which also holds for the maximum power output of the turbine. Since these relations are known, the output power of the turbine can be written as a function of the turbine inlet temperature and pressure ratio. For a configuration with four HX82 turbines, the relation between the turbine inlet temperature and pressure ratio is given for a power output of 1MWe, displayed in figure 25.

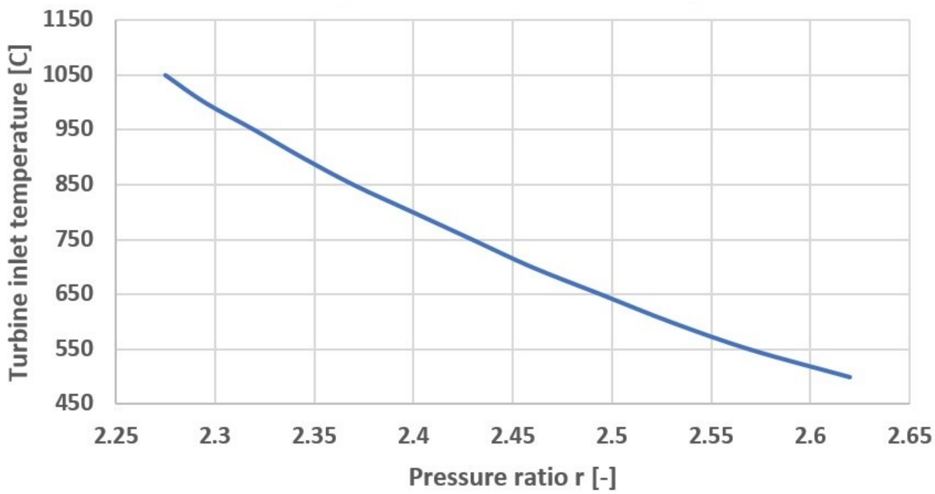


Figure 25: Pressure ratio as a function of the turbine inlet temperature for a configuration with 4x HX82 turbines that deliver an electrical power output of 1MWe. For the power output, a generator efficiency of 95% has been assumed.

The required mass flow of air that has to be compressed by the compressor is related to the properties of the power turbine. It can be calculated by re-writing equation 6 . These included the turbine power (P_{pt}), turbine efficiency (n_{pt}), specific heat (C_p), turbine inlet temperature (TIT), pressure ratio (r) and specific heat ratio (k). These are displayed in equation 29.

$$\dot{m} = \frac{P_{pt}}{n_{pt} \cdot C_p \cdot TIT \cdot (1 - r^{\frac{k-1}{k}})} \tag{29}$$

Since the pressure ratio for the configuration is known for different inlet temperatures, the mass flow of the turbine can be written as a function of the turbine inlet temperature. For an electrical power output of 1MWe, the required mass flow of air is displayed for varying turbine inlet temperatures in figure 26. The KA22 single stage centrifugal compressor has a maximum mass flow of 6.11 kg/s and the KA44 compressor has a maximum mass flow of 10.56 kg/s. This means that for turbine inlet temperatures above 650C, the slightly cheaper KA22 compressor can be used, while turbine inlet temperatures below 650C require the use of the KA44 compressor.

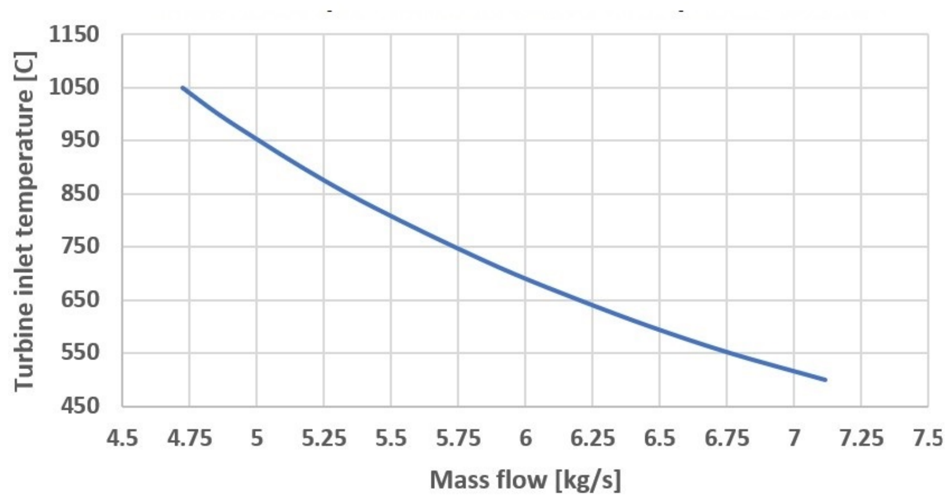


Figure 26: Mass flow of the HX82 turbine as a function of the turbine inlet temperature.

4.4.2 Compressor selection RCG options C and D

For a power output of 1MWe, the required mass flow for varying pressure ratio's of the HGAT is displayed in figure 27. For a pressure ratio below 2.5, the slightly smaller KA22 compressor can be selected, while for a pressure ratio above 2.5, the more expensive KA44 compressor is required.

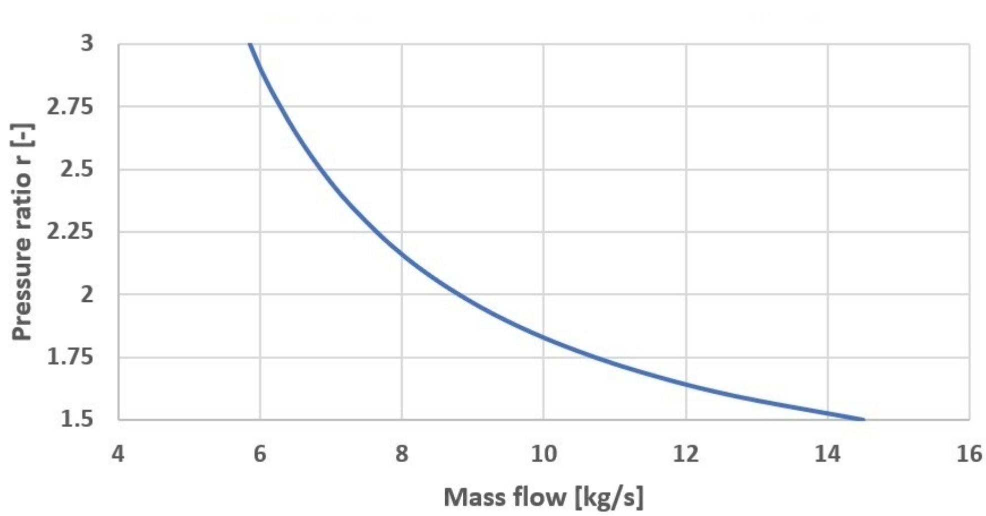


Figure 27: Mass flow for varying pressure ratio's of the HGAT used in RCG options C and D.

The components that have been used in all four configurations are displayed in table 10. The main difference between the RCG options can be assigned to the used compressor and steam turbine. In option A and C, the required mass flow of air exceeds the limitation of compressor KA22 and hence a slightly bigger compressor is required. Since the compressor power of option A exceeds 750kW, a bigger steam turbine, the ST-060 is required.

Option	A	B	C	D
Generator	ABB 1MWe IG		GE 1.6MWe GE HGAT	
Transmission	4-1 planetary GB			
Power turbine	4x HX82			
Compressor	KA44	KA22	KA44	KA22
Steam turbine	SST-060	SST-050	SST-050	SST-050

Table 10: Components that are used in RCG option A till D.

4.5 Costs

A break down of the equipment costs of the turbo-machinery components of all four RCG options is displayed in figure 28. The equipment costs of the steam turbine and compressor are comparable for all options. Major differences in equipment costs can be assigned to the used power turbine, gearbox and generator. For options A and B, the use of turbocharger turbines and a planetary gearbox lead to an estimated equipment costs of €220k, while the use of a hot gas expansion turbine (HGAT) in option C and D lead to an estimated cost price of €650k.

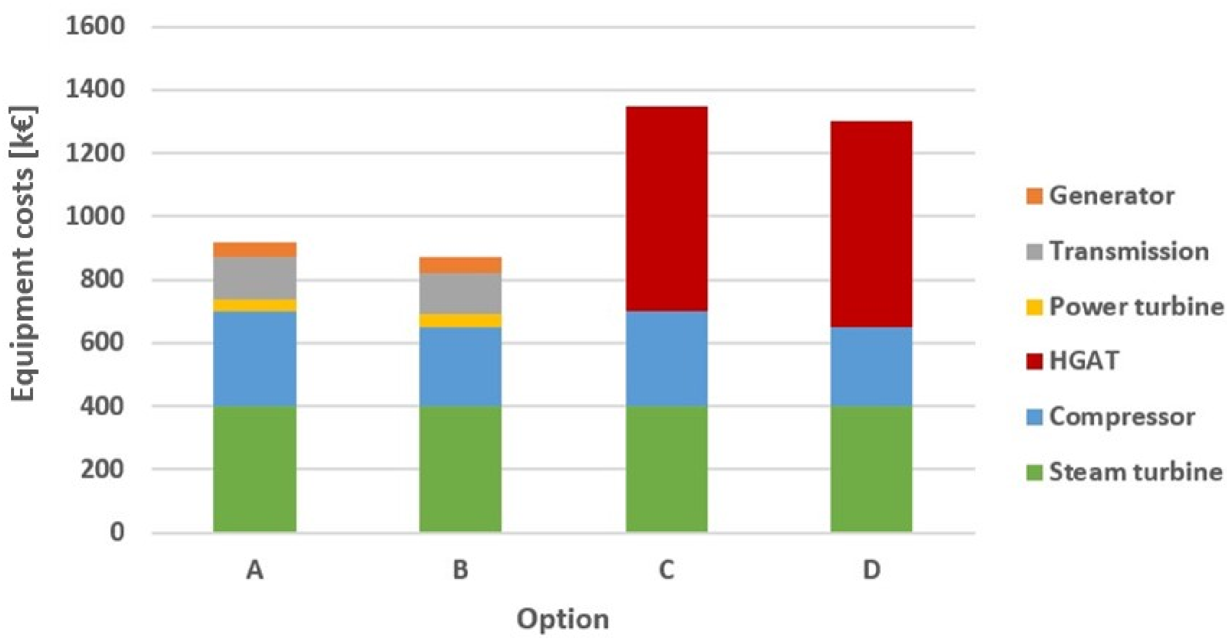


Figure 28: Equipment costs of all four RCG configurations. In this overview, HGAT is the hot gas expansion turbine.

Research on cost price estimations of components in chemical plant design suggests that the cost price of components that have been estimated by cost price estimation programs can introduce uncertainties up to +/- 30%, while costs that have been established from enquires can have errors up to +/-10% [34]. Costs that have been established from publicly available cost prices can considered to be true costs. For small scale combined heat and power systems, the total installed costs are estimated to be 50-70% of the equipment costs [20]. Since the RCG-cycles make more or less use of the same type of turbo-machinery components, this assumption has been used for the calculation of the installed cost prices. The 50-70% is based on the costs for the equipment, sensors, control system, coupling, oil system, piping and valves. These total installed costs do not include the costs for the boiler and air heat exchanger, which can vary per application.

For the RCG options A till D and for a combined steam turbine and generator plant (ST) plant, the estimated installed costs are displayed in figure 29. Since the cost price of the equipment and the translation between the equipment cost to installed costs are subjected to uncertainties, a range of potential installed investment costs is displayed. Thereby should be noticed that the development of a first industrial RCG prototype introduces risks and unforeseen problems. Therefore it is expected that these estimated costs are only valid for production in series.

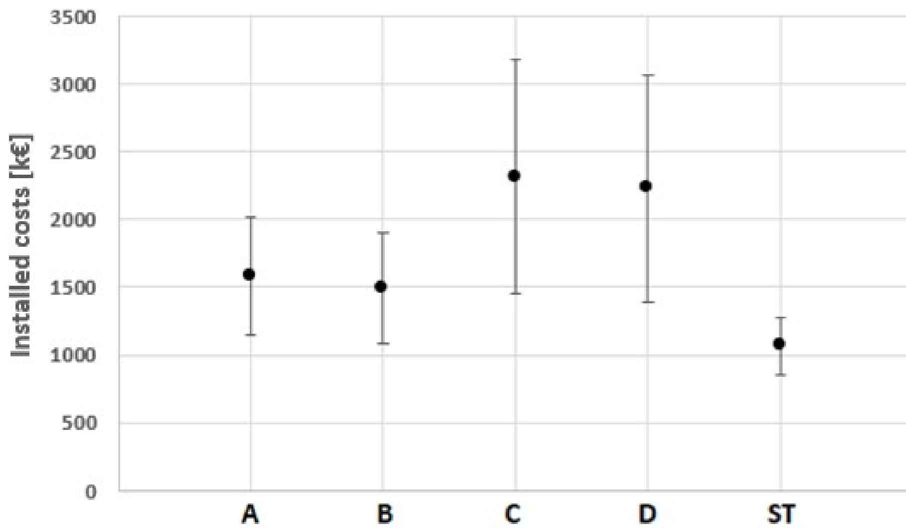


Figure 29: Installed costs of RCG option A till D and the combined steam turbine and generator (ST). The cost price of the ST has been adopted from [20]

Besides the installed costs, the operation and maintenance costs (O&M) play an important role in the economic assessment of the RCG cycle. Literature suggests that the operation and maintenance costs (O&M) of an impulse steam turbine, gearbox and generator below 1MWe are equal to around €10/kWe per year [20]. These cost include for example change of the oil filters, bearings and regular inspection. Few is known about the operational costs of the single stage centrifugal compressor, but since these also contains an oil circuit, gearbox and bearings, it is assumed that O&M costs of the single stage centrifugal compressor are in the same order of magnitude. The steam turbine and compressor have therefore an OM costs of €20k per year and the total O&M costs for the steam turbine - generator is considered to be €10k per year.

The O&M costs of the hot gas expansion turbine are assumed to be 2% of the equipment costs per year, as is suggested by an economic feasibility study of a 1.8MW turbo-expander [8] This leads to a yearly O&M cost of around €13k per year. The majority of turbo-machinery components (steam turbine, compressor, transmission, generator) have guaranteed lifetimes up to or above 35 years. Garret, the manufacturer of the turbo-charger turbines guarantees a lifetime of their components up to 3 years. This means in essence that each turbine requires replacement after 3 years. Since each turbine costs €10k, option A and B have an additional maintenance cost of €13.3k per year which is comparable to the O&M costs of the turbo expander. Therefore the expander and generator of all options introduce an O&M costs of around €20k per year.

A summary of the mean equipment costs, the mean installed costs and the O&M costs for all four RCG options and the sole steam turbine - generator (ST) are listed in table 11.

	A	B	C	D	ST	Unit
Equipment cost	920	870	1350	1300	668	k$
Installed cost	1577.8	1492.05	2315.25	2229.5	1002	k$
Uncertainty	46	47	61	62	6	%
O&M costs	40	40	40	40	10	k$

Table 11: Equipment costs, installed equipment costs and operation and maintenance costs of RCG option A till D and ST-cycle. Costs are valid for a 1MWe configuration.

4.6 Heat

The RCG-cycle makes use of two different heat exchangers and hence has two locations where heat is consumed. The first heat exchanger is the boiler, where heat is generated to drive the steam turbine. Secondly, an air heat exchanger is used to heat the ambient compressed air prior to expansion over the power turbine. At the power turbine outlet, the remaining heat can either be re-used by guiding it directly into a furnace or an additional heat exchanger. Alternatively, it can be fed back into the atmosphere.

In this research, distinction is made between two cycles:

1. When the RCG is considered as an **add-on** to an excisting furnace and boiler system, it is executed as an open-loop steam cycle where the steam at the outlet of the steam turbine is consumed by an industrial plant (industrial wood dryers, beer brewery's, etc).
2. When the RCG is executed as a **stand-alone** system, it is assumed to be a closed-loop (rankine) steam cycle where the steam at the outlet of the steam turbine is condensed.

The main difference between both cycles are the conditions at the inlet of the pump and at the outlet of the steam turbine. Thereby the following assumptions have been made:

- In the closed loop cycle, the steam is fully condensed in a condenser before it enters the pump. The temperature of the water at this state is (close to) 100°C. In the open-loop cycle the temperature is considered to be equal to 20 °C.
- The temperature of the super-heated steam in both cycles is equal to 400 °C since this is the maximum inlet temperatures for the proposed steam turbines. The pressure prior to expansion is assumed to be 29 bara.
- In the open-loop configuration it is assumed that the steam at the outlet of the steam turbine is consumed at an industrial cite. Commonly, this low pressure steam is dry with a pressure of 1.5 bara and is consumed until the steam is fully condensed.
- In the closed loop configuration, the steam at the outlet of the steam turbine is condensed. The vapor fraction at the outlet of the steam turbine is equal to 0.9. When lower vapor fractions are used, this can lead to droplets during expansion which on their turn can lead to a reduce lifespan of the turbine blades [2].

The operational conditions of both steam cycles are displayed in appendix B and have been obtained by use of the thermodynamic equations from chapter 2. In figure 30 a distribution of the heat consumption and generation is displayed for the stand-alone RCG (closed loop cycle) and the RCG add-on (Open loop cycle). The heat distribution contains the amount of heat that is generated in the boiler (Q_B), the amount of heat that is consumed by the steam turbine (Q_{ST}) and the amount of heat that is left at steam turbine outlet (Q_{STO}). These have been obtained by use of equations 11 till 25.

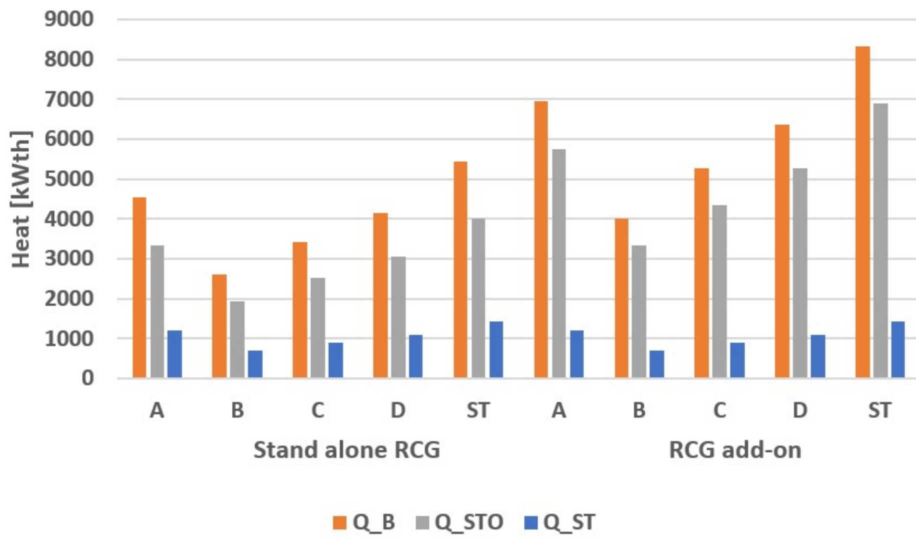


Figure 30: Heat distribution of the steam cycle for the RCG cycle configured as stand-alone and add-on to an existing boiler and furnace system. In the stand-alone execution, the steam cycle is a closed loop rankine cycle. When the RCG is configured as an add-on, the steam cycle is executed as an open-loop steam cycle. This overview contains the amount of heat that is required from the boiler (Q_B), the amount of heat that is consumed by the steam turbine (Q_{ST}) and the amount of heat that is left at the outlet of the steam turbine (Q_{STO}) for a power output of 1MWe for RCG option A till D and the sole steam turbine and generator (ST). The properties of both steam cycles are displayed in appendix B.

RCG options A till D and the sole steam turbine and generator differ in the amount of heat that is consumed by the steam turbine. This is can be assigned to the fact that the RCG options and the ST vary in the required steam turbine power. Since the amount of power is independent from the steam cycle configuration, these are the same in both stand-alone (Closed loop) and add-on (open-loop) configuration. The main difference can be assigned to the amount of heat that is required by the boiler and the amount of heat that is present at the outlet of the steam turbine. Since the conditions of the steam for options A till D and the ST are equal per individual steam cycle, the ratio between the amount of required boiler heat, the heat that is consumed by the steam turbine and the heat that is present at the outlet of the steam turbine is equal.

In figure 31 the heat that is required by the air heat exchanger and the amount of heat that is present at the outlet of the power turbine is displayed for RCG configuration A till D. The amount of heat that is used by the air heat exchanger depends on the outlet temperature of the compressor, the mass flow of air and the turbine inlet temperature. At RCG option C, the turbine inlet temperature is 1000C, while at the other configurations a turbine inlet temperature of 500C is assumed. This, in combination with the relatively high mass flow (due to a low power turbine pressure ratio) leads to a relatively high heat consumption at the air cycle. In option D, the amount of heat that is consumed by the air heat exchanger is relatively low, since a low mass flow of air is required at a low turbine inlet temperature. In RCG option A and B, a less efficient power turbine is used. This leads to relatively more heat at the outlet of the power turbine w.r.t. the amount of heat that is consumed by the air heat exchanger than in RCG option C and D.

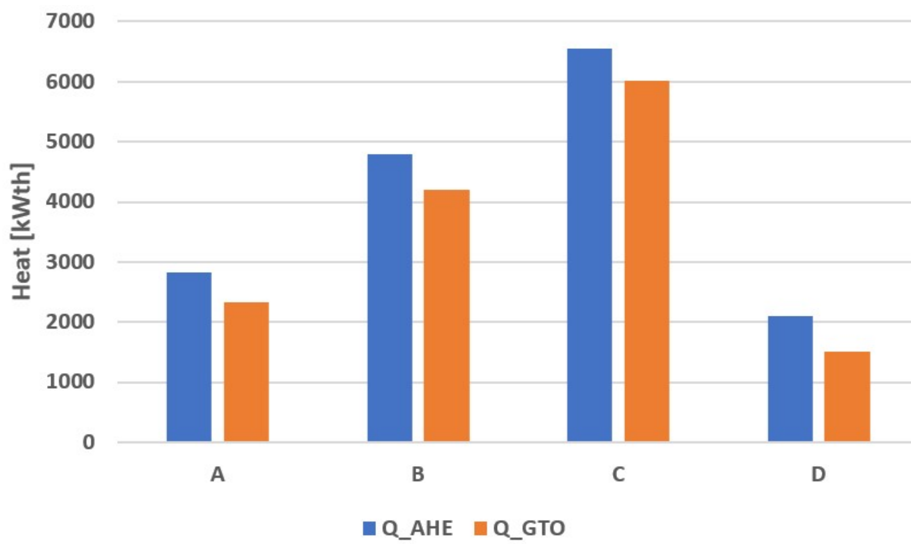


Figure 31: overview of heat that is consumed by the air heat exchanger and the amount of heat that is present at the outlet of the power turbine for RCG options A till D.

The amount of air at the outlet of the power turbine can either be re-used when it is fed back to the heat source, or fed into the atmosphere. The required heat for RCG options A till D and the sole steam turbine and generator (ST) is displayed in figure 32. Since the amount of consumed heat per option is known and the power output of the cycles are equal, the heat to electricity of the cycles can be calculated. For each cycle, the heat to electricity efficiency is listed in table 12. It contains the heat to electricity efficiency (n_{H2E}) for when the heat at the outlet of the power turbine is either re-used or not re-used. Thereby should be noticed that only the consumed heat from the steam turbine is taken into consideration.

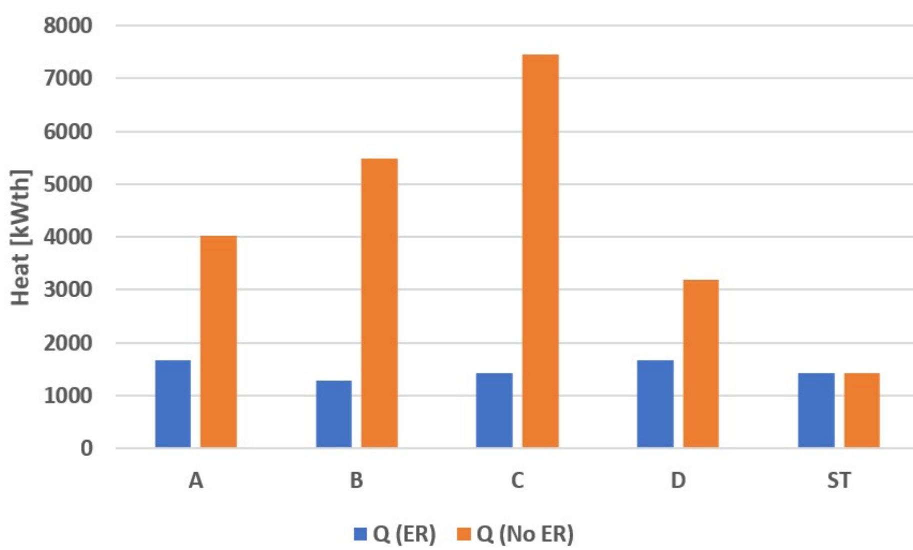


Figure 32: Amount of heat that is required for RCG option A till D and the sole steam turbine and generator (ST) for the generation of 1MWe electrical power output. The required heat is given for when the heat at the outlet of the power turbine is recovered (ER) or fed back into the atmosphere (No ER).

The heat to electricity efficiency has been calculated by use of equation 26 and are displayed in table 12. From these can be concluded that RCG option B and C can achieve heat to electricity efficiencies that are higher than the sole steam turbine and generator when the heat at the outlet of the power turbine is recovered.

Option	(n_{ER})	(n_{NoER})
A	59.3%	24.7%
B	77.7%	18.2%
C	69.6%	13.2%
D	59.2%	31.2%
ST	69.4%	69.5%

Table 12: Heat to electricity efficiency for RCG options A till D and the sole steam turbine and generator (ST). The heat to electricity efficiency is obtained for configurations where the heat is recovered (ER) and not recovered (No ER)

4.7 Technical risks

Like any process installation, the proposed RCG options and the ST cycle are subjected to technical risks. In essence, no installation is completely safe. The technical risks of an installation are commonly assessed by use of a hazard and operability (HAZOP) study. In this study, the causes and affects of potential hazardous events are listed and sufficient safety measures are taken into account. These can include for example over-speed protection, break protection on an axis and pressure safety valves. This is not within the scope of this research and for all installations it is assumed that the there are sufficient safeguards available to guarantee an acceptable risk for the environment and employees.

The technical risks that are assessed in this report are the risk of a non-functioning installation due to unforeseen technical challenges. These typically take place at hardware -and software level when individual components are integrated into a system. It is common practice to identify these risk by use of technology readiness levels (TRL), a scale that is introduced by NASA to measure the maturity of an technique. These scales are constructed such that the maturity of the majority of hardware -and software projects can be measured. The downside of this approach is that the descriptions per TRL are quite vague and leave lots of room for interpretation and subjective choices. Luckely, these TRL's have also been developed for the chemical industry [5], an industry which has lots of similarities with the power generation industry. Therefore, the technical risks of the RCG options and ST-cycle has been measured according to these scales. The description of the TRL's are listed in appendix C.

Between the RCG options, distinction can be made in how the cycle is executed. In stand-alone mode, the RCG cycle makes use of an additional (fast responding) boiler system. When the cycle is performed as an add-on, the cycle makes use of steam that is consumed for the industrial installation where the RCG has been installed.

The concept of the RCG-cycle, as a stand-alone version, has been validated in a laboratory environment and measurements have been carried out on the transient thermodynamic behaviour of the plant. According to the chemical TRL's presented in appendix C, this would indicate a TRL of 5 when the cycle is executed as a stand-alone version. At an industrial environment, a small demo version (5kWe) of an RCG add-on has been constructed. This demo plant is operated with power outputs that are significantly lower than a commercial RCG cycle. From the component overview of chapter 3 can be concluded that by use of the same type of turbo-machinery components, an economical viable installation can be made. Therefore is concluded that the RCG add-on is currently at TRL-7. In order to obtain TRL-8, the working principle of the cycle should be demonstrated in an industrial environment at full scale. Although the fast responding boiler system has been demonstrated at the capacity that is required to power the ST-cycle, the combined system has not been demonstrated in an industrial environment. Therefore the ST-cycle as a stand-alone version has a TRL-8. Since ST-cycle systems are widely available as a "turn-key" solution, the stand-alone ST-cycle is scored at TRL-9.

One could argue in what sense the proposed RCG-options A till D correspond to the prototypes that are used in the laboratory environment (TRL-4) and at the industrial site (TRL-7). The previous industrial prototypes make use of the same impulse steam turbine as the proposed RCG options. However, in the proposed RCG options, a different compressor and power turbine is used. In option A-B, the same type of power turbine is used, however, the power train includes an additional high speed gearbox. In option C-D, a hot gas expansion turbine is used which is a different type of turbine. The risks that are introduced by these changes are hard to estimate. Since these concepts are not validated in practice, the chemical TRL scale of Appendix D suggests a TRL of 3 for all proposed RCG options, both in stand-alone and add-on configuration. Both TRL's, where the RCG is presented as a stand-alone version and add-on are listed in table 13.

Option	Stand-alone	Add-on
A	3-5	3-7
B	3-5	3-7
C	3-5	3-7
D	3-5	3-7
ST	8	9

Table 13: Technology readiness level (TRL) per RCG option and the ST-cycle for stand-alone and add-on configuration.

4.8 Summary of main characteristics

In total, four different RCG options have been developed. Their technical and economical characteristics are displayed in table 14.

Categorie	Criteria	A	B	C	D	ST	Unit
Economical	Installed costs	1580	1490	2215	2230	1000	k€
	Uncertainty installed costs	46	47	61	62	6	%
	O&M costs	40	40	40	40	10	k€/y
Technical	n_{H2E} (ER)	59	78	70	59	69	%
	Technical risk add-on	3-7	3-7	3-7	3-7	3-7	[-]
	Technical risk stand-alone	3-5	3-5	3-5	3-5	3-5	[-]

Table 14: Summary of the main technical and economical characteristics of four different RCG options and the ST-cycle. In this table, n_{H2E} is the heat to electricity efficiency.

5 Industrial applications

In this chapter, the technical and economical performance of the proposed RCG options A till D and a sole steam turbine and generator (ST) has been evaluated by use of two different industrial use cases:

1. **Stand alone RCG to a thermal asphalt cleaning system**
2. **RCG add-on to a metal fuel combustion system**

At first, the electricity markets have been analysed that guide as an input for the economical performance of the cycles.

5.1 The Dutch electricity market

The transmission system operator (TSO) have the mission to ensure that there is a balance between the supply and demand of electricity. They have adopted a price driven model where electricity suppliers and consumers can trade electricity on time-dependent markets:

- In the **long term market**, electricity is traded between four years and one month prior to delivery. These markets are mainly dominated by large electricity suppliers and consumers. The price of electricity that is traded on this market is commonly referred to as the wholesale electricity price (Ep_w).
- As the name suggests, in the **day-ahead market**, electricity is traded via an auction with hourly intervals for the coming twenty-four hours. Electricity suppliers and buyers are therefore able to ensure a fixed price to consume or deliver electricity for a specific moment in the next day. During the day, the **intraday market** is a place where energy consumers and suppliers are able to trade electricity 5 minutes prior to delivery. Together with the day-ahead market, the intraday market is commonly referred to as the electricity spot market.
- As last, there is the so-called **unbalanced market**. In contrast to the previous discussed markets, the unbalanced market is a real time market and used by TSO's to fill the real time gap between demand and supply of electricity. The price of electricity is regulated by the TSO and is affected by the real time difference in electricity supply and demand.

Between 2020 and 2022, electricity prices in both the wholesale (Ep_w) and unbalanced market (Ep_u) show major deviations. In 2020, the wholesale price of electricity was initially €130/MWh while in 2022 prices have inflated up to €160/MWh [9]. These deviations are also visible in the unbalanced market. For the year 2020 and 2022, a histogram of electricity prices on the unbalanced market are displayed in figure 33. While in 2020, the mean revenue of electricity on the unbalanced market (Ep_U) was equal to €32/MWh, the mean price in 2022 was equal to €234/MWh.

From the data-sets of the unbalanced electricity prices (Ep_u) of 2020 and 2022, the amount of hours where the electricity prices is below and above zero has been obtained. From the dataset, the corresponding mean electricity prices for these occasions has been calculated. These prices, including the yearly mean electricity prices for the wholesale and unbalanced market is listed in table 15. Thereby is considered that the plant is operational for 90% of the year.

	Runtime [h/y]				Electricity price [€/MWh]			
Year	$Ep_u > 0$	$Ep =< 0$	Ep_u	Ep_w	Ep_u >0	$Ep_u =< 0$	Ep_u	Ep_w
2020	7056	850	7884	7884	42.6	-38.8	32.8	130
2022	6761	1145	7884	7884	290.9	-95.5	234.9	160

Table 15: Operational hours and mean electricity prices of the unbalanced (Ep_u) and wholesale market (Ep_w) for the year 2020 and 2022.

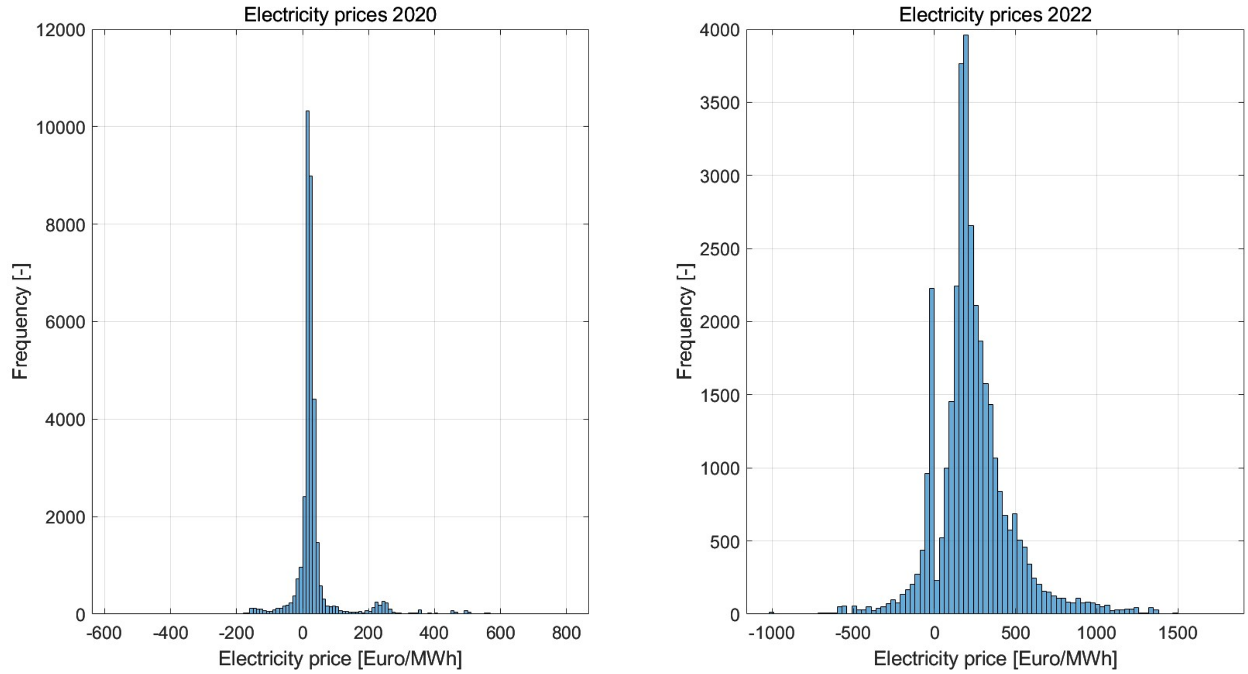


Figure 33: Histogram of the selling price of electricity on the dutch unbalanced market in 2020 and 2022. Data has been obtained from Dutch TSO Tennet [31]. The data set contains electricity prices with 15 minutes interval.

From this data, the potential revenue of a 1MWe electricity plant can be been computed for the year 2020 and 2022. In figure 34, the yearly revenue is computed for trading on the wholesale market, the unbalanced market, the unbalanced market when the electricity price is positive and on the unbalanced market when the electricity price is negative.

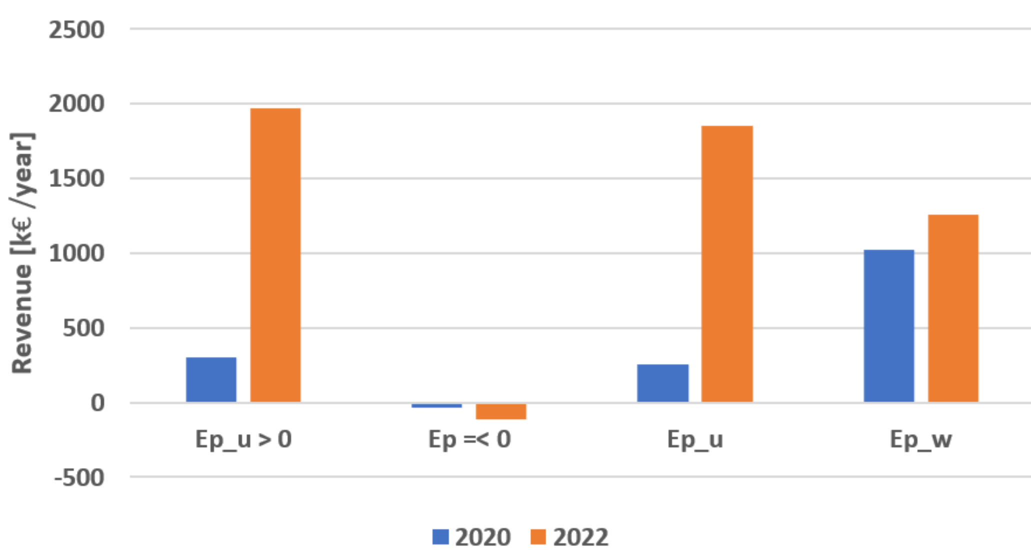


Figure 34: Potential electricity revenue per year for the long term electricity market (Ep_w), and the potential revenue on the unbalanced market when electricity prices are positive ($Ep_u >= 0$), negative ($Ep_u < 0$) and the total revenue when the unbalanced electricity prices are both positive and negative (Ep_u).

From this figure can be noticed that in the year 2020, electricity trading on the unbalanced market does not lead to a higher revenue than trading on the wholesale market. However, for the year 2022, trading on the unbalanced market can be lucrative. Despite the fact that trading on the unbalanced market instead of trading on the wholesale market for this year would have led to a higher revenue, also the contribution of negative energy prices is higher in the year 2022. This endorsed the potential benefit of the RCG-cycle, the ability to vary their flexible power output in a mater of seconds.

The existence of the unbalance market endorses the fact that real time demand and supply, and thus prediction of real time electricity prices is a tricky business. If the volatility and magnitude of the electricity prices will continue to increase or decrease, cannot be told with any certainty and is up to the reader. However, since for the year 2022 the electricity prices on the unbalanced market seem to be lucrative, the financial analysis of the use cases will be evaluated by use of the electricity prices of the unbalanced market for the year 2022.

5.1.1 Financial performance indicators

Levelized cost of electricty (LCOE)
The levelized cost of electricity is an important metric to assess the feasibility of an electricity plant. It is in essence the total life cycle cost of a plant divided by the total lifetime produced energy. It can be calculated by use of equation 30.

$$LCOE = \frac{C_i + Ct_o + Ct_f}{Et} = \frac{C_i + C_o \cdot y + F_c \cdot h \cdot y}{h \cdot y} \tag{30}$$

The initial investment costs (C_i) include the installed equipment costs of the plant and the operational costs (Ct_o) are defined as the expenses that are require to keep the plant operational (O&M costs). As can be noticed from equation 30, the LCOE is related to the total amount of hours the plant is operated. When the amount of years increases this results in a yearly reduction of the LCOE.

Yearly operated hours
Only a revenue can be made when the fuel price of electricity (F_c) is lower than the selling price of electricity (E_p). Therefore the RCG options should only be operated under these conditions. Since the distribution of electricity prices for the year 2022 are known, the amount of hours that a plant can be operational can be calculated by distracting the amount of hours when the cost price of electricity is lower than the selling price of electricity from the total amount of hours per year. Since the plants are subjected to maintenance intervals, it is assumed that the plants are operational for 90% of the time. Hence in the calculation of the hours, a capacity factor of 0.9 is introduced.

$$Hours = 0.9 \cdot (365 \cdot 24 - Hours(F_c < E_p)) \tag{31}$$

The mean electricity selling price ($\bar{E_{sp}}$) can consequently be calculated by taking the mean value of the electricity prices where the cost price of electricity is lower than the selling price of electricity.

Payback period (PBP)
The payback period (PBP) of the plant is equal to the year where the LCOE is lower than the mean electricity selling price.

5.2 Case 1: Asphalt thermal cleaning plant

Asphalt mainly contains sand, granulate, tar and bitumen. Recycling of tar-containing asphalt can be done by use of a thermal cleaning process. Heating of the asphalt results in the evaporation of the tar and bitumen and what is left is "cleaned" sand and gravel. Combustion of asphalt is a challenging process and cannot be done in a conventional furnace since the tar becomes a viscous and sticky substance at high temperatures. Therefore this thermal cleaning process is commonly done by use of a special type of rotating combustor: a rotary kiln (RK). An image of this furnace is displayed in figure 36.

Figure 35: Rotary furnace used for the recycling of tar-containing asphalt, adopted from [28]

A process layout of an asphalt recycling process is displayed in figure 36. The lay-out of the plant and the corresponding temperatures have been adopted from a pressure flow diagram (PFD) of an asphalt recycling plant near Eindhoven.

The polluted asphalt first enters a hot air driven pre-dryer (PDR), where the asphalt is heated to reduce the moister content. Next, it enters a natural gas fired rotary kiln (RK), where the tar and bitumen are evaporated. Heat from the hot polluted flue gasses are first applied to the pre-dryer (PDR) before it enters a natural gas fired oxidizer that demolishes parts of the hazardous pollutants. These "cleaned" flue gasses are on their turn guided through the pre-dryer where remaining heat is used to power the drying process of the asphalt.

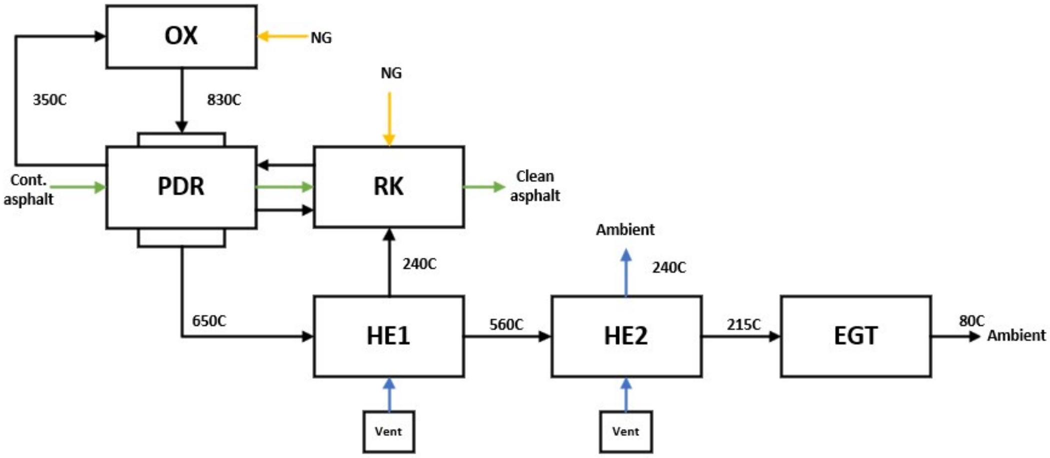


Figure 36: Process layout TAR recycling plant. Temperatures and process layout has been adopted from a pressure flow diagram (PFD) of a thermal asphalt cleaning plant near Eindhoven. The plant contains an oxidizer (OX), paddle dryer (PDR), rotary kiln (RK), two heat exchangers (HE), ventilators (Vent) and an exhaust gas treatment (EGT).

The flue gasses that leave the pre-dryer (PDR) can reach temperatures up to 650C. The filter and treatment of the exhaust gasses requires a temperature of maximum 250C. To reduce the temperature of the flue gas, two heat exchangers are used; HE1 and HE2. HE1 is used to extract heat from the hot flue gasses to pre-heat the rotary kiln, which reduces the consumption of natural gas. In the second heat exchanger, hot flue gasses are actively cooled by use of a ventilator that consumes 200kWe per hour to reduce the temperature of the flue gas back to 215C.

5.2.1 Layout waste heat fueled stand alone RCG and rankine cycle

In order to re-use to excess heat of the asphalt combustion plant, the RCG cycle is placed between HE1 and HE2. Besides the air heat exchanger, the plant requires an additional boiler since no steam is present at the site. The TIT of the power turbine is considered to be higher than the TIT of the steam turbine. Therefore it makes sense that the heat exchanger of the power turbine is placed in front or parallel to heat exchanger of the boiler. The remaining heat at the outlet of the power turbine can be fed into the atmosphere since in this case, it is desired to cool the hot flue gasses.

Since the plant where the cycle is located does not consume any steam, the steam turbine cycle is executed as a close loop cycle. When desired, the heat that is present in the condensor can be re-used or a air cooled condensor/cooling tower can be applied.

The cost price of the boiler and air heat exchanger is determined by the heat exchangeable area. This is related to the overall heat transfer coefficient and the amount of heat that is extracted by the heat exchanger. Considered that the heat transfer coefficients and the extracted heat are independent from the location of the heat exchanger, the heat exchangeable is mainly affected by the inlet temperature of the heat source. To conclude, the heat exchangeable area and thus the cost price of the heat exchanger, can be minimized when it is placed at the highest temperature of the heat source. Therefore the layout of figure 37 is proposed, a configuration where two heat exchangers are placed in parallel.

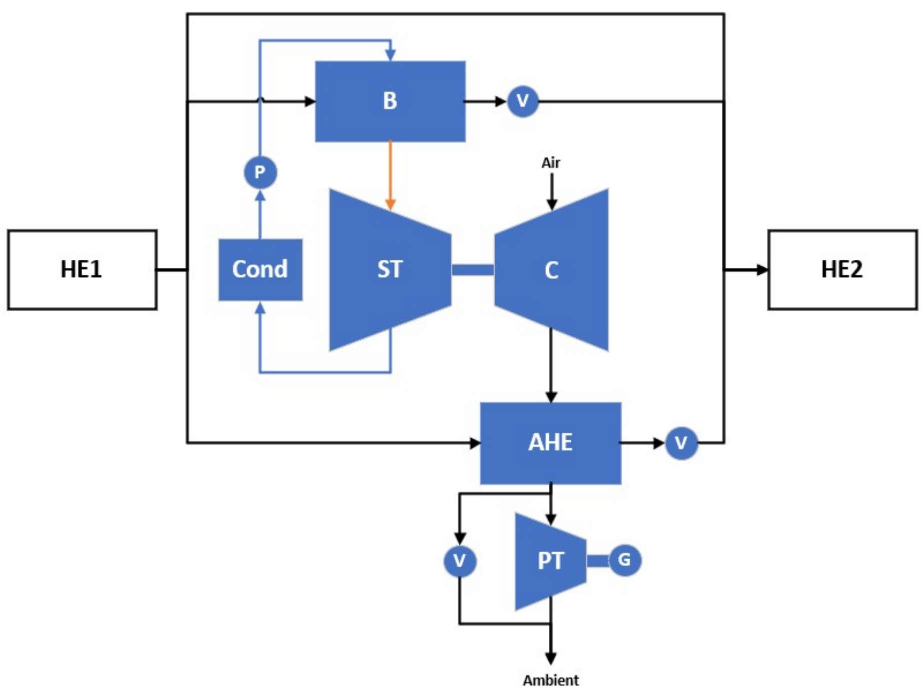


Figure 37: RCG layout for apshalt granulate plant. RCG cycle is placed between both flue gasses coolers HE1 and HE2. The system contains a boiler (B) with pump (P), steam turbine (ST), condensor (Cond), compressor, heat exchanger (HE), power turbine (PT), valves (V), generator (G) and an air heat exchanger (AHE).

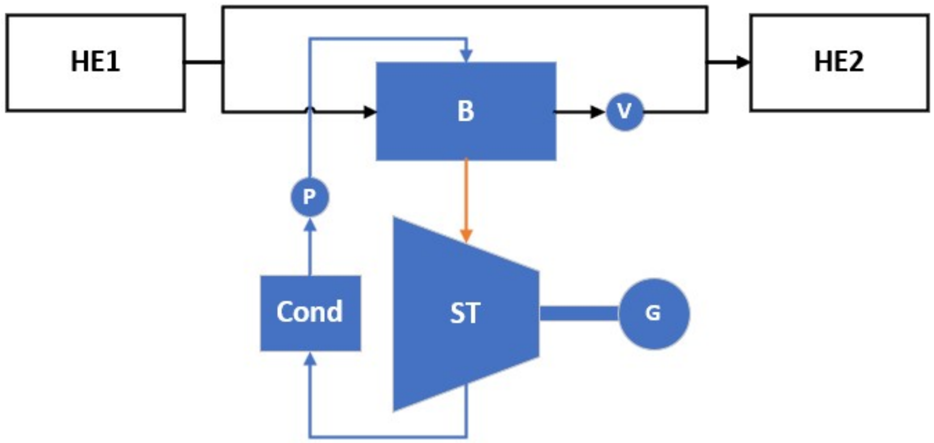


Figure 38: Proposed layout of a waste heat driven rankine cycle for electricity generation at a Tar-granulate plant. The layout contains a boiler (B), pump (P), steam turbine (ST), condenser (Cond), generator (G) and a valve (V).

5.2.2 Steam properties

The steam properties of a closed-loop RCG cycle from chapter 4 have been used. These are in more detail given in appendix B. This means that the following assumptions have been made:

- The condenser pressure is equal to 1 barg.
- The high pressure steam is has a pressure of 29 barg and a temperature of 400C.

5.2.3 Financial performance

Although some could say that the cost price of the fuel for waste heat is negative since a ventilator is used to actively cool the heat, it is common practice to evaluate the fuel price of waste heat as €0/MWh. The levelized cost of electricity (LCOE) has been computed for RCG option A, C and D an for a conventional CHP cycle. RCG option B has not been taken into account since the power turbine of this option requires a TIT that is higher than the temperature of the heat source. The LCOE has been computed for up to 15 years, since this is expected to be a reasonable investment horizon for the plant owner. The yearly LCOE for this period is displayed in figure 39. Previously was concluded that the RCG options should only be operated when the fuel price per hour is lower than the potential hourly revenue. Since the sole steam turbine and generator is not able to vary their electrical power output, these are considered to run for the entire year. For the yearly operational hours, a capacity of 90% is assumed. The amount of operated hours per year, the selling price of energy and the payback period are listed in table 16.

Option	runtime [h/y]	Return Sp [€$/MWh$]	PBP [y]
A	6761	291	0.9
C	6761	291	1.2
D	6761	291	1.2
ST	7884	235	0.5

Table 16: Hours per year, return per MWe and the payback period

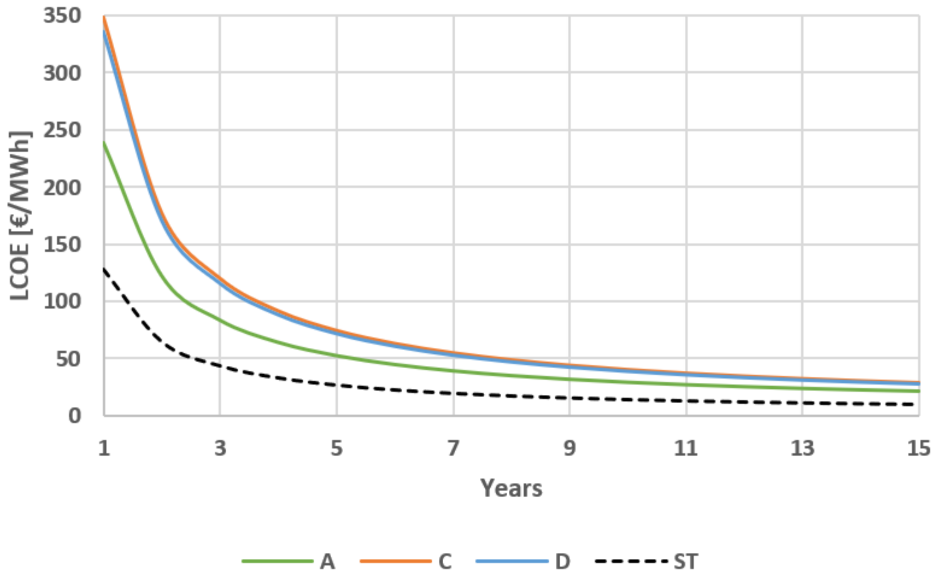


Figure 39: Yearly levelized cost of electricity (LCOE) for option RCG option A, C, D and a steam turbine and generator (ST) considered for a 1MWe plant with waste heat as fuel source.

5.3 Case 2: Metal fuel combustion

In the past couple of decades, intensive research has been carried out in the field of low-carbon renewable energy carriers such as electrochemical batteries, hydrogen, and biomass. Berghtorson et al [4] concluded that future renewable energy carriers should match fossil fuels in terms of energy -and power density so they can be transported in a commodity market and used for high power generation systems. According to that reasoning, metal fuels such as iron, boron, and aluminum are proposed as a promising alternative to the current mix of fossil fuels for power generation systems. From an economic perspective, iron powder is a very promising low-carbon renewable energy carrier as it is widely available and can be harvested and transformed into powders at a relatively low cost.

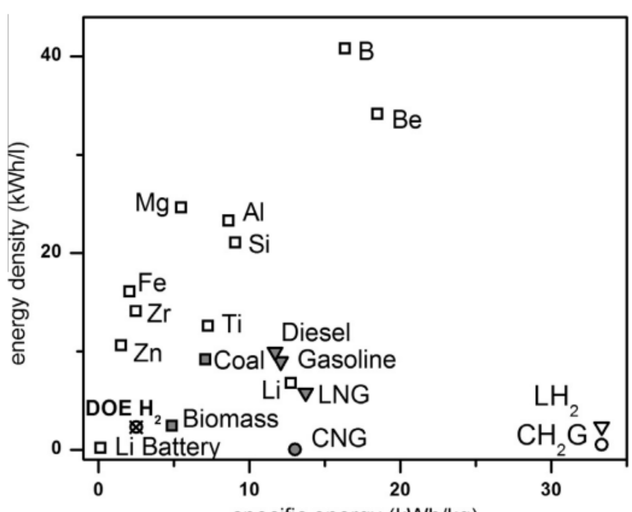


Figure 40: Energy density [kWh/L] and specific energy [kWh/kg] for a variety of solid (square), liquid (triangle) and gaseous (circle) energy carriers. Unfilled symbols indicate potential zero-carbon fuels. Figure has been adopted from [4].

The amount of chemical energy that is present in the metal can be harvested in the form of heat by use of an exothermal reaction with air: by direct combustion of metal fuel powders. The oxidized iron particles (rust) can be regenerated into iron by use of hydrogen. The hydrogen can be generated by use of renewable energy sources, minimizing the total carbon pollution of the cycle. From all metal fuel candidates in figure 40, it is expected that iron can have a relatively high regeneration cycle efficiency as the oxidized iron particles are relatively easy to collect and regenerate w.r.t. other metal fuel candidates.

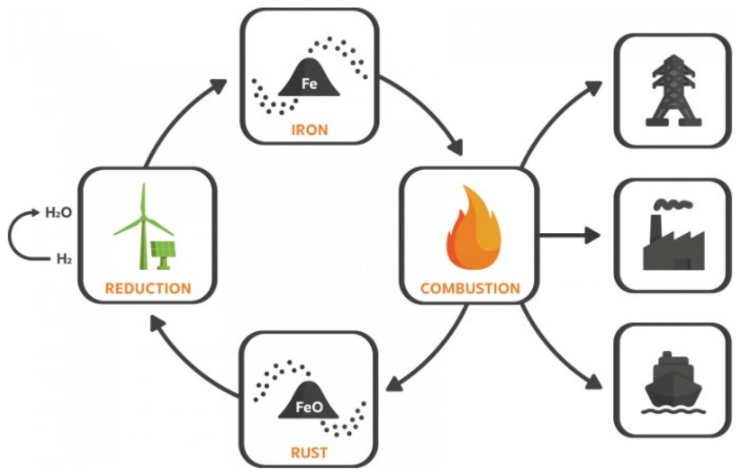


Figure 41: Metal fuels combustion and regeneration process. Figure is adopted from [4].

Berghtorson et al proposed an iron fuel combustion engine that can be used for different applications [4]. This can be for example direct use for drying purposes, heat engines in automotive or shipyard propulsion, to heat water for district heating networks or to generate steam for industrial or combined heat and power (CHP) applications. This principle formed the bases for the first 100kW metal fuels combustion system, that demonstrated the feasibility of the metal fuels combustion cycle on a commercial scale. A photo of the test-setup is displayed in figure 42.

Figure 42: First experimental metal fuels combustion system, for iron fuel regeneration process. Photo is adopted from [17]

Janica et al recently researched the potential of retrofitting the iron fuel combustion system to an existing power plant [19]. This study included a plant layout of a potential combustion system which is displayed in figure 43. The proposed installation contains a feeding system where iron particles and air are combined and guided to a combustor. In the combustor, iron particles are combusted and transform into their final oxidized end products. The heat that is released from the combustion process, can be extracted by use of a boiler system, where water is transformed into steam. After combustion, hot flue gasses are guided to a cyclone that filters the combusted end products for regeneration purposes. The hot flue gasses can partly be fed back into the combustion process to minimize heat losses and increase the overall efficiency of the cycle. At the outlet of the cycle, the end product of the combustion process is nitrogen rich waste gas.

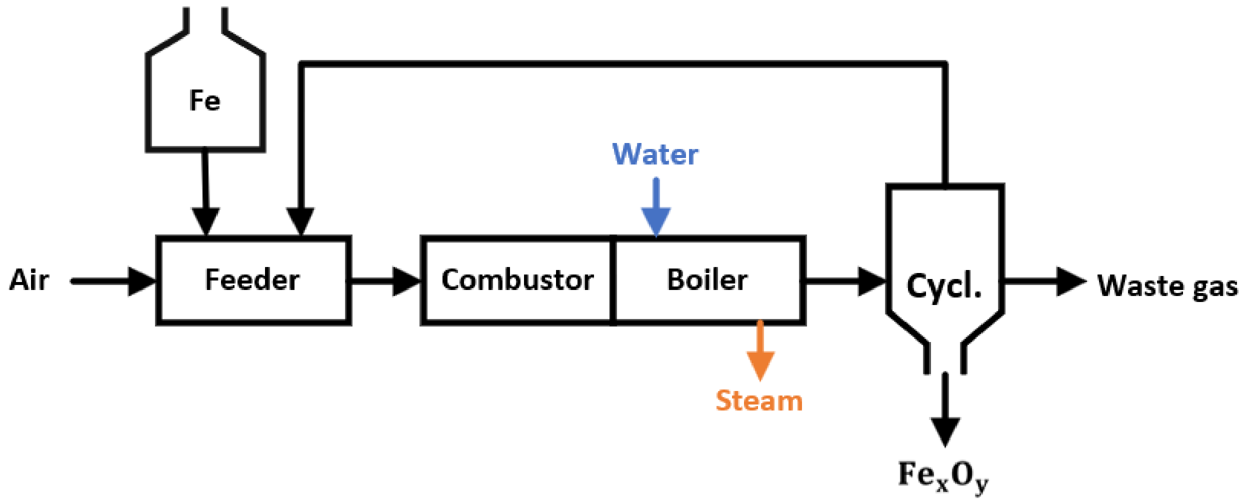


Figure 43: Layout iron fuel combustion system with iron particle dispersion system (Fe), feeder, combined combuster and boiler and cyclone (Cycl.)

The composition of the oxidized end products, wustite (FeO), hematite ($Fe2O3$) and magnetite ($Fe3O4$) relate to the combustion conditions and is a topic that requires further research. Samples from the combusted end products in the 100kWe test setup show a rough division of 50% hematite and 50% magnetite [6] and therefore these conditions are assumed for the evaluation of the metal fuels combustion case.

5.3.1 Technical and economical feasibility study iron fuel technology

In 2021, a techno-economic feasibility study has been performed for the potential of an iron fuel combustion system [24]. The report assesses the technical and economical potential of three different potential market applications; district heating, process heating and electricity generation. In the case of district heating, an economizer is used to heat water up to temperatures between 45-120°C. In the case of industrial heating, indirect heating by use of water (e.g. generation of steam) has been proposed. Consequently, the generated steam can either directly be used in the plants industrial process, or can be used for local electricity generation. Lastly, the application of electricity generation mainly focuses on high power (>50MWe) decentralized and centralized (>500MWe) electricity generation plants.

In the cases of process heating and electricity generation, the combustion cycle includes generation of steam, which is essential for the functionality of the RCG and sole ST cycle. Hence these cases are most relevant within this research. Since it is expected that the case of process heating is developed prior to the multi mega watt electricity generation system, the technical and economical feasibility of an RCG and sole steam turbine and generator will be evaluated for this use case.

The method of transportation and regeneration of the iron and iron oxide products has major influence on the cost price and carbon emissions of the metal fuels. Since iron particles do not contain carbon molecules, the combustion process alone does not introduce carbon emission. However, the regeneration and transportation process of iron and iron oxide introduce emissions. The amount of emission highly depends on the method of transportation and the method of hydrogen generation that is used for the regeneration process. The technical and economical feasibility study describes three different scenario's regarding the regeneration process; local regeneration and combustion, regeneration and transportation at short distances (3.500km) and regeneration and transportation at long distances (10.000km). The carbon emissions and fuel costs for the three regeneration processes are listed in table 17.

	Carbon emission [kgCo2/GJ]	Fuel costs [€/t]
Local regeneration	17	154
Regeneration 3.500km	21	129
Regeneration 10.000km	29	120

Table 17: Carbon emissions and fuel costs of iron and iron oxide transportation and regeneration process. The values have been adopted from [24] and are based on hydrogen production by use of natural gas.

The research shows that transportation at long distances reduces the fuel costs, while it increases the carbon footprint. At long distances, it is the other way around. Therefore is assumed that the regeneration process occurs at transportation of short distances, which is considered to be a balance between fuel costs and carbon emissions. For the regeneration process, it is assumed that the hydrogen is generated by use of natural gas. Although this is a bit conservative since the use of renewable energy sources would decrease the carbon emission in the regeneration process, it is most likely this will be the case for the next couple of years due to the lack of renewable energy sources in the current national energy supply [24].

5.3.2 Metal fuels fueled RCG add-on

The combustion cycle from figure 43 leaves room for the implementation of an RCG. In this scenario, it is assumed that the steam is consumed by the industrial process of a client and hence it is executed as an open-loop cycle. It contains a high pressure steam line that can be used by the consumer to power a steam turbine that drives prime movers (compressors) or electrical generators to generate (base-load) electricity. A low pressure steam line, at the outlet of a steam turbine can consequently be used as a heat source in for example drying or pasteurization.

Since the RCG is equipped to an existing furnace and boiler system, the performance of the RCG options and the sole steam turbine and generator are evaluated as an add-on. Thereby part of the steam that is generated in the boiler can be used to drive the steam turbine. The second heat exchanger that is used in the RCG cycle can be used to pre-heat air prior to expansion over the power turbine.

The efficiency of the iron combustion cycle mainly depends on the end temperature of the waste gas and the oxidized iron particles. Considered that no energy re-used at the outlet of the cyclone, the temperature of the regenerated hot flue gas (from the cyclone to the feeder) and the temperature at the outlet of the cyclone are equal. It is expected that the temperature at the outlet -and at the regeneration stream have temperatures that are far below the considered turbine inlet temperature (500-1000°C) of the air heat exchanger. Hence it is assumed that the air heat exchanger is integrated in or between the boiler tubes. Therefore the TIT of the RCG's air cycle can reach up to temperatures of 1000°C. A layout of a proposed RCG add-on to an metal fuels combustion and boiler is displayed in figure 44.

When no air is expanded over the power turbine, the heat that is extracted by use of the air heat exchanger can be regenerated at the outlet of the bypass valve and power turbine. In the proposed configuration it is chosen to directly feed the air back into the cycle. However, this affects the combustion properties (e.g. air to fuel ratio) which is expected to be problematic for small combustion systems and for RCG configurations that require a high mass flow of air. Alternatively, heat can be indirectly regenerated by use of a second heat exchanger that can be used to pre-heat the air before it is applied to the feeder system. The downside of this is that this introduces additional costs and not all heat can be regenerated since this would require an infinitely large heat exchanger.

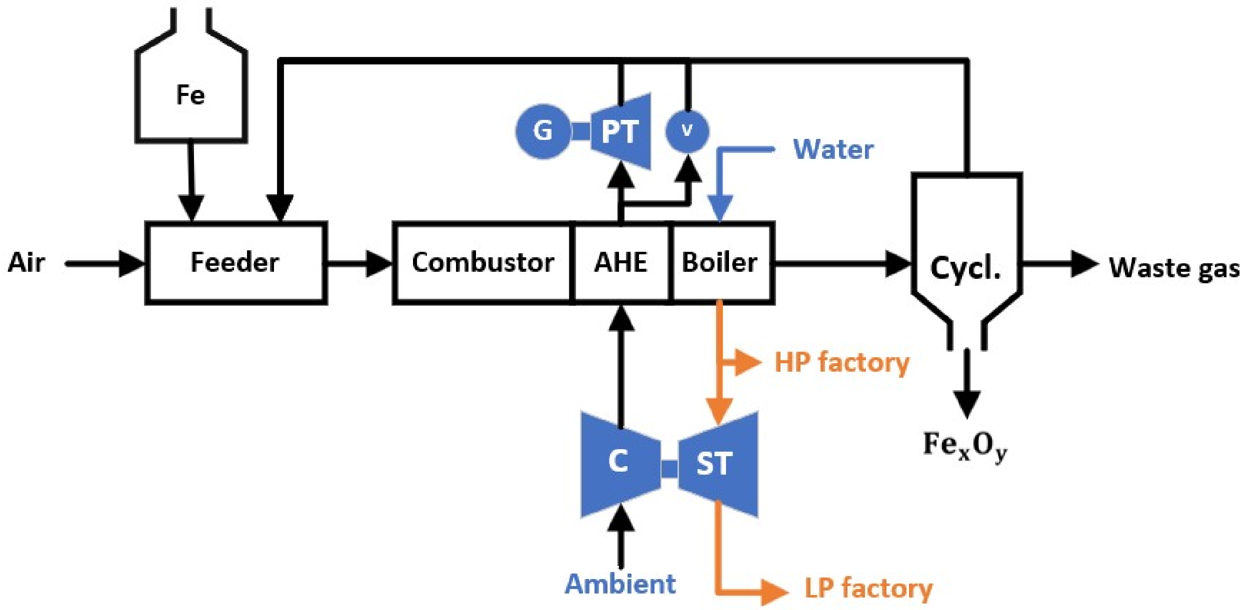


Figure 44: RCG add-on to metal fuels combustion system. The excisting cycle, containing an integrated combustor and boiler system with a cylcone (cylc.) for particle filtration is displayed in black. The additional required RCG componentens, including a steam turbine (ST), compressor (C), air heat exchanger (AHE), regenerative heat exchanger (RGE), power turbine (PT), generator (G) and valves is displayed in blue.

In figure 45 the layout of a steam turbine and generator for a metal fuel combustion cycle is displayed.

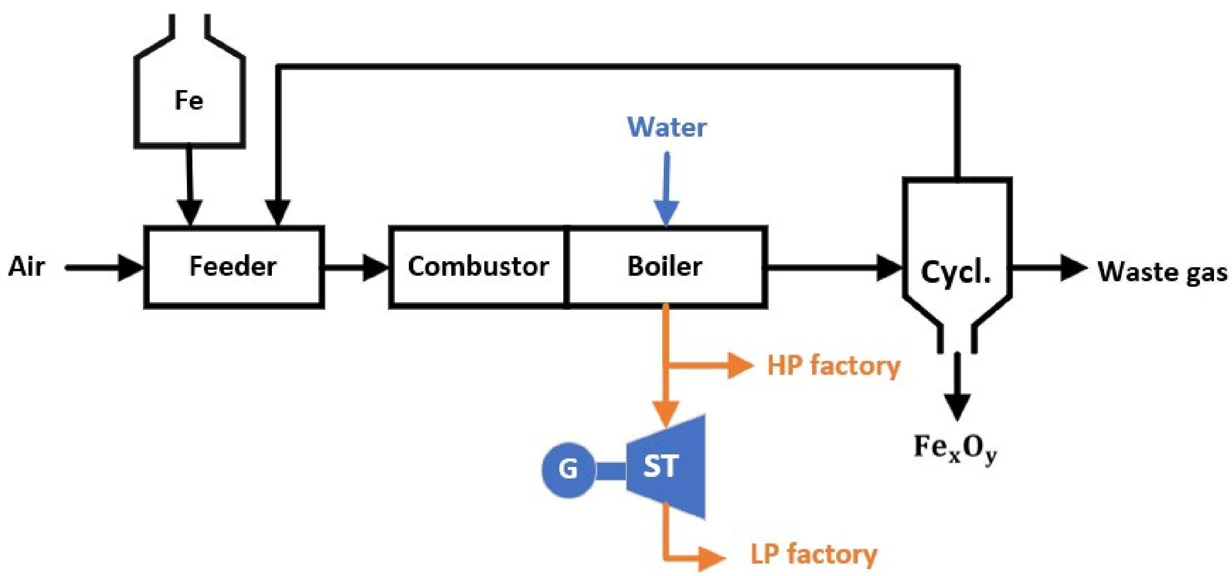


Figure 45: Proposed steam turbine (ST) and generator (G) configuration for a metal fuels combustion cycle.

5.3.3 Steam properties

The steam properties of an open-loop RCG cycle from chapter 4 have been used. These are in more detail given in appendix B. This means that the following assumptions have been made:

- The properties of the water (prior to heating) are 20C and 29 barg.
- The high pressure steam is 29barg at 400C.
- The low pressure steam is 1.5 barg saturated dry steam.

5.3.4 Financial performance

The hourly fuel cost of the metal fueled cycles are determined by the cost of iron per kg and the required mass flow of iron per hour. The required mass flow of iron can be calculated by dividing the required heat of the cycle by the storage capacity of iron. Since the combustor and boiler system is assumed to be "non-ideal", (the stack temperature is above the ambient temperature) a boiler efficiency is taken into account. The fuel cost per hour are calculated by use of the following equation:

$$Fc = \frac{(\frac{Q_{fuel}}{n_B})}{\Delta E_{fe}} \cdot 3600 \cdot C_{fe} \tag{32}$$

Since the energy storage capacity of wustite and hematite are equal to respectively 6.8MJ/kg and 7.4MJ/kg, an overall energy storage capacity of 7.1MJ/kg is assumed. The calculation of the cost price of the iron fuel is therefore subjected to the following assumptions:

- The storage capacity of iron (ΔE_{fe}) is equal to 7100kJ/kg.
- The boiler efficiency (n_B) is equal to 0.92, an assumption that is adopted from [24].
- The cost price of iron is equal to €0.13/kg.

In table 18, the amount of heat that is required per option to generate 1MWh of electricty, the corresponding hourly fuel price and the yearly operated amount of hours (runtime) are listed. The runtime of the RCG-cycles has been evaluated for the yearly amount of hours where the cost price of electricity is below the fuel price. A capacity factor of 90% has been taken into account.

Option	Heat (Q) [kWth/MWe]	Fuel price (F_c) [€/MWhth]	Runtime [h/y]	Ep [€/MWhe]	PBP [Y]
A	1677	123	5995	317	1.4
B	1282	94	6329	306	1.6
C	1429	105	6219	310	1.9
D	1679	123	5995	317	2
ST	1429	105	7884	235	1

Table 18: Metal fuels cycle required heat, cost price, fuel, revenue, runtime and payback period

Since the hourly fuel price is known, the LCOE of electricity is computed for all RCG options and the ST-cycle. By use of this, the levelized cost of electricity (LCOE) has been computed up to 15 years.

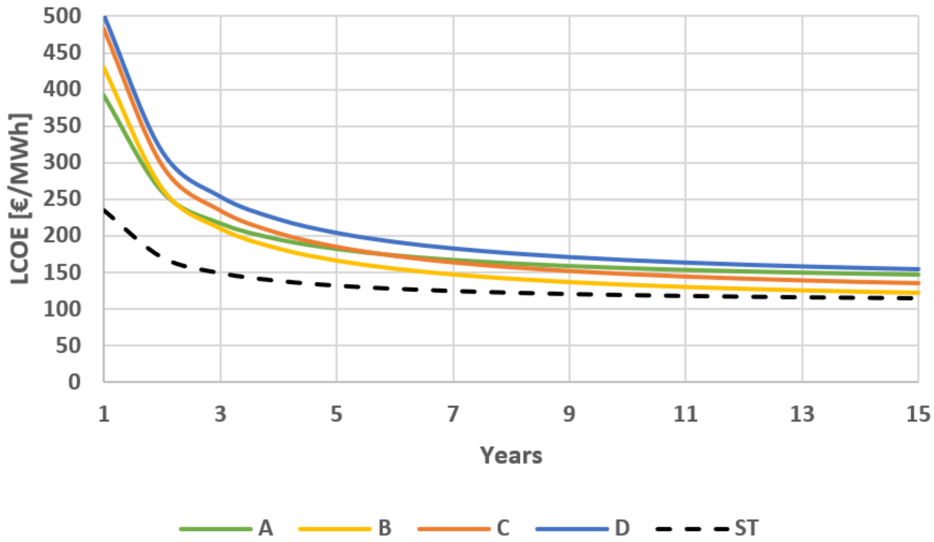


Figure 46: Levelized cost of electricity (LCOE) per year for RCG options A till D and the sole steam turbine and generator, executed as an add-on to an excisting boiler and furnace system.

5.3.5 Carbon emissions

For the calculation of the carbon emissions, the following assumptions have been made:

- The carbon emission for regeneration and transportation of medium distance (3500km) has been used which leads to a carbon emission of 21 $kgCO_2/GJ$.
- The boiler efficiency is equal to 0.92.

The amount of carbon emissions per MWhe electrical power output can consequently be calculated by use of the following equation:

$$e_{CO_2}(MWhe) = \frac{Q_{fuel}}{n_B} \cdot 21 \cdot \frac{3600}{1000} \tag{33}$$

For the RCG options A till D and the sole steam turbine and generator (ST) the mass of CO2 per MWh is displayed in table 19.

Option	Required fuel [MWh]	Carbon emission [kgCO2/MWhe]
A	1823	6563
B	1391	5015
C	1553	5591
D	1824	6566
ST	1553	5591

Table 19: kg CO2 emissions per MWh electrical power output

6 Multi criteria decision making analysis

The proposed RCG options and the sole steam turbine are subjected to technical, social and economical performance indicators. In the decision making process of the most suitable technology, these performance indicators can become conflicting criteria. This problem can be overcome by use of a multi criteria decision making analysis (MCDMA). Within the MCDMA there are a variety of methods that can be used to select the most applicable option. Examples are the analytic hierarchy process (AHP), the technique for order of preference by simularity to ideal solution (TOPSIS) and the weighted sum model (WSM) [23] None of these techniques are in essence good or bad, however, once a methodology is selected, the decision maker should be aware of the limitations of the used methodology. Due to their simplicity and transparency, the weighted sum method is used in this MCDMA since this method is transparent, straightforward and accessible to non-experts in the field of MCDMA [22].

Via the WSM methodology, a variety of performance indicators are compared and normalized. The normalized indicators are consequently weighted to adjust the importance per criteria. The scores are than aggregated to an overall ranking score. The option with the highest score can be considered to be the most suitable option.

The first limitation of this methodology is that it is assumed that the scale of a criteria is linear, which is not necessarily true. This can be the case in for example the assessment of the maturity of a technique which is commonly done by TRL's, a non-linear scale. Secondly, the criteria of weights are somewhat subjective and are influenced by the bias of the decision-makers. At last, the weighted sum method can be sensitive and hence small changes in weights can lead to significant changes in the end results.

In the selection of the most suitable option two stakeholders have been taken into consideration; the society and the plant owners. Often, the weights of the criteria are established by use of discussion that are held with peer groups that represent all stakeholders. This not only limits the bias of the decision makers, but can also be used to establish a list of the performance indicators. Since this process is time consuming and the weighting factors for these particular case are not covered by literature, three different weighting factors for the performance indicators have been proposed. The weighting factors are varied to derive a decision that is based on the perspective of the society and the plant owner and can be used as a method to get a sense on the sensitivity of the performance indicators.

6.1 Performance indicators

Ideally, the performance indicators are selected such that these are independent from technical parameters. However, when this is done this reinforces the linearity limitation of the chosen methodology. The maintenance costs are for example an important parameter but their contribution in the levelized cost of electricity (LCOE) can be negligible when the cycle is subjected to high initial investment costs and fuel costs. Thereby, when solely individual parameters are compared, it is not possible to evaluate the performance from a social and economical perspective. This is relevant in for example the power to heat efficiency of the evaluated cycles, that affects both the LCOE and the carbon emissions.

6.1.1 Technical performance indicator

The technical risk is selected as the main technical performance indicator. This indicator does not indicate the degree of security, since previously was assumed that the safety of an technique can be optimized by use of sufficient safeguard. The technical risk in this research represents the degree of risk that the cycle will not function. These are measured by use of the technology readiness levels (TRL's).

6.1.2 Economical performance indicators

The used economical performance indicators are:

1. The initial investment costs [k€]
2. The payback period (PBP) [y]
3. The levelized cost of electricity (LCOE) [€/MWe]

The initial investment costs and the payback period highly influence the willingness to invest in the proposed options. A high initial investment costs can potential results in no investment since the investor (e.g. plant owner) does not have sufficient funds. The payback period is not solely related to profit but can also be considered as the amount of risk is accompanied by the investment. This is especially relevant within this research since it is assumed that the payback period is determined for high volatile electricity prices on the unbalanced market of the year 2022. The LCOE contain the initial investment costs, the maintenance costs and the fuel costs. Previously was concluded that the LCOE reduces during the lifetime of the plant since the initial investment costs are spread out over multiple years. To minimize the influence of the initial investment costs during the evaluation of the LCOE, these have been used at a lifetime of 15 years. This is assumed to be the minimum investment horizon of the plant owner.

6.1.3 Social performance indicators

The social performance indicators that are used in this MCDMA are:

1. The carbon polution [kgCO2/MWe]
2. The capability to peak shave [Yes/No]

6.2 Weighting factors

In table 20 an overview of the different performance indicators is displayed. Thereby is given which performance indicator relates the most to either the plant owner or the society.

Performance indicator		Society	Plant owner
Social	Carbon emissions	X	
	Peak shaving capability	X	
Technical	Technology readiness level		X
Economical	Investment costs		X
	Payback period		X
	Levelized cost of electricty after 15 years		X

Table 20: Social, technical and economical performance indicators. Thereby distinction is made in which performance indicator relates most to either the society or the plant owner.

In this research, three different weighting factors have been used that lay emphasis on the perspective of either the society or the plant owner.

- **Societal perspective**: 70% of the weightings have been assigned to social indicators and 30% has been assigned to economical and technical performance indicators.
- **Societal - plant owner perspective**: The weighting factors of the social and economical and technical performance indicators are equally distributed.
- **Plant onwer perspective**: 70% of the weightings have been assigned to economical and technical performance indicators and 30% has been assigned to social indicators.

These have led to a distribution of the factors which are displayed in figure 47. The exact weighting factors per performance indicator are displayed in appendix D. In the societal perspective, most weights are assigned to the carbon pollution and the peak shaving capability, which have been equally distributed. In the weighting factors of the plant owner, most emphasis is laid on the economical and technical performance indicators. The technology readiness level, the investment costs and the payback period all relate to the willingness to invest, while the LCOE is more related to profit that can be generated over time. Therefore the LCOE receives the same weight as the combined remaining economical and technical performance indicator(s). In the tar-granulate case, the carbon emissions are equal for all considered options. Therefore the weight of this performance indicator has been distributed among the other technical performance indicators.

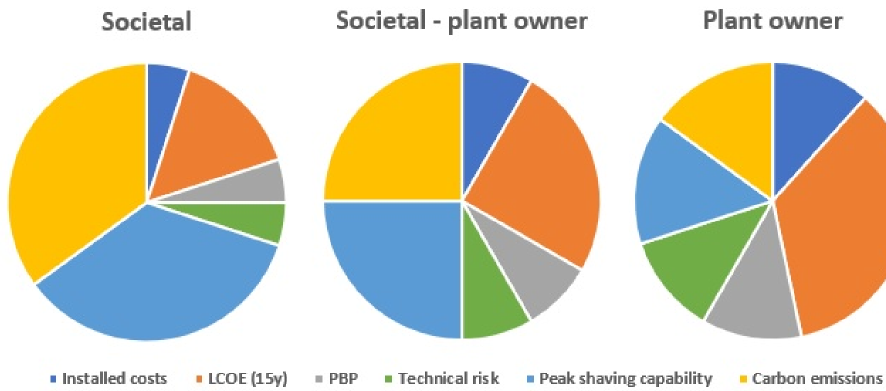


Figure 47: Three different weighting factors where either more emphasis is placed on the perspective of the society, the plant owner, and a weighting distribution that evaluates both perspectives equally.

6.3 Tar-granulate

The outcome of the multi criteria analysis from a societal and plant owner perspective for the tar-granulate case displayed in figure 48. In this comparison, RCG option B is not taken into account since the required turbine inlet temperature is lower than the temperature of the heat source.

From this overview can be noticed that RCG option A is the best option when a sole societal perspective is used and when the systems are compared from both a societal and plant owner perspective. When most emphasis is laid on the perspective of the plant owner, the sole steam turbine is considered to be the best option.

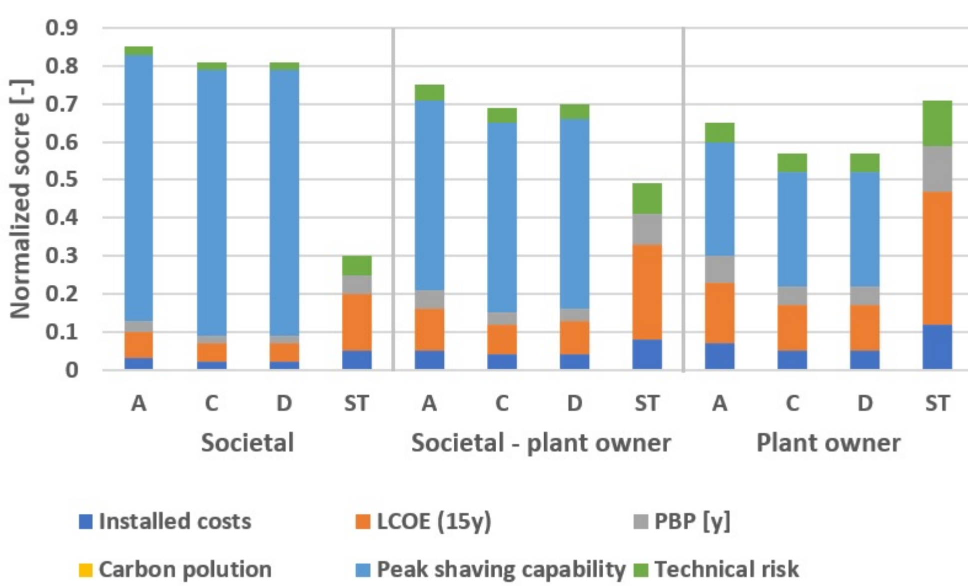


Figure 48: Normalized scores of the MCDMA for a scenario that emphasis social and economical criteria and equally distributed social - economical criteria of the tar-granulate case.

One should be aware of the fact that, especially from the societal perspective, the peak shaving capability receives high weights due to the lack of carbon emissions in this particular use case. When this performance indicator becomes less relevant, the sole steam turbine option becomes more appealing. Since all RCG options in this analysis have equally operation and maintenance costs and the fuel price is equal to €0/MWe, the initial investment costs have a major contribution to the LCOE and the payback period. In RCG option A, the inital investment costs are the lowest, while in RCG option C and D these are slightly lower. Therefore RCG option A remains the most appealing RCG option viewed from all perspectives.

6.4 Metal fuels

The outcome of the multi criteria analysis for the social, social-economic and economical weighting distribution for the metal fuels case is displayed in figure 49. From this overview can be noticed that from a societal and plant owner perspective RCG option B is the best option.

RCG option B has the highest heat to electricity efficiency of all considered option. Therefore the generation of 1MWh electricity leads to the lowest amount of carbon pollution and is favourable regarding the LCOE. From the considered RCG options, it thereby has the lowest initial investment costs. When the choice is considered from the plant owners perspective, the peak shaving capability becomes less relevant. Therefore the steam turbine option becomes more appealing, mainly because it has substantial lower initial investment costs than the RCG options. Due to the relatively high heat to electricity efficiency, it also scores relatively high on the carbon pollution and LCOE indicators.

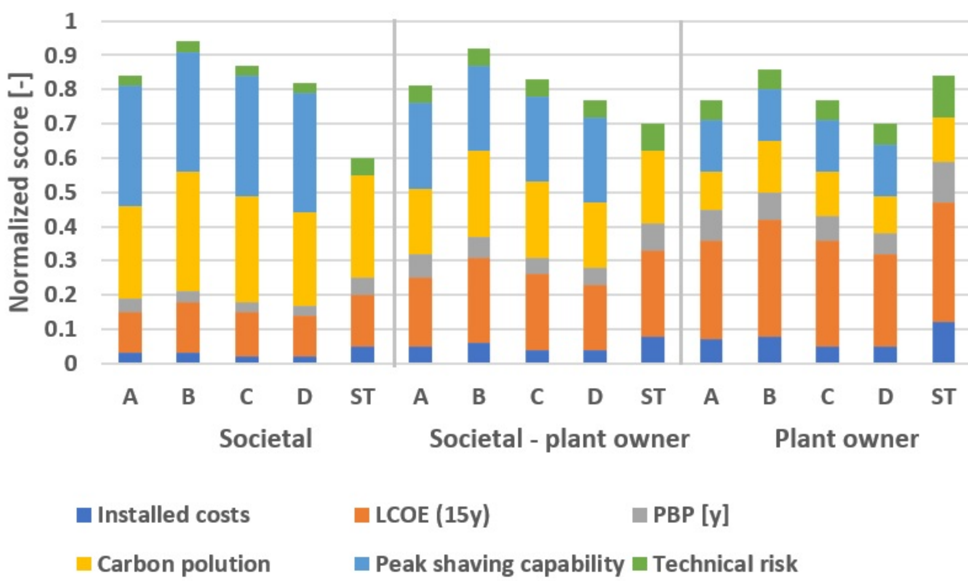


Figure 49: Normalized scores of the MCDMA for a scenario that emphasis social and economical criteria and equally distributed social - economical criteria of the metal fuels combustion case.

7 Conclusion

A RCG-cycle is made from a variety of turbo-machinery components such as a compressor, steam turbine and power turbine. These are available in different types and sizes. By use of a literature study (chapter 3), the most promising component options have been compared in price and technical performance.

A market study (chapter 4) has been performed and "off the shelf" components have been selected. In the RCG-cycle, both the steam turbine and compressor and the power turbine and generator are physically coupled. Therefore these drive-trains have been developed independently. For the steam turbine and compressor drive train, an impulse steam turbine with a single stage centrifugal compressor is proposed. For the power turbine and generator drive train, two appealing options have been established which leads to two different RCG configurations. The first configuration makes use of a hot gas expansion turbine including helical gearbox and electrical generator. The second configuration uses four automotive turbocharger turbines combined with a planetary 4-1 gearbox and induction generator.

The two power turbine options can be operated on different turbine inlet temperatures (TIT) and pressure ratio's. Per power turbine option, these have been varied and consequently this has led to four different RCG options, referred to as option A till D. For RCG option A and B, the automotive turbocharger turbines are used with a TIT of 500°C and 1000°C. For RCG option C and D, the maximum TIT of 500°C is used while the pressure ratio is respectively 1.5 and 3. As a result, the heat to electricity efficiency of option A and D is below the heat to electricity efficiency of the steam turbine and generator, while option B and C have a comparable or higher heat to electricity efficiency.

The technical and economical performance indicators of the RCG options have been compared to a conventional steam turbine and generator. The initial investment costs of RCG option A and B is expected to be 1.5 times the costs of a steam turbine and generator, while the initial investment costs of option C and D are estimated at 2.5 times the costs of a steam turbine and generator. The operation and maintenance costs of all RCG options is expected to be 4 times the operation and maintenance costs of the steam turbine and generator.

The economical performance of the RCG options and the steam turbine and generator have been evaluated by use of two use cases (chapter 5). The first use case is applied at a Tar-granulate recycling plant and the second case makes use of a iron fuel combustion plant. The main difference between these cases is the used heat source. The tar-granulate case makes use of low temperature waste heat while the metal fuels combustor uses iron as a fuel. In the tar-granulate case, waste heat is used and therefore the carbon emission and cost price of the fuel is zero. In the metal fuels combustion case, iron fuel is used that is accompanied with fuel costs and carbon emissions. The economical evaluation of both cases is based on the unbalanced electricity market. In this market, prices vary in real time and can become negative when a surplus of electricity is applied to the grid. For the year 2022, the potential revenue can be higher than the conventional wholesale electricity market and therefore the economical performance of the cycles has been evaluated for this particular year. The main benefit of the RCG cycles with respect to the steam turbine cycle is these are able to limit power generation when the cost price of fuel is lower than the cost price of electricity which can increase the potential revenue of the cycle.

In the tar-granulate case, the levelized cost of electricty (LCOE) solely depends on the initial investment costs and the O&M costs while in the metal fuels case these also include the fuel costs. In both cases, the steam turbine and generator is able to achieve a lower LCOE and payback period which can be assigned to the lower initial investment costs, O&M costs and the relatively high heat to electricity efficiency. Since the tar-granulate case is driven by waste heat, operation of the RCG cycle does not introduce carbon emissions. For the metal fuels case, the combustion reaction does not generate carbon emissions, however the regeneration process does. Since the cycle with the lowest heat to electricity efficiency consumes the lowest amount of iron, these cycles also pollute the lowest amount of carbon emissions.

In the decision making process (chapter 6), the most suitable option is evaluated per use case. Thereby there are two shareholders with conflicting interests. On one hand, there is the plant owner that desires favourable economical performance. On the other hand there is the society that benefits from peak shaving capabilities for grid stability and desires low carbon emissions. In order to overcome these conflicting interests, a multi-criteria decision making process is used. During this process, key performance indicators of the different cycles have been established. Consequently, different weighting factors have been assigned to each performance indicator. Lastly, these performance indicators have been normalized and aggregated. The option with the highest score can be considered as the most desired option.

At first, the interests of both shareholders has been weighted equally. Additionally, two analysis have been carried out where more emphasis is laid on the interests of the plant owner and the society. In the tar-granulate case, RCG option A is considered to be the best option when both interests are weighted equally or when more emphasis is laid on the societal interest. When the options are evaluated from the plant owners point of view, the steam turbine and generator is considered to be the best option. In the iron fuel combustion case, RCG option B is the most applicable option when the application is considered from both a societal and plant owner point of view.

8 Discussion and recommendations

This research has led to proposed RCG options for the tar-granulate and metal fuels combustion case. The conclusion that have been drawn are based on technical and economical performance indicators that are subjected to assumptions and uncertainties. The reader should be aware of these to understand to what extend the drawn conclusions remain valid.

The initial investment costs of the cycles depend on the cost price of components. Since these prices mainly have been gathered from literature, these are accompanied with a level of uncertainty. Thereby should be noticed that these costs exclude the costs of an air heat exchanger and potential boiler system. Depending on the application, these can substantially contribute to both the initial investment costs and the operation and maintenance costs.

The levelized cost of electricity (LCOE) is based on the electricity prices of the Dutch unbalanced market of the year 2022. This particular year was characterized by high price volatility and a high mean electricity price. Lower price volatility would reduce the potential economical and social benefits of the RCG-cycles, especially those that are accompanied with lower heat to electricity efficiencies. Whether the trend of an increased price volatility will hold for the future depends on a variety of different factors that are hard to predict. These uncertainties can be accessed by for example a Monte Carlo analysis of the electricity prices or by use of scenario based price predictions.

The economical potential of the RCG cycles is purely based on the Dutch electricity market. There might be other markets that are worth investigating where the RCG-cycle can present more social and economical benefits. These can be for example other types of electricity markets or real time markets in other area's. These do not necessarily have to be price driven markets when the cycle is applied to industries that have no or an unstable electricity grid in for example rural area's or maritime applications.

The outcome of the MCDMA analysis is somewhat subjective since no intensive analysis has been carried on the interests of all stakeholders. Since no research has been carried out about the interests of all stakeholders, the chosen weighting factors are poorly substantiated. Although the MCDMA analysis would suggest that a decision is made by weighting the interests of multiple stakeholders, one should be aware of the fact that the unbalanced market is a price driven market model. In essence there is only one decision maker, that is likely to focus on the economical performance. However, the outcome of the MCDMA can be used by policy makers to introduce a "carrot or stick" approach to alter the initial investment costs or the LCOE such that the RCG options can become more lucrative. These can include for example policy for carbon credits or fines for electricity use in particular period of time which are excluded within this research.

Lastly, questions can be raised about the organization of the current electricity market. One could argue that the privatization of the transmission system operators and the rather capitalistic price model is something that is beneficial for the society as a whole. While energy intensive industries overload the electricity grid without any consequence, the Dutch electricity grid is suffering and foreign shareholder returns remain positive. The technical challenges that arise by building the electricity grid of the future are challenging and ask a lot from the society. However, the elephant in the room continuous to be responsible use of electricity. This includes a change of behaviour for both the industry and households which can be considered as the real challenge within the energy transition.

A Automotive turbine efficiency estimation

A major influence on the efficiency of turbines can be assigned to isentropic losses due to the gap between the rotor and turbine housing. It is common practice to estimate the efficiency of radial turbines by use of turbine maps, displayed by the manufacturer. Often these maps aren't publicly available and only the maximum efficiency of the turbine is mentioned. Heat Power B.V. is in possession of one turbine map that displays the turbine efficiency of the K44 turbine for a given temperature at varying pressure ratio's.

Literature suggests a method to determine the efficiency curves of radial turbines by use of the blade speed ratio (BSR), the ratio between the linear velocity of the rotor tip (U) and the isentropic velocity through the turbine stage (C_s). It can be calculated by use of equation 34.

$$BSR = \frac{U}{C_s} \tag{34}$$

Where the linear velocity of the rotor tip (U) is a function of the rotational velocity of the turbine wheel (N) and the turbine radius (r_t), as can be noticed from equation 35.

$$U = 2 \cdot \pi \cdot r_t \cdot \frac{N}{60} \tag{35}$$

The isentropic velocity through the turbine stage is given by equation 36 and depends on the heat capacity (C_p), turbine inlet temperature (TIT), pressure ratio (r) and specific heat ratio (k).

$$C_s = \sqrt{2 \cdot TIT \cdot C_p \cdot (1 - \frac{1}{r}^{\frac{k-1}{k}})} \tag{36}$$

Literature suggests a relation between the maximum efficiency, actual BSR and optimum BSR that is given by the equation 37 . Due to the geometry of the vanes, the optimum blade speed ratio of radial turbines is equal to $\frac{1}{\sqrt{2}}$, assumed that there is negligible swirl at the turbine outlet.

$$n_t = n_{max}(\frac{2BSR}{BSR_{opt}} - (\frac{BSR}{BSR_{opt}})^2) \tag{37}$$

For the BorgWarner K44 turbine, the estimated efficiency - BSR plot is displayed in figure 50. By use of the BSR and optimum BSR, efficiency curves of the K44 turbine have been estimated for varying pressure ratio's and angular velocities at the specified turbine inlet temperature. These curves have been compared to the original efficiency map of the K44 and show few deviations.

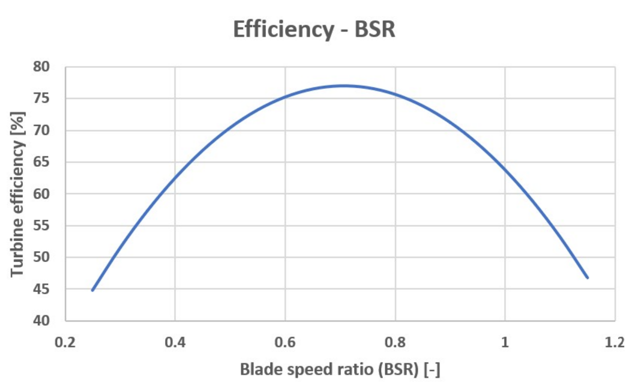


Figure 50: Efficiency - BSR plot of the BorgWarner K44 turbine

By use of this method, the efficiency of the turbine can be estimated for operating points that differ from the specified turbine inlet temperature. Beside that, the efficiency curves of other radial turbines can be estimated as long as the radius of the turbine wheel is known.

B Thermodynamic properties of cycles

In this appendix, the thermodynamic properties of RCG option A till D and the CHP plants are given. The location of the states in the rankine and air cycle correspond with figure 5. Calculations of the power and heat have been computed by use of the thermodynamic equations as described in chapter 2.

B.1 Rankine cycle

The thermodynamic properties of the open -and closed loop rankine cycles are displayed in table 21 and 22.

Nr	State	p [bara]	T [C]	h [kJ/kg]
1	Saturated liquid	1	100	417
2	Compressed liquid	29	100	421
3	Superheated vapor	29	400	3233
4	Liquid and vapour (VF = 0.9)	1	100	2492

Table 21: Thermodynamic properties of the closed-loop rankine cycle

Nr	State	p [bara]	T [C]	h [kJ/kg]
1	Saturated liquid	1	20	84
2	Compressed liquid	29	20	87
3	Superheated vapor	29	400	3233
4	Liquid and vapour	1.5	100	2693

Table 22: Thermodynamic properties of the open-loop rankine cycle

B.2 Air cycle

The thermodynamic properties of the air cycle for RCG configuration A till D are listed in table 23 till 26. For the calculation of the air cycle is assumed that there is no pressure loss between the outlet of the compressor and the inlet of the power turbine. The inlet pressure and temperature of the ambient air is assumed to be 1 bara at 300k and the air is considered to be an ideal gas, free of moist.

Nr	State	P [bara]	T [C]	Cp [kJ/kgk]	Cv [kJ/kgk]	k [-]
7	Ambient air	1	25	1	0.72	1.4
8	Compressed ambient air	2.62	139	1.02	0.73	1.39
9	Heated compressed ambient air	2.62	500	1.09	0.8	1.36
10	Heated ambient air	1	333	1.05	0.76	1.38

Table 23: Thermodynamic properties of the air cycle of RCG option A

Nr	State	P [bara]	T [C]	Cp [kJ/kgk]	Cv [kJ/kgk]	k [-]
7	Ambient air	1	25	1	0.72	1.4
8	Compressed ambient air	2.28	120	1.02	0.73	1.39
9	Heated compressed ambient air	2.28	1000	1.12	0.89	1.25
10	Heated ambient air	1	789	1.12	0.86	1.29

Table 24: Thermodynamic properties of the air cycle of RCG option B

Nr	State	P [bara]	T [C]	Cp [kJ/kgk]	Cv [kJ/kgk]	k [-]
7	Ambient air	1	25	1	0.72	1.4
8	Compressed ambient air	1.5	70	1.01	0.72	1.4
9	Heated compressed ambient air	1.5	500	1.09	0.80	1.36
10	Heated ambient air	1	426	1.07	0.79	1.37

Table 25: Thermodynamic properties of the air cycle of RCG option C

Nr	State	P [bara]	T [C]	Cp [kJ/kgk]	Cv [kJ/kgk]	k [-]
7	Ambient air	1	25	1	0.72	1.4
8	Compressed ambient air	3	157	1.02	0.73	1.4
9	Heated compressed ambient air	3	500	1.09	0.80	1.36
10	Heated ambient air	1	280	1.04	0.75	1.38

Table 26: Thermodynamic properties of the air cycle of RCG option D

B.3 Heat and power

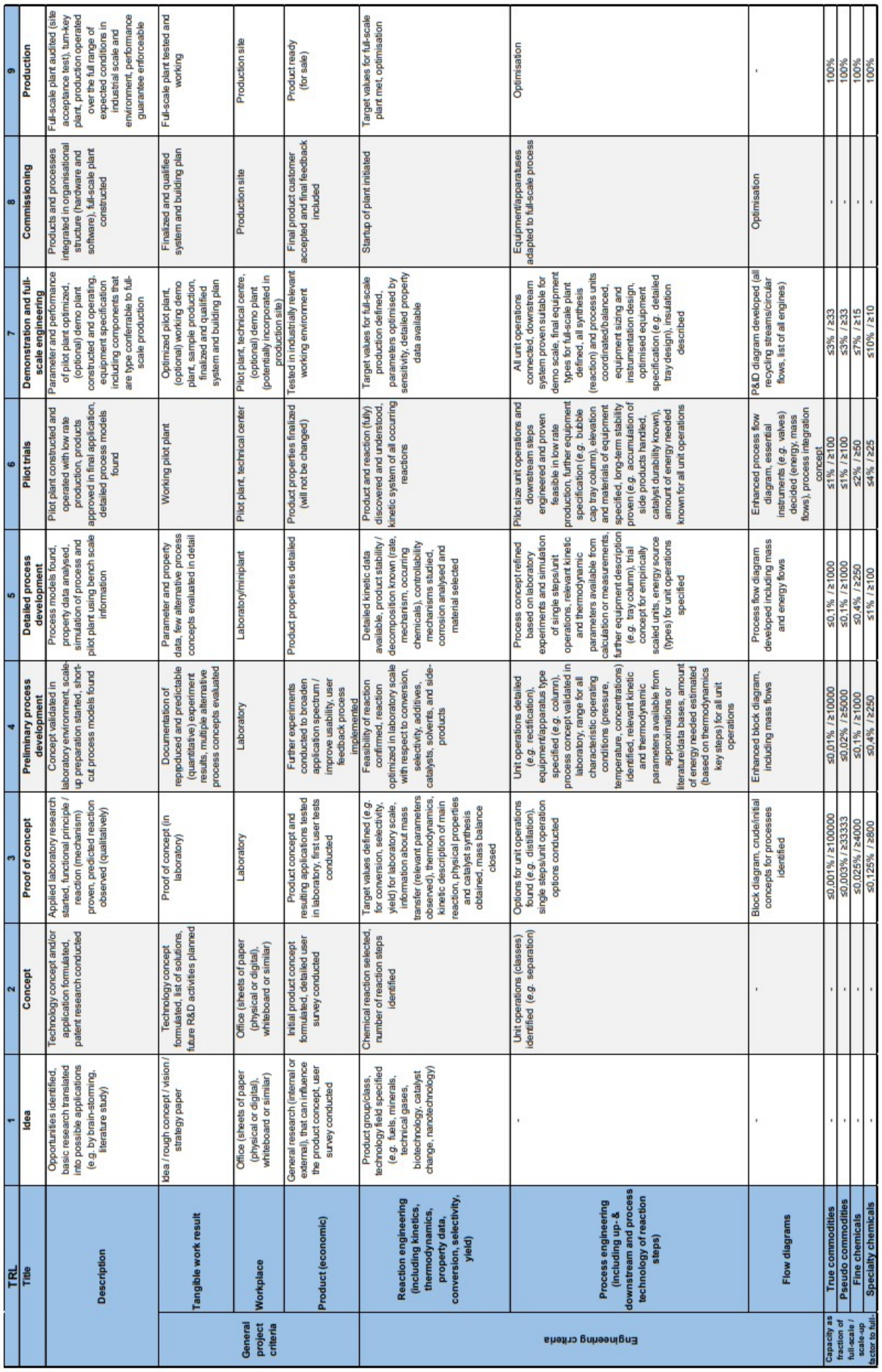
The heat and power properties of the RCG option A till D and the CHP option are listed in table 27.

	A	B	C	D	ST	Unit
$\dot{m_{fl}}$	1.25	0.72	0.94	1.14	1.49	kg/s
m_{air}	7.43	5.11	14.5	5.85		kg/s
W_{pump}	3.75	2.16	2.82	3.42	4.47	kW
P_{st}	833.78	480.54	630.88	763.7	1052.63	kW
n_{st}	0.7	0.7	0.7	0.7	0.7	
P_{comp}	833.78	480.54	630.88	763.7	0	kW
n_{comp}	0.85	0.85	0.85	0.85		[-]
P_pt	1052.63	1052.63	1052.63	1052.63	0	kW
n_{pt}	0.75	0.75	0.85	0.85		[-]
P_{gen}	1000	1000	1000	1000	1000	kWe
n_{gen}	0.95	0.95	0.95	0.95	0.95	[-]
Q_b	3528.75	2032.56	2653.62	3218.22	4206.27	kWth
Q_{cond}	2927	1685.95	2201.1	2669.42	3488.98	kWth
Q_{ahe}	2825.86	4787.25	6545.04	2100.03	0	kWth
Q_{gto}	2340.06	4193.47	6017.52	1512.77	0	kWth
Q_{st}	1191.11	686.49	901.25	1091.01	1503.76	kWth

Table 27: Heat and power properties and heat to energy efficiency of RCG option A till D and the CHP option

C Technology readiness levels

A description of the technology readiness levels (TRL) are presented in figure 51.



TRL		1	2	3	4	5	6	7	8	9
Title		Idea	Concept	Proof of concept	Preliminary process development	Detailed process development	Pilot trials	Demonstration and full-scale engineering	Commissioning	Production
Description		Opportunities identified, basic research translated into possible applications (e.g. by brain-storming, literature study)	Technology concept and/or application formulated, patent research conducted	Applied laboratory research started, functional principle / reaction (mechanism) proven, predicted reaction observed (qualitatively)	Concept validated in laboratory environment, scale-up preparation started, short-cut process models found	Process models found, property data analysed, simulation of process and pilot plant using bench scale information	Pilot plant constructed and operated with low rate production, products approved in final application, detailed process models found	Parameter and performance of pilot plant optimized, (optional) demo plant constructed and operating, equipment specification including components that are type conferrable to full-scale production	Products and processes integrated in organisational structure (hardware and software), full-scale plant constructed	Full-scale plant audited (site acceptance test), turn-key plant, production operated over the full range of expected conditions in industrial scale and environment, performance guarantee enforceable
General project criteria	Tangible work result	Idea / rough concept / vision / strategy paper	Technology concept formulated, list of solutions, future R&D activities planned	Proof of concept (in laboratory)	Documentation of reproduced and predictable (quantitative) experiment results, multiple alternative process concepts evaluated	Parameter and property data, few alternative process concepts evaluated in detail	Working pilot plant	Optimized pilot plant, (optional) working demo plant, sample production, finalized and qualified system and building plan	Finalized and qualified system and building plan	Full-scale plant tested and working
	Workplace	Office (sheets of paper (physical or digital), whiteboard or similar)	Office (sheets of paper (physical or digital), whiteboard or similar)	Laboratory	Laboratory	Laboratory/miniplant	Pilot plant, technical center	Pilot plant, technical centre, (optional) demo plant (potentially incorporated in production site)	Production site	Production site
	Product (economic)	General research (internal or external), that can influence the product concept, user survey conducted	Initial product concept formulated, detailed user survey conducted	Product concept and resulting applications tested in laboratory, first user tests conducted	Further experiments conducted to broaden application spectrum / improve usability, user feedback process implemented	Product properties detailed	Product properties finalized (will not be changed)	Tested in industrially relevant working environment	Final product customer accepted and final feedback included	Product ready (for sale)
Engineering criteria	Reaction engineering (including kinetics, thermodynamics, property data, conversion, selectivity, yield)	Product group/class, technology field specified (*e.g.* fuels, minerals, technical gases, biotechnology, catalyst change, nanotechnology)	Chemical reaction selected, number of reaction steps identified	Target values defined (*e.g.* for conversion, selectivity, yield) for laboratory scale, information about mass transfer (relevant parameters observed), thermodynamics, kinetic description of main reaction, physical properties and catalyst synthesis obtained, mass balance closed	Feasibility of reaction confirmed, reaction optimized in laboratory scale with respect to conversion, selectivity, additives, catalysts, solvents, and side-products	Detailed kinetic data available, product stability / decomposition known (rate, mechanism, occurring chemicals), controllability mechanisms studied, corrosion analysed and material selected	Product and reaction (fully) discovered and understood, kinetic system of all occurring reactions	Target values for full-scale production defined, parameters optimised by sensitivity, detailed property data available	Startup of plant initiated	Target values for full-scale plant met, optimisation
	Process engineering (including up- & downstream and process technology of reaction steps)	-	Unit operations (classes) identified (*e.g.* separation)	Options for unit operations found (*e.g.* distillation), single steps/unit operation options conducted	Unit operations detailed (*e.g.* rectification), equipment/apparatus type specified (*e.g.* column), process concept validated in laboratory, range for all characteristic operating conditions (pressure, temperature, concentrations) identified, relevant kinetic and thermodynamic parameters available from approximations or literature/data bases, amount of energy needed estimated (based on thermodynamics key steps) for all unit operations	Process concept refined based on laboratory experiments and simulation of single steps/unit operations, relevant kinetic and thermodynamic parameters available from calculation or measurements, further equipment description (*e.g.* tray column), trial concept for empirically scaled units, energy source (types) for unit operations specified	Pilot size unit operations and downstream steps engineered and proven feasible in low rate production, further equipment specification (*e.g.* bubble cap tray column), elevation and materials of equipment specified, long-term stability proven (*e.g.* accumulation of side products handled, catalyst durability known), amount of energy needed known for all unit operations	All unit operations connected, downstream system proven suitable for demo scale, final equipment types for full-scale plant defined, all synthesis (reaction) and process units coordinated/balanced, equipment sizing and instrumentation design, optimised equipment specification (*e.g.* detailed tray design), insulation described	Equipment/apparatuses adapted to full-scale process	Optimisation
	Flow diagrams	-	-	Block diagram, crude/initial concepts for processes identified	Enhanced block diagram, including mass flows	Process flow diagram developed including mass and energy flows	Enhanced process flow diagram, essential instruments (*e.g.* valves) decided (energy, mass flows), process integration concept	P&ID diagram developed (all recycling streams/circular flows, list of all engines)	Optimisation	-
Capacity as fraction of full-scale / scale-up factor to full-	True commodities	-	-	≤0,001% / ≥100000	≤0,01% / ≥10000	≤0,1% / ≥1000	≤1% / ≥100	≤3% / ≥33	-	100%
	Pseudo commodities	-	-	≤0,003% / ≥33333	≤0,02% / ≥5000	≤0,1% / ≥1000	≤1% / ≥100	≤3% / ≥33	-	100%
	Fine chemicals	-	-	≤0,025% / ≥4000	≤0,1% / ≥1000	≤0,4% / ≥250	≤2% / ≥50	≤7% / ≥15	-	100%
	Specialty chemicals	-	-	≤0,125% / ≥800	≤0,4% / ≥250	≤1% / ≥100	≤4% / ≥25	≤10% / ≥10	-	100%

Figure 51: Technology readiness level for chemical plants. Figure has been adopted from [5].

D Multi criteria decision making analysis criteria (MCDMA)

D.1 Weighting factors

The social, social-economical and economical weighting factors for the tar granulate case are displayed in table 28 and for the metal fuels case are displayed in table 29.

Categorie	Criteria	Social	Social - economic	economic
Financial	Installed costs	5	8.33	11.66
	LCOE (15y)	15	25	35
	PBP	5	8.33	11.66
Societal	Carbon emissions	0	0	0
Technical	Peak shaving capability	70	50	30
	Technical risk	5	8.33	11.66

Table 28: Weighting factors of for the tar-granulate case.

Categorie	Criteria	Social	Social - economic	economic
Financial	Installed costs	5	8.33	11.66
	LCOE (15y)	15	25	35
	PBP	5	8.33	11.66
Technical	Technical risk	5	8.33	11.66
Societal	Peak shaving capability	35	25	15
	Carbon emissions	35	25	15

Table 29: Weighting factors of for the metal fuels combustion case.

D.2 Normalized scores and results

The normalized scores of the tar-granulate case are displayed in table 30 and the normalized score of the iron fuel combustion case are displayed in table 31.

	Societal				Societal - plant owner				Plant owner			
	A	C	D	ST	A	C	D	ST	A	C	D	ST
Installed costs	0.03	0.02	0.02	0.05	0.05	0.04	0.04	0.08	0.07	0.05	0.05	0.12
LCOE (15y)	0.07	0.05	0.05	0.15	0.11	0.08	0.09	0.25	0.16	0.12	0.12	0.35
PBP [y]	0.03	0.02	0.02	0.05	0.05	0.03	0.03	0.08	0.07	0.05	0.05	0.12
Carbon polution	0	0	0	0	0	0	0	0	0	0	0	0
Peak shaving capability	0.7	0.7	0.7	0	0.5	0.5	0.5	0	0.3	0.3	0.3	0
Technical risk	0.02	0.02	0.02	0.05	0.04	0.04	0.04	0.08	0.05	0.05	0.05	0.12
SUM	0.85	0.81	0.81	0.3	0.75	0.69	0.7	0.49	0.65	0.57	0.57	0.71

Table 30: Normalized scores for a societal, societal-plant owner and plant owner perspective on the tar-granulate case

	Societal					Societal - plant owner					Plant owner				
	A	B	C	D	ST	A	B	C	D	ST	A	B	C	D	ST
Installed costs	0.03	0.03	0.02	0.02	0.05	0.05	0.06	0.04	0.04	0.08	0.07	0.08	0.05	0.05	0.12
LCOE (15y)	0.12	0.15	0.13	0.12	0.15	0.2	0.25	0.22	0.19	0.25	0.29	0.34	0.31	0.27	0.35
PBP [y]	0.04	0.03	0.03	0.03	0.05	0.07	0.06	0.05	0.05	0.08	0.09	0.08	0.07	0.06	0.12
Carbon pol.	0.27	0.35	0.31	0.27	0.3	0.19	0.25	0.22	0.19	0.21	0.11	0.15	0.13	0.11	0.13
Peak shav.	0.35	0.35	0.35	0.35	0	0.25	0.25	0.25	0.25	0	0.15	0.15	0.15	0.15	0
Technical risk	0.03	0.03	0.03	0.03	0.05	0.05	0.05	0.05	0.05	0.08	0.06	0.06	0.06	0.06	0.12
SUM	0.84	0.94	0.88	0.81	0.6	0.81	0.91	0.83	0.76	0.71	0.78	0.87	0.78	0.72	0.83

Table 31: Normalized scores for a societal, societal-plant owner and plant owner perspective on the metal fuels combustion case

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