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MASTER'S THESIS

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Small Scale Combined Heat and Power Units Using External Combustion

Abstract

Combined heat and power plants are of increasing interest due to the rising concern over global warming as they can lower emissions, by having very higher efficiencies than traditional power plants. Implemented on a small scale units and with external combustion they allow for great flexibility in implementation and fuel, and can allow for remote locations to serve their own heat and power needs.

This thesis investigates in the first part the available technologies for such plants on a small scale and compares them on criteria which are important for efficiency, economy, implementation and operation in a remote area. Organic- and steam Rankine cycles are evaluated as well as gas turbines, Stirling engines and thermoelectric generators. The first part concludes with the choice of gas turbines as the best technology for small scale power and heat generation based on defined criteria.

In the second different concepts for the basic layout and components of such a plant is evaluated. A commercially available gas turbine is chosen as a base and a solution where the gas turbine feeds its outlet air into the combustion chamber is selected. Two alternative layouts are simulated using the open source DNA, and compared along with a base case. While both layouts compared have advantages and drawbacks, only one has an efficiency comparable to larger plants, and should therefore the preferred concept under most circumstances.

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1 Abbreviations and symbols

Abbreviations

CHP - Co-generation of Heat and Power
CU – University of Colorado
GT - Gas Turbine
HMIS - Hazardous Materials Identification System
HX - Heat Exchanger
IRIS - International Research Institute of Stavanger
LHV - Lower Heating Value
OMTS - Octamethyltisiloxane
ORC - Organic Rankine Cycle
RAMS - Reliability, Availability, Maintainability, Supportability
SRC - Steam Rankine Cycle
TEG - Thermoelectric Generator
TiT - Turbine inlet temperature
U.S. EPA – United States Environmental Protection Agency

Symbols

Note – for properties a small letter indicates specific values, e.g. per unit mass or volume.

ϵ	Effectiveness
α	Seebeck coefficient
$\dot{m}$	Mass flow
η	Efficiency
A	Area
c	Specific Heat
h	Enthalpy
k	Thermal conductivity
p	Pressure
q	Heat
s	Entropy
T	Temperature
U	Heat transfer coefficient
V	Volume
w	Work
Z	Thermoelectric figure of merit

2 Preface

This thesis was written by Pål André Johansen as part of achieving a masters degree in sustainable energy technology at the University of Stavanger. The thesis was written on initative and with the support of Peter Breuhaus of IRIS, whom I'd like to thank especially for his time and help. I'd also like to thank my instructor at UiS, Bjørn Hjertager for his support. Finally thanks goes to my parents for housing and feeding me with great tolerance as I worked on this thesis.

This thesis was written with OpenOffice writer and LibreOffice writer. Illustrations made with Inkscape and OpenOffice Draw, with additional manipulation done with GIMP. All thermodynamical simulations done with DNA.

3 Introduction

This thesis was done on assignment from Peter Breuhaus at the International Research Institute of Stavanger (IRIS). It started its life as project to create a CHP system to be integrated with a machine making pellets from bales developed at a local agricultural corporation. While the corporation found other research sources, the idea of small scale CHP plants for large farms and similar users remained an interesting one. As the world tries to reduce its emissions of CO_2, localized production of heat and power will likely be an important part of this effort.

The objective of this thesis has been to investigate the present and near future technologies for such applications, and to look further into how such an implementation would be done using the best available technology.

The thesis is therefore divided into two parts. The first parts looks into the relevant technologies, and is divided into three well known thermodynamical power cycles and their different applications. In addition thermoelectric generators are briefly looked into as a possible technology for CHP plants. At the end of the first part the technologies are evaluated relative to each other based on the criteria of the technologies performance in areas deemed important for the application.

The second part of the thesis investigates more closely the application of the technology deemed most fit. An open source thermodynamic simulation tool, DNA, has been used to closer investigate and predict the technology's performance, and evaluate different alternatives in application.

Part One

4 CHP: Co-generation of Heat and Power

CHP stands in contrast to traditional power generation by recovering the otherwise wasted heat and applying it to heating, chemical processes or other uses. This means more of the energy from the power source, normally the combustion of hydrocarbons, is used. We say that the efficiency, the useful energy extracted from the process divided by the energy put into it, rises.

For the recovered heat to be of use however, the consumer needs to be close to the source of the heat, as transporting heat over large distances is very inefficient and costly. Traditional power generation in developed countries take place at large central power stations, usually a distance away from population centers. This is either because of the large amount of pollutants created, or because of fears over radiation in the case of nuclear power plants. While centralized CHP is gaining in popularity as fossil fuel power plants become cleaner, it is in decentralized power production that CHP really shines.

Decentralized power production on its own has many advantages. It reduces electrical grid losses, and makes the grid itself less vulnerable to failures caused by power lines going down. Where locally produced fuel is used it reduces the pollution and extra costs related to transport of fuel. Taken together with the efficiency increases enabled by the use of CHP this makes small scale decentralized power production a viable source of energy.

Traditionally however, relatively low energy prices coupled the high investment costs and difficulties with grid interconnection has made decentralized power production unattractive. This is changing as consumers and regulators are responding to environmental concerns, largely driven by the looming threat of global warming. The different technologies available for small scale CHP are also becoming cheaper and more efficient, and several new technologies are reaching maturity.

System overview

Any CHP system will need a power source. With rare exceptions this will be a heat source of some kind. The heat source can be direct combustion, but other examples can be excess heat from industrial processes, heat from the sun or from decay of radioactive materials. In most CHP systems this heat will be used to produce power first, before remaining heat is exported. This ensures that power is produced when the temperature of the heat carrier is highest, ensuring higher electrical efficiency of the cycle. A simple diagram of the energy flows is shown in illustration 1.

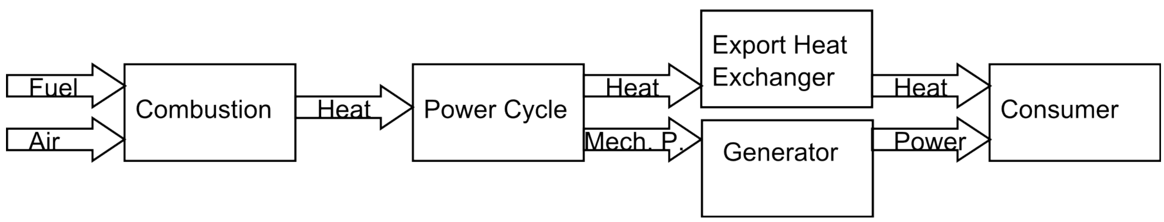

Illustration 1: The primary energy flows in an CHP System

The heat is converted to power usually by the intermediate generation of mechanical energy. An example of this would be heat used to boil water, the steam produced then used to turn a turbine, generating mechanical power. This mechanical power could then be used to drive a generator, producing electricity. Some technologies work differently, such as thermoelectric cells, which generate power directly from a temperature difference between the two sides of the cell.

After power is produced remaining heat can be extracted in a heat exchanger for heating or cooling purposes. In the case of combustion for heat generation, excess heat from the flue gases can also be extracted to further improve the efficiency of the process.

Heat

Depending on the planned use, the heat that is produced can be of different temperatures. As heat is extracted the temperature sinks, and the lower the temperature, the more heat can be extracted. Thus the lower the temperature, the more efficient the cycle is, as the heat not extracted is wasted. The drawback is that the lower the temperature the less uses it has. If the heat is used for some chemical process, for example, the temperature needed for this might be very high, decreasing the efficiency of the whole cycle. For domestic heating applications usually a temperature of 70 – 100ºC is needed.

Power

Electricity is clearly the most useful product of the CHP process. In the event that it is not needed at the time of production it can either be sold or stored. In any event the electricity needs to be of a standard that it can be used. For power generation when selling to the grid, the quality of the electricity is usually tightly regulated, primarily in terms of frequency and voltage, and the range of allowable variations of these properties.

While selling excess power to the grid has earlier been difficult, the increasing popularity of solar photovoltaic panels and small windmills is driving regulations in many countries to make this easier. Power can also be stored, and for small scale long term storage, this is usually in large banks of batteries. Power storage is however often very costly and cumbersome.

While a decentralized CHP plant can be run continually on full power, it is usually either driven to fulfill the heat or power need at any given time. Running the power plant to fulfill the heat needs of its consumers is usually the most efficient way, as excess electricity can usually be sold, while excess heat is usually wasted. Because running the plant on as close to rated power is most efficient, and heat needs vary with the time of year, storing excess heat can be an attractive way of increasing the efficiency of a plant. Shown in chart 1 is a simplified graph showing how production of heat can be held steady while using heat storage to ensure that heat is available as consumption varies throughout the year.

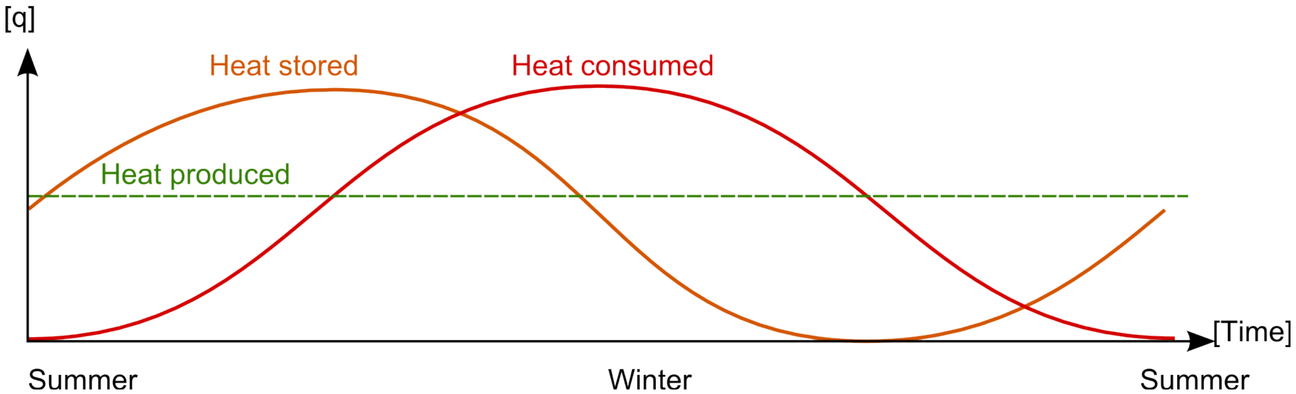

Chart 1: Simplified graph of storage and heat consumed through the year, with constant amount of heat produced

In the case that the CHP plant is the main source of heat it also allows the plant to be smaller, as when heat consumption peeks some of the heat can be taken from storage. Storing heat for such long periods of time as is needed to compensate for yearly fluctuations is called seasonal storage. Seasonal storage usually takes place by heating up large volumes of ground or rock.

Application of the plant

A CHP plant can find many applications in areas such as industry, public buildings and even private homes. Little is predefined about the application of our system, however, and as such we take a general approach to the technologies. The assumption has been made that the application is placed so that the possibility for sale of excess electricity production is available and that the heat needed is for heating buildings.

5 Reliability, Availability, Maintainability and Supportability (RAMS)

Reliability, availability, maintainability and supportability, collectively known as RAMS are important factors for all technological solutions. In this text availability, maintainability and supportability is understood as the factors making up the reliability of a solution, shown in illustration 2.

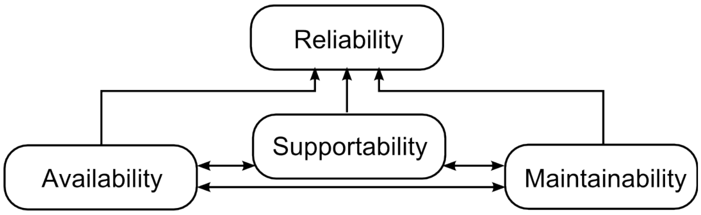

Illustration 2: Availability, supportability and maintainability interact, and make up the reliability of an system

Availability for a continuously running system is the average up time over the total time for some time average, usually a year. It is influenced by how often the system needs to shut down for preventative or corrective maintenance, and how long these tasks take. The time consumed in these tasks are effected by the maintainability and supportability of the system

Maintainability is the degree to which the system is easy to perform preventative and corrective maintenance on. This touches on supportability in the sense that it defines how much expertise is needed to maintain a repair the system. High maintainability should also mean that the system is quick to do maintenance on, as components should be easy to physically access and replace.

Supportability is the ease and cost of getting support from outside sources when the it is needed, foe example if the system fails or is showing signs of going to fail. Often support is also needed for major preventative maintenance activities, such as regular overhauls. As our system is meant to be in a somewhat remote location, supportability becomes more difficult, and important, as we can not expect the type of needed expertise to be locally available. This might lead to higher costs and more time consumed when outside resources are needed. The degree to which spare parts are easy and cheap to acquire, and how easy they are to store also plays a part in supportability.

6 Efficiencies

Electrical Cycle Efficiency

Electrical efficiency can be defined many ways, and normally for a power cycle would be defined as the electrical power out divided by the heat in. Heat in could then be defined as the amount of heat being supplied from the combustion of fuel, simply the lower heating value(LHV) multiplied with the mass flow of the fuel. All efficiencies where relevant use the lower heating value unless otherwise noted.

$$\eta_{el} = \frac{w_{\mathrm{el,net}}}{LHV \times \dot{m}_{fuel}}$$

However, as the combustion process is not specified, electrical efficiency for the different cycles becomes more important. The cycle electrical efficiency is defined as the amount of electricity produced divided by the heat input to the cycle.

$$\eta_{\mathrm{el,cycle}} = \frac{w_{\mathrm{el,net}}}{q_{\mathrm{in,cycle}}}$$

The Total Efficiency and the Power to Heat Ratio

Since a CHP plant for our application will most likely be driven by the heat need, heat may be looked at as the primary product of the cycle. Since we are looking only at the power cycles and not at the combustion or the connected heat exchange into the cycle, a possibility could be to look at the total cycle efficiency, defined as:

$$\eta_{\mathrm{total,cycle}} = \frac{w_{\mathrm{el,net}} + q_{\mathrm{export,\ cycle}}}{q_{\mathrm{in,\ cycle}}}$$

Note the additional subscript cycle to the amount of heat exported. This is added since in a full view of a CHP system there might be other sources of heat for export used, such as the flue gas. Because our need is for heating, the temperature required is low, so most technologies will have very high total efficiencies.

Power, however can always be converted to heat with a close to 100% efficiency, and since it can be used for other applications, as well as sold, it is the preferred product of the CHP process. By using a power to heat ratio rather than total efficiency, we reflect that fact. The power to heat ratio is defined as:

$$C_{CHP} = \frac{w_{\mathrm{el,net}}}{q_{\mathrm{export,cycle}}}$$

The Carnot Efficiency

The Carnot efficiency is the highest efficiency that can be achieved by a thermal cycle working between two temperatures. It is the efficiency of a totally reversible heat engine and is independent of the working fluid in the cycle and how the cycle is executed. As such, the Carnot efficiency is an important value we can measure that real processes against.

Given a reversible adiabatic engine working between a high temperature heat source Q_{in} and a low temperature one Q_{out} and doing work $W_{out,}$ the general efficiency is :

$$\eta_{cycle} = \frac{w_{\text{out}}}{q_{\text{in}}}$$

Since the process is adiabatic, that means the work done most be equal to difference between the heat flows:

$$w_{\text{out}} = q_{\text{in}} - q_{\text{out}} \Rightarrow \eta_{cycle} = \frac{q_{\text{in}} - q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}}$$

And since the process is reversible the change in entropy must be zero:

$$\Delta s = s_{\text{out}} - s_{\text{in}} = 0 \Rightarrow s_{\text{in}} = s_{\text{out}} = \frac{q_{\text{in}}}{T_{\text{in}}} = \frac{q_{\text{out}}}{T_{\text{out}}} \Rightarrow \frac{q_{\text{out}}}{q_{\text{in}}} = \frac{T_{\text{out}}}{T_{\text{in}}}$$

$$\Rightarrow \eta_{\text{Carnot}} = 1 - \frac{T_{\text{out}}}{T_{\text{in}}} = 1 - \frac{T_{\text{low}}}{T_{\text{high}}}$$

Thus the maximum achievable efficiency of any power cycle working between two temperatures is only function of those two temperatures.

7 Heat Exchangers

Heat exchangers are a key part of all power cycles with external combustion. In addition the remaining heat from the power cycles should be transferred to a heat export system. The heat exchanger to the heating export system is the export heat exchanger. Since heat exchangers play such a major role in an CHP plant, a short discussion of their workings is needed.

Simply put a heat exchanger moves heat between two mediums. For the heat transfer to take place a temperature difference between the mediums is needed, and the medium that enters the heat exchanger with the lower temperature is called the cold medium, and the other the hot medium. In some heat exchangers this is done by alternately running the fluids through the exchanger and letting them heat and cool the mass of the exchanger.

For our use the fluids in an heat exchanger are usually physically separate; they might have different pressures, or mixing them might not be wanted. Such heat exchangers can be divided into groups by the directions of the hot and cold flows, and most usual are parallel-, counter and cross-flow, shown in illustration 3.

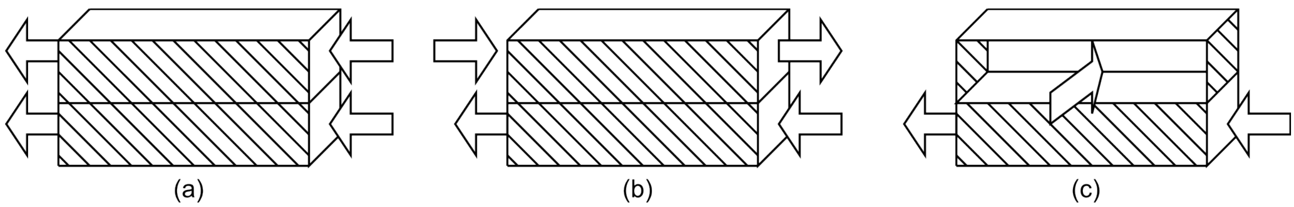

Illustration 3: The tree main flow designs of heat exchangers. (a) Parallel flow, (b) counterflow and (c) crossflow

A large number of different heat exchanger designs exist, with designs being optimized for size, effectiveness, cost, pressures or any number of variables. For example in gas-liquid heat exchangers where the heat transfer coefficients of the fluids are very different, finned-tube designs are often used. Shown in illustration 4, such a design increases the area for heat transfer for the gas relative to that available for the fluid. This keeps the temperature of the tubes down, as the same amount of heat can be transferred to or from the gas at a lower temperature.

When high effectiveness is needed in gas-gas heat exchangers, a plate-fin design, shown in illustration 5, is used. Plate-fin exchangers consists of layered plates, often either with rectangular or triangular cross sections with flat plates between them to keep the fluids apart. When classified by their heat transfer area per volume, both finned-tube and plate-fin designs usually fall under the definition of compact heat exchangers, meaning they have a surface area density of above 600 m^2/m^3.

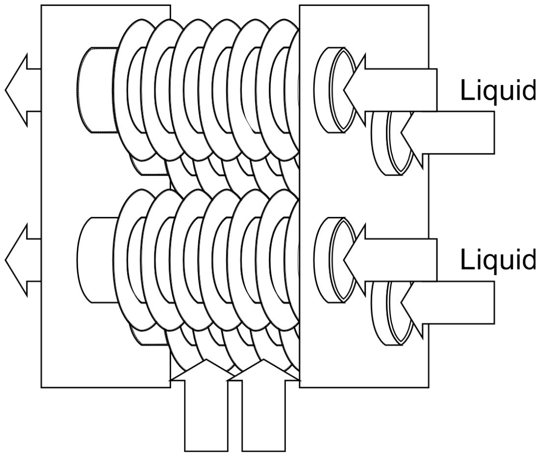

Illustration 4: Crossflow finned tube design heat exchanger

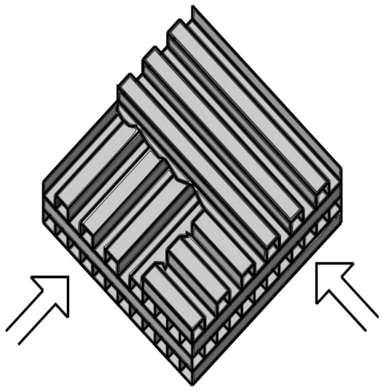

Illustration 5: Crossflow plate-fin design heat exchanger

A simplified way of calculating the properties of an heat exchanger can be deduced from the standard formula for heat transfer. The heat transferred per mass unit in a heat exchanger is $q= U*A*\Delta T_{lm}$,with U being the overall heat transfer coefficient, being dependent on the materials and design of the HX, as well as the properties of the fluid and the nature of its flow. A is the surface area over which the heat transfer takes place, and ΔT_{lm} is the logarithmic mean temperature difference. Where a ΔT_1 and ΔT_2 the difference between the temperatures of either side, the logarithmic mean temperature difference is:

$$\Delta T_{\ln} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\Delta T_2 / \Delta T_1\right)}$$

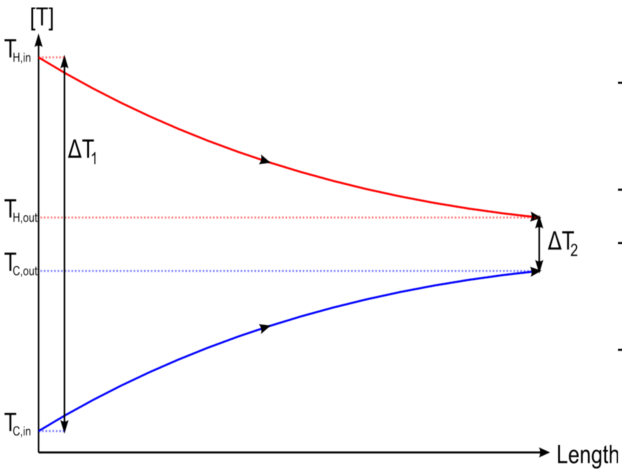

Illustration 6: Temperature as a function of length in a parallel flow heat exchanger

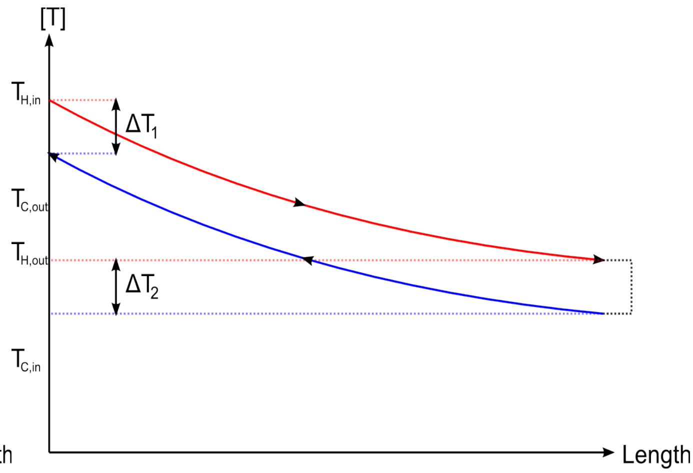

Illustration 7: Temperature as a function of length in a counterflow heat exchanger

An obvious advantage of counterflow heat exchangers, as seen in illustration 7 is that they allow for the outlet temperature on the cold side to be higher than the outlet temperature on the hot side. Seen in illustration 6, this is not the case for the parallel flow. Another advantage of counterflow is that for given temperatures these heat exchangers are more space efficient.

For a simple model, heat exchangers are characterized by their effectiveness, ϵ, as well as the relative pressure drop over the hot and cold side. The effectiveness is defined as the recovered energy by the actual heat exchanger divided by the maximum heat available for transfer. The largest temperature difference to be gained is between the hot and cold inlet, so q_{max} becomes $q_{max} = c_{min}(T_{H,in} - T_{C,in})$, with c_{min} being the heat capacity that is lowest of the interacting fluids. Depending on whether the hot or cold fluid has the lowest specific heat capacity, the effectiveness of a heat exchanger is then:

$$\epsilon \equiv \frac{q}{q_{max}} = \frac{c_H(T_{\text{H,in}} - T_{\text{H,out}})}{c_{min}(T_{\text{H,in}} - T_{\text{C,in}})} \quad \text{or} \quad = \frac{c_C(T_{\text{C,out}} - T_{\text{C,in}})}{c_{min}(T_{\text{H,in}} - T_{\text{C,in}})}$$

The information on heat exchanger is largely gathered from [Incropera et al., 2007] and [Lee, 2010].

8 Gas Turbine

Cycle

In a gas turbine the fluid remains in the gas phase throughout the cycle, giving the process its name. The ideal gas turbine cycle is the Brayton cycle, also sometimes known as the Joule cycle. The cycle starts with isotropic compression before heat is added at constant pressure and is then isotropically expanded, creating power. Heat is usually added in an internal combustion chamber, although other means of adding heat works just as well. In the case of external combustion, the air in the cycle will be routed through a heat exchanger called the gas heater.

The ideal cycle is shown in illustration 8 in a T-s diagram. The cycle illustrated in the T – s diagram is a closed one, meaning that the medium keeps circulating. While it is possible to use a closed cycle, usually air is constantly drawn in from the outside, and the exhaust gasses are vented likewise, removing the need for cooling. The system sketch shown in illustration 9 is of an open cycle. Gas turbines on larger scales will have several stages of connected compressors and turbines. Inter coolers between compressors or reheating between turbines can also be found in the most efficient large scale systems. Small scale gas turbines are usually single stage however, prohibiting the use of intercooling or reheating. A recuperator however is usually cost effective even for small systems, and is a standard component for most gas turbines.

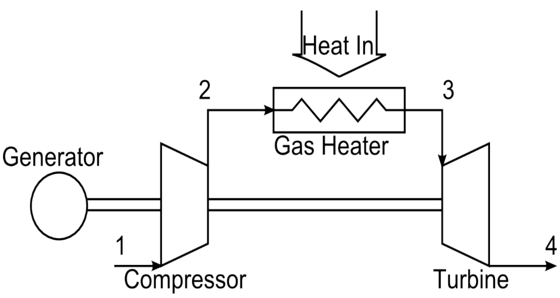

Illustration 9: System sketch for a gas turbine

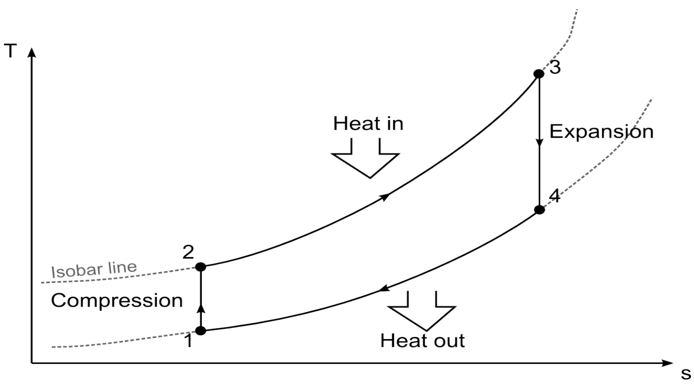

Illustration 8: The ideal Brayton cycle in a T- s diagram

Recuperator

Adding an internal heat exchanger before the gas heater, to recover some of the heat exiting the turbine is a common way to increase the efficiency of the cycle. This heat exchanger, known as a recuperator is shown integrated into the system in illustration 11. As some of the heat that would otherwise exit the cycle is recuperated, the amount of heat that is added to the cycle is reduced.

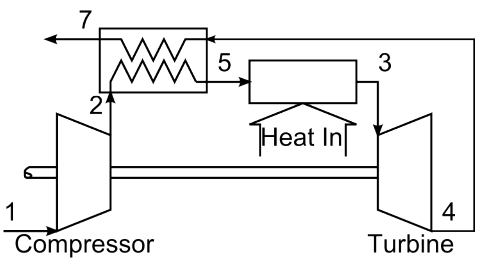

Illustration 11: System sketch for a gas turbine with recuperator

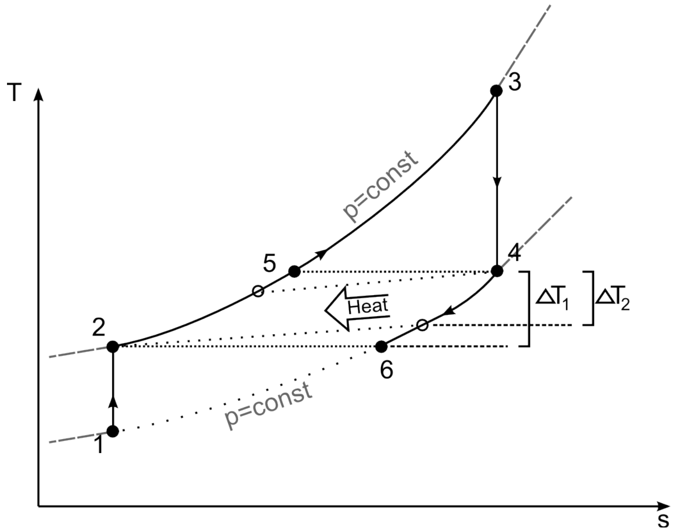

Illustration 10: Ideal and real recuperator in a Brayton cycle in the T - s dimensions

In illustration 10 the heat recuperation is shown both in the case of an ideal and real heat exchanger for an ideal process. In both cases the temperature difference from entering and exiting the recuperator is equal, as the specific heat capacity of the fluid does not change, and the mass flow is the same. The real heat exchanger is illustrated with the dotted line 2-6 and 5-4. As temperature difference is needed for the heat to transfer, the temperature gain ΔT_2 is less than the ΔT_1 in the ideal version.

Turbine and Compressor

Small scale GT are usually equipped with a centrifugal compressor and centrifugal turbine. For low volume flows centrifugal turbomachines are preferred as they are more compact and have higher pressure ratios per stage.

While in the ideal Brayton cycle both compression and expansion is done isentropically, in the real process, irreversiabilities occur. Both compressors and turbines are defined by the deviation of the actual performance from the isentropic ideal. This isentropic efficiency is defined as:

$$\eta_{is,C} = \frac{h_{2s} - h_1}{h_{2a} - h_1} \qquad \eta_{is,T} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

The effect of these irreversiabilities are shown in the T-s dimensions in illustration 12.

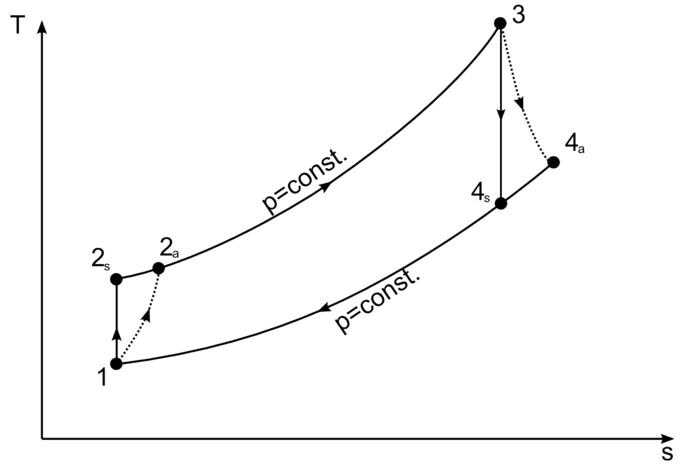

Illustration 12: Actual compression and expansion is denoted by a.

Gas heater

In a gas turbine with internal combustion the pressure drop during the heat addition is normally small. As we need to route the gas through a gas heater, the pressure drop will be larger, and thus have more of an impact than would be the case with internal combustion. This is shown in illustration 13, where the pressure drop in the recuperator is also illustrated. The light gray lines represent constant pressure.

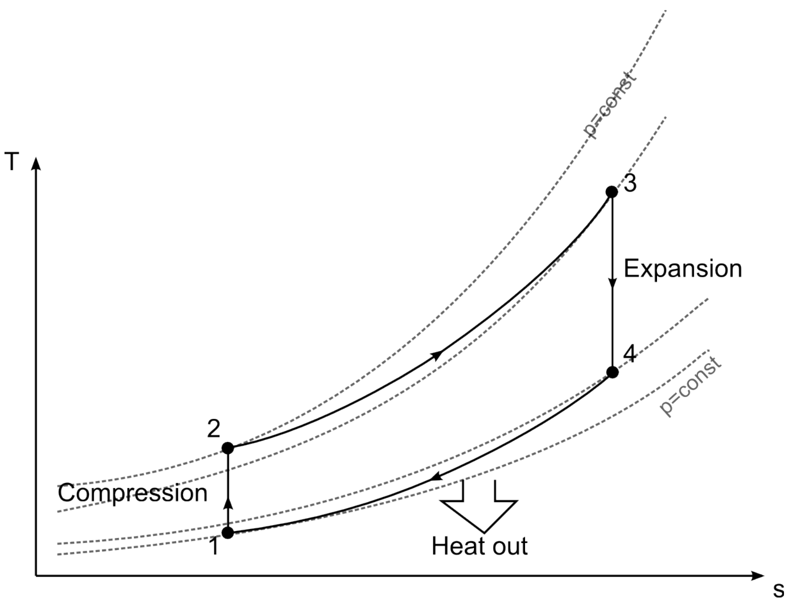

Illustration 13: Pressure drop over gas heater and recuperator illustrated in a T - s diagram

Closed cycle

For use with external combustion a closed cycle might be considered. In theory the closed cycle has many advantages over the open cycle for this application. Among the most important would be better efficiencies under part load, less wear on turbine blades, and, depending on the working fluid, smaller heat exchangers [Saravanamuttoo et. al., 2009]. The main drawbacks using a closed cycle compared to an open one would be the need for extra cooling with connected pressure losses in the heat exchanger, as well as the need for pressurization of the closed cycle. This leads to a more expensive, complicated and bulky power unit than in the case of the open cycle. In practice only a few closed cycle gas turbines have been designed and built.

RAMS

A problem with GT's is that the technology itself is less well known and understood by the average person. This means that even simple problems in running it might require outside expertise. Any faults with the GT itself will basically be unrepairable without professional support in our scenario. While spare parts will probably be available, this will be from the manufacturer, and not locally.

While reliability data for microturbines in the 20 – 30 kW scale are rarely published and available, values for the slightly larger Turbec T100 are available. The manufacturer for that GT claims more than 60,000 hours life time for all parts except the combustion chamber, which isn’t used in our application anyway [Turbec, 2013]. Inspection should be for every 3000 hours and overhaul at 30,000 hours. Time between overhauls should be longer in an application without the combustion chamber. The manufacturer estimates 24 hours of outage for inspection, and the same for overhaul. While our application is about one third of the size of the T100, the numbers should be in the same range.

Efficiency

Gas turbines in general have relatively high electrical cycle efficiencies, often in the range of 35 – 45 % for large turbines and 20 - 30% for micro turbines (<250kW$_e$). Taking the previously mentioned T100, a 30% electrical efficiency is cited by its producer, while the Capstone C30, a smaller, 30kW$_e$ gas turbine is cited a having a 25% electrical cycle efficiency [Capstone, 2010]. These efficiencies are however predicated on the heat source being internal combustion of hydrocarbons. As mentioned the gas heater will involve additional pressure losses. Both of these will negatively impact a GT's efficiency.

9 Rankine Cycle

The Rankine cycle is based on the phase change of the working fluid from liquid to gas and back. The produced steam is then expanded trough an expander to create mechanical energy. While usually some part of the steam will by design condense in the expander, the steam will completely return to liquid in the condenser before entering the pump again. A system sketch of an rankine cycle plant is shown in illustration 14.

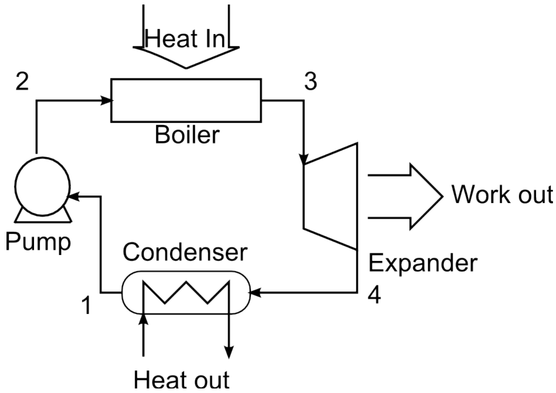

Illustration 14: System sketch for a gas turbine

Shown below in illustration 15 and 16 is the ideal Rankine cycle for water and an organic fluid in the T – s dimensions. The phase envelope (saturation vapor curve) is of the fluid being illustrated with dashed lines. The cycle starts at point one, before the liquid is pumped up to the boiler pressure. The height of the line 1 – 2, i.e. the pressure differential over the pump, is greatly exaggerated for the purposes of illustration. The boiler heats the water until it is superheated steam at point 3. For cycles with other mediums than water, superheating is often not needed, depending on the fluid used, and the gas is expanded directly from the dew point. The vapor is then expanded to the condensation pressure. Shown in illustration 16 is an organic fluid that, because of the shape of the phase envelope, starts as saturated vapor, and enters the superheated area as it cools down. The fluid then goes through the condenser before it reenters the pump as saturated liquid.

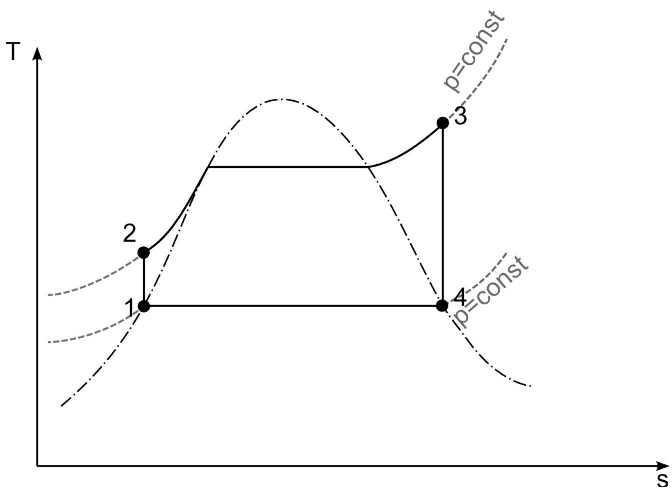

Illustration 15: The Rankine cycle in an T-s diagram with water as the working fluid

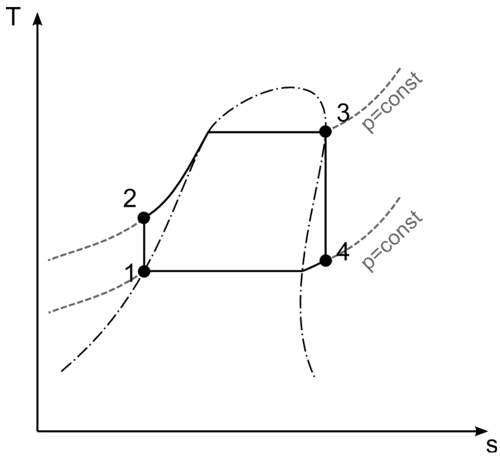

Illustration 16: The Rankine cycle in an T-s diagram with an organic liquid with a positive vapor curve as the working fluid

The T-s diagram can also be used for looking at the factors affecting the efficiency of the cycle. A useful property of T – s diagrams is that the area enclosed by the curves of the reversible processes equals the specific net work done by the system.

Since condensation happens isothermally the condensation pressure is mainly dictated by the temperature of the cooling source. Lowering the condensation temperature will also decrease the pressure in the condensator, increasing the necessary pump work, and the heat needed to reach the desired temperature in the boiler. This is however offset by the gain in work done by the expander, increasing the efficiency of the cycle. The temperature available for cooling is thus a major factor in the efficiency of the cycle.

Superheating the vapor after boiling is common practice when water is the working fluid. It allows for extraction of more work for set vapor content and temperature out of the expander. This increases the heat needed as well as the maximum and average temperatures. On the other hand the gains in output from the expander outweighs this and the efficiency of the cycle increases. For cycles with other working fluids super heating of the vapor can lead to drops in efficiency, as the gain in work does not offset the extra heat input in those cases. [Drescher and Brüggemann, 2007]

Another aspect of the cycle worth looking closer at is the boiler pressure. Keeping the maximum temperature of the cycle steady, increasing the pressure means less heat is needed to reach that temperature. This has the backside of requiring more work done by the pump, but the total effect is an increase in efficiency of the cycle. Increasing the boiler pressure might also mean that more of the heat addition happens in the superheated gas phase. This results in larger and more costly heat exchangers, as the gas will have a lower heat transfer coefficient than the liquid.

Working fluid

The most common working fluid in the Rankine cycle is water. This is also known as the water-steam cycle or steam Rankine cycle (SRC). The alternative is the organic Rankine cycle (ORC). In ORC the working fluid is an organic component with different thermodynamic properties than steam.

The main advantage of these organic fluids are lower boiling temperatures, meaning that an ORC can function on lower temperatures than SRC. As given by the Carnot efficiency, the maximum efficiency decreases with a lower T_H for a given T_L. While the real efficiency will always be lower than the Carnot efficiency, the effect is that real efficiency also drops with lower source temperatures. While negatively affecting the cycles efficiency, lower temperatures in the boiler means lower boiler pressures, thus making low temperature ORC power units cheaper and smaller. This also leads SRC to be a viable technology for generating electricity out of low temperature waste heat.

In addition, the phase envelope of the organic fluids have a different shape than water, as seen in illustrations 15 and 16. This property; either positive, as is illustrated, or isentropic vapor curve, means that superheating is not necessary to avoid moisture in the expander. This again leading to lower temperatures being possible, as well as allowing for the use of an recuperator. [Quoilin and Lemort, 2009] Many different fluids are available for ORC, with the possibility of selecting the working fluid most suited for the process in question. Typical fluids are refrigerants or light hydrocarbons. For example the refrigerant R245fa is often used, and for biomass applications Octamethyltisiloxane (OMTS) being most common.

The downside of using other fluids than water in an Rankine cycle is the potential negative safety aspects. The Hazardous Materials Identification System (HMIS) is commonly used to rate the risk aspects of a fluid with regards to health hazard, flammability and physical hazard (reactivity) on a scale from 0 - 4. In the HMIS system R245fa is rated with a 2 in terms of health hazard and a 1 with regards to fire hazard. OMTS is not rated as a health hazard (score 0) but is highly inflammable and a low (score 1) physical hazard. The safety risks of the fluids used in ORC means that precautions needs to taken when handling and storing the fluid, and that extra efforts need to be implemented to avoid leakages from such a system. In addition, the these fluids are often costly, leading to the use of intermediate cycles between the flue gasses and the cycle. Since fluid-fluid heat exchangers are more space efficient, an intermediate cycle containing thermal oil can thus lessen the amount of ORC fluid needed.

Another downside is that most fluids have a lower specific heat capacity than water, leading to higher mass flow in the cycle. This means that bigger pumps are needed, negatively affecting the efficiency of the cycle. [Vankeirsbilck et al., 2011]

As with the Brayton cycle, Rankine cycles might use several stages, and re-heating or intercooling, but for our purposes, those will not be considered on the background of cost.

Boiler

As the working fluid in a steam cycle goes through two phases during the heat addition part, the heat addition process is more complex than the other cycles. While a gas heater, for example, mainly consists of a single gas to gas heat exchanger, q boiler is normally divided into three parts; economizer, evaporator and superheater. This means that a boiler consists of three different heat exchangers, adding heat to the medium in the different phases of the fluid, as shown in illustration 17. Additionally a steam drum is often used, to ensure that the fluid enters the superheater as pure steam. The boiler can be somewhat simpler in ORC, as superheating is not needed. In sum this this increases the cost and scale of a Rankine power plant.

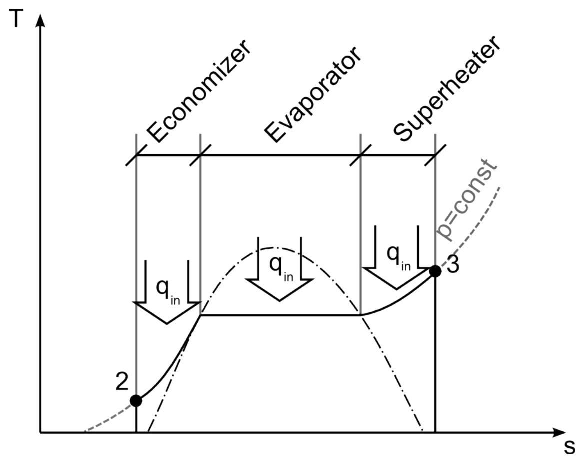

Illustration 17: The boiler is divided into the economizer, evaporator and superheater, by the phase the fluid is in

Recuperator

SRC cycles cannot use recuperators since the process reaches its lowest temperature in the expander. For fluids with positive vapor curves however, there is the option of using an recuperator to increase the cycle efficiency.

Condenser

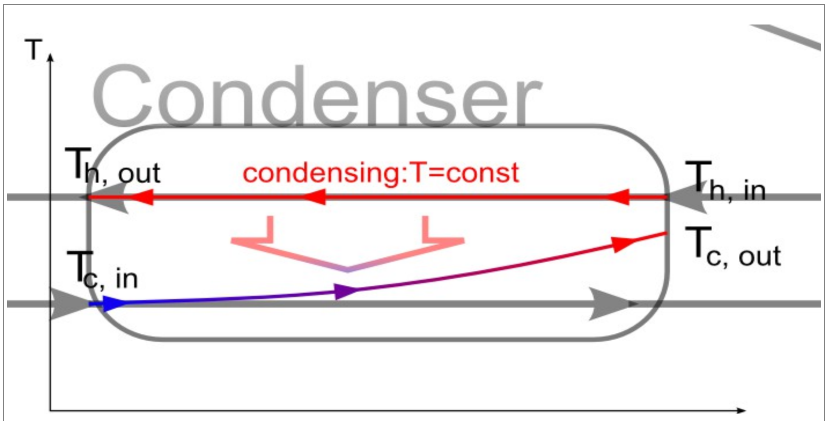

Illustration 18: The condenser temperature must be greater than the outlet temperature for heating

Since the Rankine cycle is closed, cooling is needed to return the working medium to its original state. While extracting the export heat from the condenser loop might seem an obvious solution, this can affect the efficiency of Rankine cycle. The temperature used for heating purposes is usually around 60 – 70ºC, effectively limiting the temperature that can be achieved in the condenser to above this as shown in illustration 18. Thus a CHP plant using the Rankine cycle could expect lower electrical efficiency than the same plant able to cool to lower temperatures.

Expanders

The expander has a major impact on the efficiency of the power cycle, and needs to be of appropriate type to the operating conditions and the mass flow in the cycle. In general we separate between turbo and displacement types expanders. In turbomachinery energy exhange happens between a rotor and the fluid, while in displacement types the space in which the fluid is situated changes volume. Displacement type expanders are usually more efficient for low mass flows and higher pressure ratios per stage. Turboexpanders also tend to have higher rotational speeds than displacement expanders, leading to higher costs and more complexity.

Centrifugal turboexpander

Either of axial or centrifugal type, with centrifugal expanders being used for smaller scales, for the same reasons as in GTs. That is centrifugal expanders have higher pressure ratios per stage as well as being more compact than axial expanders overall.

Screw Exapander

A screw expander consists of one to three enclosed interconnected screws. As the screws turn they force the gas through the treads as shown in illustration 19 below.

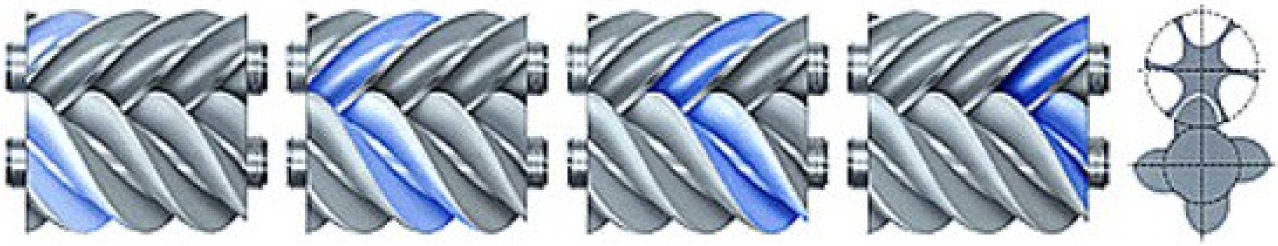

Illustration 19: Screw compressor. Image of an Atlas Copco ZT/ZR 110-900 rotary screw compressor. [ESI.info, 2013]

Scroll Expander

A scroll type expander consists of two spirals, one being static and the other moving eccentrically without rotating. This creates a volume between the spirals that becomes larger as the fluid moves outward radially. In illustration 20 below we can see the expanding volume as the red spiral moves without rotating, while the black spiral is static. The cross-hairs in the illustration are added for clarity of the motion.

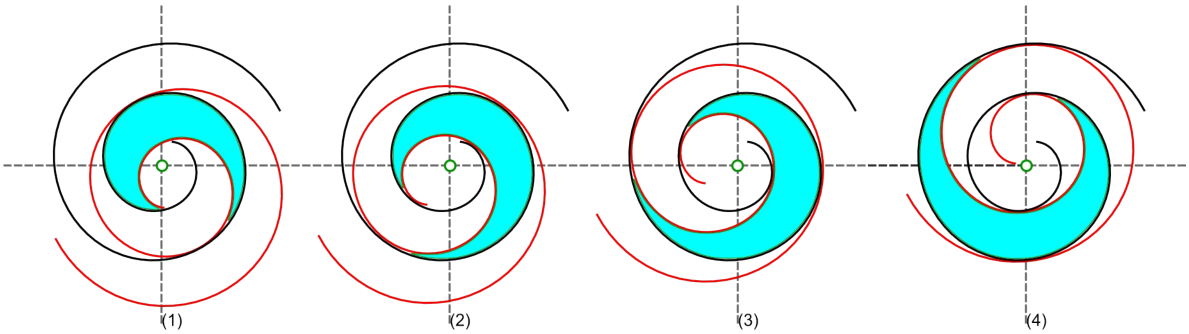

Illustration 20: Workings of an scroll expander. As the fluid expands, the red scroll is moved around without rotating.

Reciprocating/piston expander

Reciprocating expanders are an old and well known technology, so much so that water-steam cycles using them are simply known as steam engines. The expander consists of a cylinder and a piston, with the piston moving inside the cylinder as the gas first expands and is expelled afterward. Work is extracted usually by a turning wheel connected to the piston by a rod. Two or more cylinders are needed, as the some of the work of one piston is used to discharge the fluid in other pistons after expansion. A problem with reciprocating expanders is that they require inlet and outlet valves. These are historically a weak point in the construction in comparison to other expander types. Modern electronic or hydraulic control over these valves are superior than older mechanical solution however, lessening this drawback.

Reciprocating expanders are being revisited as a technology for small- and micro-scale CHP, as advances made for internal combustion engines are applied to the Rankine cycle [Ferrara, Manfrida and Pesconi, 2012][Clemente et al., 2011].

Efficiency

While large scale SRC power plants can reach very high electrical efficiencies, this drops off rapidly as the scale becomes smaller. This is in part because the low mass flows caused by the high specific heat of water cause the efficiency of the expander to drop rapidly. The data in table 1 is taken from a 2008 review of CHP technologies done by the U.S. Environmental Protection Agency (U.S. EPA). [U.S. EPA, 2008] It shows the drop off in isentropic efficiency and electric power efficiency with the scale of the plant.

	System 1	System 2	System 3
Nominal Electrical Power	500 kW	3000 kW	15000 kW
Turbine Isentropic Efficiency	50.0%	70.00%	80.00%
Cycle Electrical Efficiency	6.40%	6.90%	9.30%
Power to Heat Ratio	0.09	0.1	0.13

Table 1

The efficiency difference between SRC and ORC is mainly dependent on the temperature available, and for temperatures above 430ºC, SRC should have the highest electrical efficiency. [U.S. EPA, 2012]

RAMS

Time before maintenance and life time of the system will vary greatly with the type of cycle chosen. In general ORC cycles with lower pressures, and displacement type expanders should have lover maintenance needs than a SRC with an turboexpander. In the previously mentioned U.S EPA study they found that a small scale SRC CHP power plant should have more than 50,000 hours between maintenance, and that of the reviewed technologies, it would have the lowest maintenance cost per kWh_{el} produced. The same review also gives SRC the highest availability (neither thermoelectric generators or Stirling engines was included in the review).

As previously mentioned ORC should have even better performance regarding maintenance needs, caused by the cycles lower pressure. However maintainability will be lower, caused by the usually either toxic or inflammable nature of the fluid used, as well as the increased sealing of the system needed because of this. Since the fluids used are similar to those used in heat pumps and refrigerators, very widespread technologies, we might assume that these factors affects supportability somewhat less. [Alvarez, 1990][Cengel and Boles, 2006]

10 Stirling Engine

A Stirling engine runs, unsurprisingly, on the Stirling cycle, shown below in illustration 21. The ideal Stirling cycle starts with isothermal expansion by increasing the volume in the engines cylinder. The heat is then regenerated, stored internally, while the working fluid is moved from the heat source to an heat sink at constant volume. Heat is then rejected isothermally to the heat sink by compression, before the fluid is again moved back to the heat source while returning to the temperature of the source by absorbing the heat stored in the regenerator.

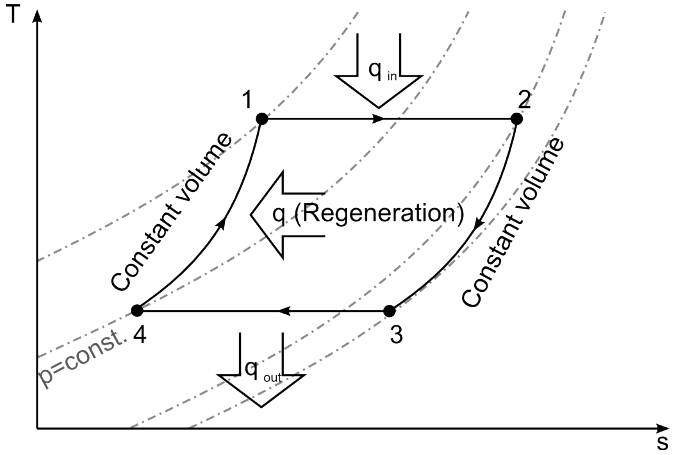

Illustration 21: The Stirling cycle in a T-s diagram

The attractiveness of the Stirling cycle for power generation is that the ideal Stirling cycle has the same thermal efficiency as an ideal Carnot cycle. Several factors prevent Stirling engines reaching their theoretical maximum. In processes where Stirling engines are used, they typically achieve electric cycle efficiencies of around 20 - 35%.

While the theoretical Stirling cycle is easy to understand, physically accomplishing the process in an engine poses more of a challenge. Many different mechanical solutions exists, and shown in illustration 22 is a simple example, intended to show a physical solution to achieve the Stirling cycle.

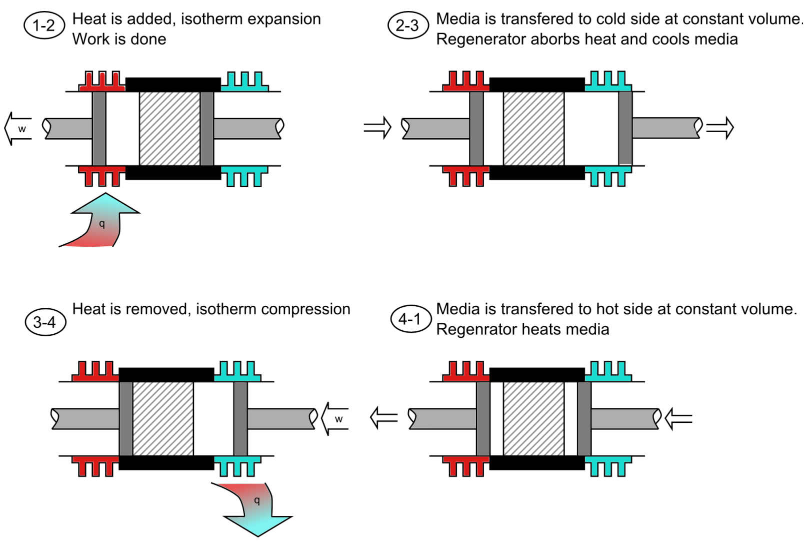

Illustration 22: Simplified workings of a Stirling engine

The real Stirling cycle deviates a lot from the ideal one. The ideal Stirling cycle involves isothermal heat transfer, something witch would require either an infinite surface for heat transfer, or an unlimited amount of time. There is also the ever present heat loss, as well pressure loss in the machine, as well as the fact that the regenerator will always be imperfect.

Of the advantages of the Stirling cycle is that as a closed cycle, different working fluids may be used. These are usually gasses with low molecular weights, to increase heat transfer between the working fluid and the cylinder walls. However, the low molecular weigh means that avoiding leakages from the process is difficult. The largest problem has usually been sealing around the piston shaft, a problem that in some Stirling engines has been solved by designing the whole engine in a closed pressurized chamber, putting the generator inside.

In sum these problems add up, and Stirling cycles with efficiencies high enough to be practical have been difficult to produce. While there are some solutions available on the <3kW_{el} scale [SenerTec, 2013] [Whispergen, 2011], for our system the 10 – 50 kW_{el} range would be most interesting; several engines in this range are under development or nearing market deployment [BIOS, 2013] [Viking, 2009] . As time of writing only one solution, the SOLO Stirling 161, with a rated power of 2 – 9.5 kW_{el} seems to be commercially available [Build Up, 2009].

An option would then be connecting several such smaller Stirling engines in parallel. This would have the added advantage of adjusting the number of engines being active with the heat available, leading to better part load efficiencies.

RAMS

As few systems are commercially available, there is little data on RAMS for Stirling engines. With the relatively low maturity of the technology for CHP purposes we might expect supportability to be low. Because of the closed and complex nature of the engine itself maintainability would also be poor.

11 Thermoelectric Generator

A thermoelectric generator uses semiconducting materials to generate electricity from heat. The electricity generated comes from the Seebeck effect, which causes charge carriers to diffuse in a material from the hot side to the cold side. A generator will contain thermoelectric cells consisting of p-blocks and n-blocks connected between two plates of non-electrically conductive material with high heat conductance. The n- and p-blocks conduct either electrons or "charge holes" with the direction of the temperature difference. For thermoelectric heat pumps the same technology is used in reverse with the heat being transferred with the direction of the current in the p-blocks and against it in the n-blocks as hown in illustration 23. When used to drive heat transfer, e.g. cooling or heating, its called the Peltier effect. The effects are collectively sometimes called the Peltier-Seebeck effect.

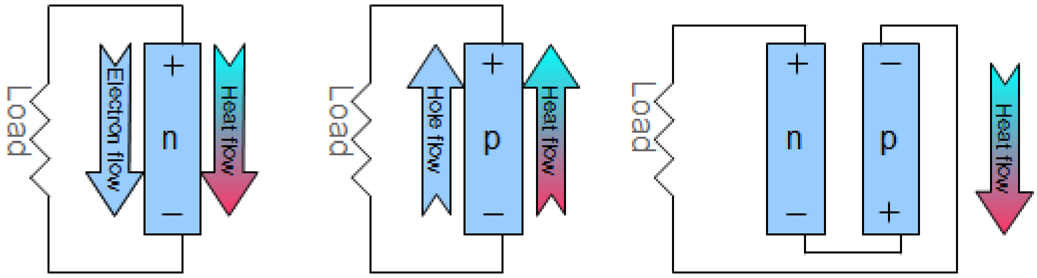

Illustration 23: Thermoelectric blocks of the n- and p- types, and a thermocouple

Usually p – and a n-blocks are connected in series to create a thermocouple as in illustration 23, and several such couples make up a thermoelectric cell.

The efficiency of thermoelectric cells, and thus thermoelectric generators (TEGs), are mainly dependent on the size of the temperature difference between the hot and cold side, as well as the average temperature of the process, and the thermoelectric properties of the materials used, quantified in the materials Z-value.

Also called the thermoelectric figure of merit, the Z-value expresses the the thermoelectric performance of a given material, and is defined as $Z = \alpha^2 / kR$. Where α is the Seebeck coefficient, *k* is the materials thermal conductivity, and R its electrical resistivity. The Seebeck coefficient is the voltage over the material as a function of the temperature, with a unit of $\mu V/K$. The value of α can be both positive and negative with a positive value correlating with p-block materials, having the current going the opposite way of the heatflow. [Ono and Suzuki, 1998] The thermoelectric figure of merit is often multiplied on both sides with the average temperature of the hot and cold side, and given as ZT. [Riffat and Ma, 2003] The usefulness of the ZT value is that it is defined at a specific temperature point, as the properties that make up the value will change with the temperature.

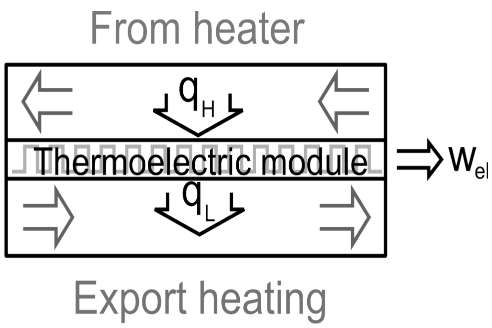

Illustration 24: Thermoelectric cells placed between the hot and cold side of an heat exchanger to create electrical power

As can be seen by its definition a higher value of ZT gives higher conversion efficiency as this means that the voltage difference is larger compared to the materials internal electrical resistance and the heat flow across it. Thus a good thermoelectric material will have a high Seebeck coefficient, and low thermal conductance and electrical resistivity at high temperatures.
A thermoelectric cell is shown above in illustration 24, as integrated into a heat exchanger. The efficiency of the cell is simply the difference the heat given from the hot side divided by the amount of power generated. That is:

$$\eta_{module} = \frac{w_{el}}{q_H}$$

TEGs are normally not standalone systems, but are designed into other systems, as a way to increase efficiency. A simple standalone TEG would simply consist of an heating source and then an heat exchanger where the TEG would be placed between the hot and cold sides.

RAMS

Having no moving parts TEGs have very low maintenance requirements beyond keeping them and the heat-exchange surfaces clean. Global Thermoelectric, a leading manufacturer for TEGs say that one to two hours yearly maintenance is required, and that a sealed module can have a lifetime of up to 20 years. [Global Thermoelectrics, 2013?]

Manufacturers rarely give hard numbers for efficiencies, but 5% is usually cited for mid range TEGs.

TEGs today mostly see use in more exotic applications and remote applications, such as in space exploration or for remote instrumentation and communication.

12 Evaluation

Factors

To make a good comparison between the different technologies, we need to decide on a set of factors which to look at.

As previously discussed electrical power is the most valuable product of the power cycles, and is thus the first factor included. The power to heat ratio is also important for many of the same reasons that efficiency is, and is also included. Part load performance speaks to the flexibility of the system in different scenarios. While we might hope that a system can run on full load all the time, this will rarely be the case, as the system most likely will be controlled by the need for heat. In the event that the system is installed where other heat sources or large enough heat storage is available, this factor lessens in importance.

Specific cost influences strongly the economy of the system. Other factors also influence this, and payback time, for example is also dependent on other factors such as running cost and the price attained for electricity sold. However, we judge installation costs to be the most important factor in the economy of the system, and it also usually represents the single largest barrier to get an system realized. Maintenance amount is included as a system that can largely run by itself is strongly preferable. Whether or not regular maintenance has to be done by outside resources, maintenance represents a significant additional running cost both in time and in cost.

Lastly, the specific footprint is a good measure of how easy a technology would be to install in existing buildings. A large system represents a barrier to implementation, while a small system will have better flexibility in where it can be used. As the system is intended for a somewhat remote location, we will strongly prefer a system that has high maintainability. This means that the system should be reparable with easily attainable resources and expertise. The maturity of a technology is important in that it influences the uncertainty in the other factors, as well as the ease of acquiring outside support and spare parts. As such it is an influencing factor in maintainability.

Table 2 below presents data for these factors for the different technologies, and is intended to be illustrative of the qualities of the different technologies.

	Gas Turbine[1,2]	Rankine - SRC[1,8]	Rankine - ORC[1]	Thermoelectric[3]	Stirling Engine[3]
Electric cycle efficiency	20 - 30%	6.00%	-	4 – 5 %	24,00%
Power to heat ratio	0.5-2	0.1	-	0.04 - .05 %	0.4
Part load performance	poor	medium	good	good	good[7]
Specific cost [€/kW]	1850-2300[4]	500 - 1000[4]	-	2500 – 2800 [4,5,6]	2600
Specific footprint [m^2/kW]	0.01	0.01 – 0.1	-	n/a	0.1
Maintenance amount	Medium	Low	Low-	Low-	n/a
Maintainability	Low	Medium	Low	Low	Low
Maturity	High	High	Medium	Low	Low

Table 2 The main properties considered in evaluation

[1] [U.S. EPA, 2008]
[2] Data for microturbine with internal combustion
[3] Calculations in Appendix B
[4] Calculated from U.S. Dollars using an exchange rate of $1USD = 0.76€.
[5] Calculated from British Pounds using an exchange rate of £1 = 1.18€
[6] Prices for thermoelectic cells only.
[7] Good part load performance predicated on a system design consisting of several parallel units.
[8] Data for SRC in the 500 kW_{el} scale.

The data presented are meant to be illustrative for the system specified

Ranking the technologies

Steam Rankine Cycle

Estimating the price of small scale steam Rankine systems is difficult. Most of these systems are custom built, and price may as such vary widely. No data on systems in the relevant range have been found. However, the previously cited 2008 review by U.S. EPA and by a 1999 review by Onsite Sycom for the California Energy Commission have found their specific installed cost to be below that of similar power microturbine systems. The efficiencies of these systems are highly dependent on their size, and the 6% figure is for a system on the 500kW scale, meaning we might expect even lower efficiencies on the small to micro scale that we consider.

The maintenance required is as previously discussed little. Maintainability is the best of the technologies required, as a SRC plant will be easy to understand and carries little risk in working on it, as long as it is shut down and cold. SRC plants have the highest maturity, as it is an old technology and widely used.

Organic Rankine Cycle

ORC suffers from the same lack of pre manufactured commercial packages as SRC, and is similarly difficult to evaluate. However, following the discussion of its workings, we can well compare it to SRC. As we have high temperatures available to us, the ORC main advantages in regard to SRC disappears. Generally having lower efficiency than SRC, this puts ORC in the same range as TEGs. As they are so close, and the outside variables that might affect this (fluid selection, efficiency of power electronics) we chose to put ORC on a shared last place for efficiency. Part load performance should however be better than that of SRC [Drescher and Brüggemann, 2007]. The lower pressures of the cycle, and the lack of a superheater should make an ORC system smaller and cheaper than SRC. The lower pressures should also lead to even less maintenance, but the safety factors concerning the flammability and toxicity of the organic fluid will make maintainability much worse. Supportability plays a role in maintainability however, and given its similarity to SRC, it should be easier to find competence for maintenance on an ORC system than on either thermoelectric- or Rankine systems. [Vankeirsbilck et al., 2011]

Thermoelectric Generator

Thermoelectric cells are a coming but still very immature technology. They also have the lowest electrical efficiency, and is in addition very costly. The prices cited are for the cells only, and installation costs would increase these costs even more. The only technology with similar costs is the Stirling engine, but that comes delivered as a whole power unit ready for CHP. This leads us to put thermoelectric as the most expensive technology. Its part load efficiency is however very good, and the efficiency should change little depending on the heat flux through it. In therms of size it should also be the smallest to install, and the solid state nature of the cells means that they are close to maintenance free. This also means that any problems with the cells themselves will be very difficult to fix, and the most likely option would be to replace any broken cells, meaning very poor maintainability unless one can afford to have enough spare cells available. The low maturity of the technology is also a big problem. Manufacturers are small and few, and the availability of spare cells or support is very insecure.

Stirling Engine

The data for the Stirling engine is based on the SOLO Stirling engine, the only one found that is in the appropriate range for our application. While it is still a bit small a way to get around this would be to run several such engines in parallel, adjusting the number of engines running to the heat demand. This solution would be highly efficient on part load, and should be better than both Rankine cycles in this regard. The technology have similar or better efficiency than that of a gas turbine, while having the next highest heat to power ratio.

The technology is expensive, and is close to thermoelectric cells in this regard, with the advantage that the SOLO Stirling engines would be easier and cheaper to install. Maintenance data is hard to come by, but as it is a closed system delivered as a unit, and marketed towards private customers, we can assume that maintenance is low. Putting the maintenance amount relative to the other technologies, we rate it as better than the gas turbine, but poorer than the other technologies, mainly because of the great uncertainty. Maintainability is rated as the poorest of the technologies, because the cylinders in the SOLO are sealed and contain Helium at high pressure, making it a complex task to maintenance on. The rest of the unit should however be simpler in maintenance. The low maturity of the technology also means that qualified technicians would be hard to come by.

Gas Turbine

For gas turbines we are lucky enough to have a commercially available package for our consideration, the Capstone C30. Together with the Turbec T100 this gives are data a good grounding in the relevant area. All data however are from the turbines using internal combustion as the heat source. As discussed we would expect this to negatively impact the efficiency of the cycles compared to what is cited from the manufacturers. The heat exchangers needed would also increase the cost and size of the system somewhat.

Looking at the numbers we should still expect the gas turbine to be the second most efficient technology, and the power to heat ratio to be the highest. The specific cost would also rise, but this is true of all the technologies, and the additional work needed for the gas turbine should not exceed those.

EPA evaluated micro turbines as the most maintenance intensive technologies, placing it above ORC in this regard, making it the worst technology in this regard. The lack of internal combustion should positively impact the need for maintenance however, as no flue gasses are run through the turbine, and no combustion chambers exist within the GT itself. The maintainability of the system should be worse than SRC, simply because the technology is less well known, but otherwise it should be simpler to work on than the other technologies.

Weighing of the Factors

Comparing the technologies against each other as above, we end up with table 3 below, ranking the systems with 1 being the best and so on. The colors indicate in general terms how good or bad a quality is, with green being good, yellow OK, and green good. So the Stirling engine, for example, has good efficiency, power to heat ratio and part load performance, OK specific footprint and poor specific cost, maintainability and maturity.

Just summing the scores with the lowest being the best score, the gas turbine seems the best, being or second best in all aspects except part load performance and maintenance amount. Looking at the colors SRC or ORC might be more attractive, with all aspects either being good or OK. All factors are not equal, however and a weighing of the factors will reflect this.

Electric cycle efficiency	2	4	3	5	1
Power to heat ratio	1	4	3	5	2
Part load performance	5	4	3	1	2
Specific cost [€/kW]	1	3	2	5	4
Specific footprint [m^2/	2	5	4	1	3
Maintainance amount	5	4	3	1	2
Maintainability	2	1	3	4	5
Maturity	2	1	3	5	4
Sum	20	26	24	27	23

Table 3:The properties of the different technologies, with number indicating rank, and color indicating general quality

The method chosen for weighing is a simple multiplication of the weighing factors with the technologies value. Since we have chosen a lower value to represent better performance, the weighing factors should be larger the more important. This will result in low values for good performance in important factors and high values for poor performance in the same. Summing these values for the technologies should result in low values for those that are good performers in the most important categories.

The factors have been weighed as follows, with what was evaluated to be the most important factor first: Electrical efficiency, maintainability, maturity, part load performance, specific cost, specific footprint, maintenance amount and finally power to heat ratio. The results are in table 4. The relative weighing thus represents the relative importance of these factors.

	Weight	Gas Turbine[1,2]	Rankine - SRC[1,8]	Rankine - ORC	Thermoelectric[3]	Stirling Engine[3]
Electric cycle efficiency	8	16	32	24	40	8
Power to heat ratio	1	1	4	3	5	2
Part load performance	5	25	20	15	5	10
Specific cost [€/kW]	4	4	12	8	20	16
Specific footprint [m^2/kW]	3	6	15	12	3	9
Maintainance amount	2	10	8	6	2	4
Maintainability	7	14	7	21	28	35
Maturity	6	12	6	18	30	24
	36	88	104	107	133	108

Table 4: Weighing of the score by multiplying the weight by the systems rank. The sum of these multiplications are on the bottom. Lower values are better.

It is important to note that these numbers does not represent objective measures of the technologies qualities for our application, but is based on qualitative values.

Conclusion to Part One

Looking at the matrix' above, it is clear that gas turbines score the lowest both when weighted and unweighted. Looking at the quality of the different aspects, represented by the colors in table 3, we also see that gas turbines have the most aspects rated as "good". The main drawback for using a gas turbine is also apparent as its poor part load efficiency. The choice of gas turbines as the preferred technology then hinges on the severity of this drawback. It is also worth noting that these drawbacks also affect the technology with the second lowest score, the SRC, although to a lesser degree.

We've rated part load efficiency as the fourth most important aspect of the technologies, and with good reason, and compensating for this problem will affect the other qualities of the gas turbine.

As mentioned when discussing CHP generally there is the possibility of adding heat storage to a CHP system. This would allow us to run the turbine on higher and more even load throughout the year, but such a solution would however significantly increase the cost of the system, as well as adding new systems to be maintained. Heat storage for the time periods that would be needed is also usually done in the ground, and requires ground to be available and have the right properties for heat storage.

Another solution is to simply run he turbine at higher loads all year, and accept the heat losses involved in this. This is of course predicated on the ability to sell the excess power. When analyzing the gas turbine, a power vs. heat input graph could be used to maximize the systems total efficiency through the year. This solution is simple to implement for gas turbines, as the flue gasses that contain the heat would simply leave the system without requiring further cooling. Another aspect of gas turbines that makes this a reasonable solution is their high power to heat ratio, which means that the potential heat wasted is less than for other systems.

The last problem with this option is the one of fuel. As the system we chose regardless of technology is to have the same maximum heat input, a solution using this full input year around, instead of running on part load for greater periods of the year, will then consume more heat. Thus if the system is to run year-round on rated power it would consume fuel to a much larger degree than a system that would be able to run on heat demand.

Since neither heat demand nor fuel availability is specified in the system description, we see that the problem of poor part load is acceptable, and that a gas turbine should be our preferred solution.

Part 2

13 Introduction to Part Two

Having chosen our preferred technology, we shall look closer at the design of a CHP plant using a gas turbine. The main aspect that is to be looked at are alternative layouts of such a plant, and the resulting efficiencies.

As a base for our CHP system we will use the Capstone C30 gas turbine. The C30 is appropriately sized, commercially available, and good data was found for it from an analysis done at the University of Colorado [Guiterman, 2005]. This gas turbine is available only with internal combustion, and needs to modified to fit our needs.

Simulation has been done using DNA, an open source thermodynamic network simulation tool developed at the Technical University of Denmark [Elmegaard, 1999]. DNA models are made up of components (turbines,valves,boilers etc..) connected by nodes, and take the form of code. The DNA code used for the simulations in part two has been included in appendix A, along with comments on the code and a short explanation of how to read that code. This should enable any reader to reproduce the results given they have access to DNA.

Ambient conditions used are 1.0132 bar and 15ºC. In addition an outlet temperature from the export heater has been set to 70ºC, assumed to be adequate for residential heat needs.

The Capstone C30 Gas Turbine

The Capstone C30 is a 30kW$_e$ rated variable speed micro turbine with integrated generator and control electronics. Without the on board gas compressor it has a rated electrical efficiency of 26% (LHV). It also has no gearbox and uses no oil or other lubricants. Designed for low maintenance and direct grid connection, it is very suitable for our CHP plant. An image of the turbine itself is shown below in illustration 25, and the manufacturer provided data is in table 5.

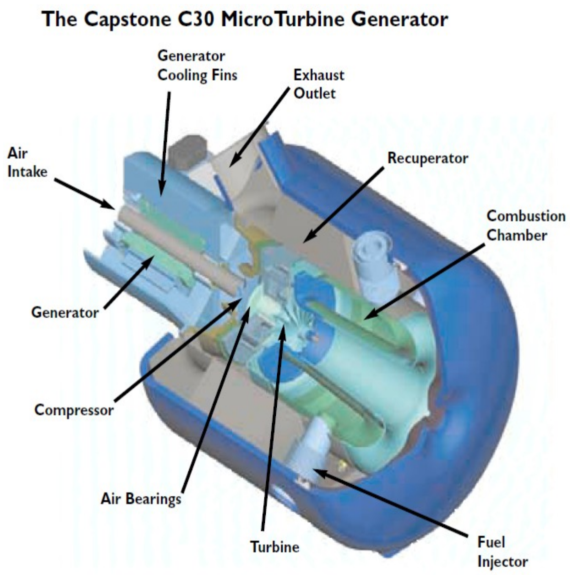

Illustration 25: The Capstone C30 gas turbine. [Capstone,2010]

Nominal Power output	$30kW_e$
Electrical efficiency	27.00%
mass flow out of turbine	0.31 kg/s
Turbine Exit Temperature	275°C
Size (W*D*H)	0.76*1.5*1.8 m

Table 5: Manufacturer data for Capstone C30

This data from the manufacturer is intended for prospective buyers, and as such does not constitute a solid base enabling us to simulate and analyze the CHP plant. For a complete picture of the gas turbine the pressure ratio and the efficiencies of the compressor and turbine needs to be known or at least possible to extrapolate. For a better basis more information than what is given from the supplier is needed. This data was found in an analysis done at the University of Colorado (CU) running the C30 as part of an attempt to reduce energy costs at the university's pool [Guiterman, 2005]. The analysis contains key properties of the internals of the gas turbine, and are presented below in table 6 and 7.

Pressure ratio	2.4
Compressor Isentropic Efficiency	70.00%
Turbine Isentropic Efficiency	81.00%
Recuperaor Effectiveness	81.00%
Pressure drop over recuprator, relative	7.50%
Electrical Efficiency	21.80%
Fuel input	48.06 kW
mass flow air	0.17 kg/s

Table 6: Data reported in University of Colorado study.

Turbine inlet temperature[1]	20°C
Turbine outlet temperature	139°C
combustion chamber inlet temperature	516°C
Compressor inlet temperature	764°C
Compressor outlet temperature	605°C
Recuperator outlet temperature	442°C

Table 7: Temperatures reported in University of Colorado study

[1]Inlet temperature was calculated by adding 6.7°C to the ambient temperature. This temperature rise is caused the generator being cooled by the inlet air.

The university of Colorado is situated at around 1600 meters above sea level, and the lower ambient air pressure and density causes the power output of the turbine to drop about 20%. This efficiency drop is caused by the reduction in air density. The data reported was for lower loads than we would expect for our plant. As mentioned earlier part load performance of gas turbines are poor, so this has a large effect on the efficiency of the system at CU. In addition these tests where performed at an ambient temperature of 13.3°C a bit lower than our conditions, but this should have little effect on the results.

The Isentropic and Polytropic Efficiency

As mentioned in the discussion of the Brayton cycle, the most used way to describe turbine and compressor performance is the isentropic efficiency. Being a measure of the real change in state over the machine in comparison to an ideal process, it is a useful way to measure the quality of the process. The isentropic efficiency is however dependent on a constant pressure ratio, and thus is not suitable for our simulations, as we will experience different pressure losses than the CU case.

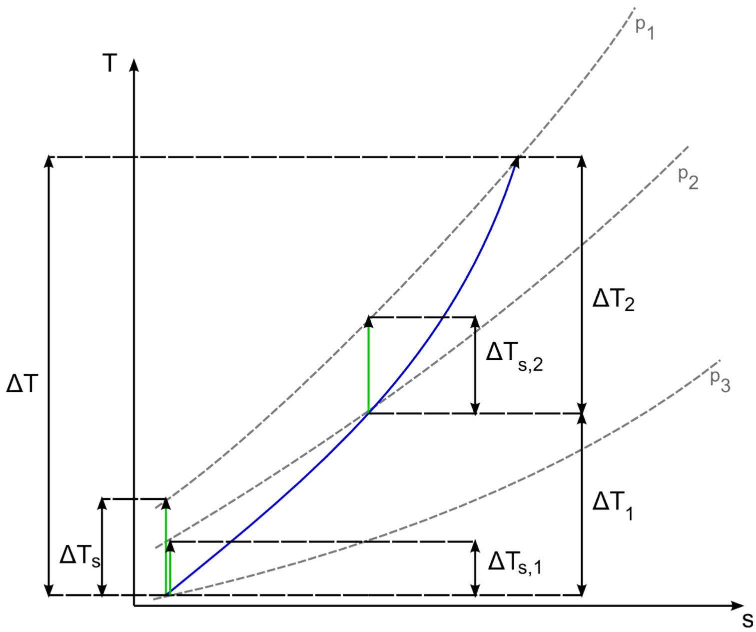

Illustration 26: An compression process in the T-s dimensions

This is shown in illustration 26 where a compression from pressure to p_1 to p_3 is broken up into two parts. Using the simplification of an constant specific heat for the process, the isentropic efficiency becomes $\Delta T/\Delta T_s$ and is illustrated by the relative length of the lines. The divergence of the constant pressure lines have been greatly exaggerated in this diagram to illustrate the fact that the isentropic efficiency is not constant during the expansion.

$$\eta_{\text{isentropic,compressor}} = \frac{\Delta T_s}{\Delta T} \neq \frac{\Delta T_{2,s}}{\Delta T_2} \neq \frac{\Delta T_{1,s}}{\Delta T_1}$$

The solution is to replace the isentropic efficiency with the polytropic efficiency. The polytropic efficiency takes it name from the basis that the process is assumed to be polytopic, that is pV^n=constant, where n is the polytropic constant, which can be any real number. Assuming the process is adiabatic, the polytropic efficiency is the relation between the specific work of the flow and the enthalpy change, and this is assumed to be constant through the whole process. The polytropic efficiency is defined as follows:

$$\eta_{polytropic,compressor} = \frac{v\, dP}{\Delta h} \qquad \eta_{polytropic,turbine} = \frac{\Delta h}{v\, dP}$$

Part Load Performance of a Variable Speed Gas Turbine

As the C30 is a variable speed turbine, made possible by integrated power electronics, the speed of the compressor and turbine will vary at part load. This is in contrast to larger gas turbines for power generation, which usually will have a set rotational speed, and thus a set mass flow. The result is that when running in part load, i.e. with a lower than design point heat input, the mass flow will lessen in response, to keep the turbine inlet temperature (TiT) steady. The result is higher efficiency at partial loads, as TiT is more critical to the systems overall efficiency than is the mass flow.

The reduction in mass flow will affect the compressor and the turbines efficiency, however, as they are optimized for a certain flow speed of the gas and rotational speed of the machine. The performance of these components under varying loads is usually presented in compressor and turbine maps, an example of which is shown below in illustration 27. As these maps are unavailable to us for the compressor and turbine in the C30, their efficiencies at lower mass flows is unknown, and part load performance becomes difficult to predict. This means that although the efficiencies of the turbine and compressor is assumed constant in the following analysis this is based on the assumption that the components would be modified or replaced to fit a changed mass flow.

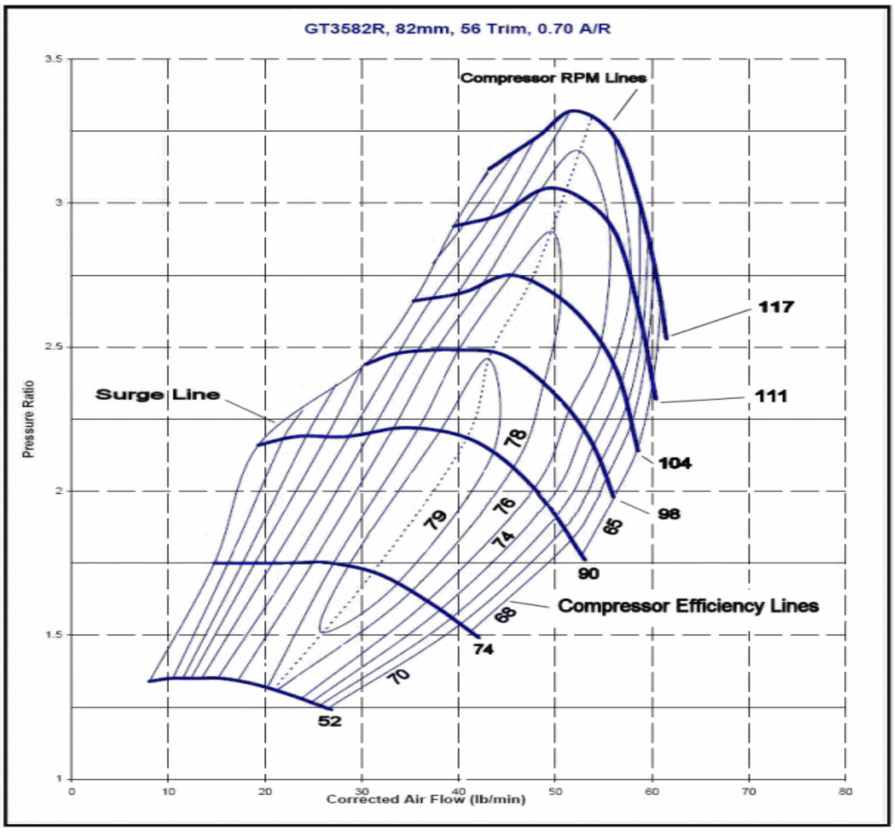

Illustration 27: Example of compressor chart [EPI Inc, 2012]

Adding to these difficulties is the fact as the TiT is kept steady and the efficiency of the turbine decreases, the turbine outlet temperatures will increase. This is apparent from the definition of turbine efficiency shown in the previous discussion of gas turbines. As load decreases the turbine outlet temperatures will reach a limit set by the tolerances of the materials and units after the turbine. When this temperature is reached, further reduction in the load is achieved by reducing the turbine inlet temperature. This relationship is sketched out in illustration 28 below.

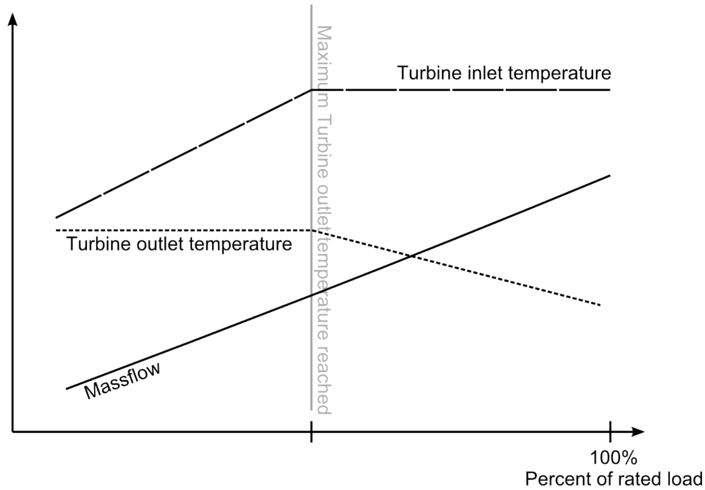

Illustration 28: Mass flow and turbine inlet and outlet temperature as function of reduction in load

14 Recreation of University of Colorado Gas Turbine

Using the data cited from the CU study a simulation of that gas turbine has been set up using DNA to ensure that the efficiencies of the turbine and compressor is correct and that the simulation is functioning correctly. Since there is no data on the air pressure at the time of the measurements, the inlet pressure was set at 0.83 bars, given in Cengel and Boles as the average air pressure of Denver, situated at the same height. The relative pressure loss over the recuperator of 7.5% was understood to be for both the hot and the cold sides. The gas burned in the simulation was pure methane. This simulation resulted in an efficiency of 15.63%, well below that reported from CU.

Needing to modify the data the question then becomes which data we should recreate, and which should be modified to fit. In cooperation with Peter Breuhaus, an experienced gas turbine engineer, the data was evaluated. The main conclusion was that the efficiency of the compressor seemed very low, and the pressure losses over the recuperator seemed excessive. The decision was made to mainly modify these with the aim of achieving the cited efficiency and power. The resulting data was a good match with the reported values. The changed values are shown in table 8 and is used to calculate the polytropic efficiencies used in the further analysis. The pressure drop is also retained for the further simulations.

New recuperator pressure loss, both sides	0.05 bar
New Turbine efficiency	0.82
New compressor efficiency	0.75
New recuperator effectiveness	0.825

Table 8: Modified values to fit the CU results

Using these values the polytropic efficiency of the turbine and compressor was found by setting them as variable in the simulation. This was achieved by setting properties calculated from a first run as additional constants. This resulted in a polytropic efficiency of 0.805 for the turbine and 0.778 for the compressor. Not changing the pressures, this resulted in a temperature drop relative to the isentropic case out of the compressor of 4ºC and a increase out of the turbine of 5ºC. This means there is a slight increase in performance of the compressor and decrease in the turbine, but within acceptable limits.

15 Base Case

A simple base case will be set up, and simulated first. The purpose is to provide background data for use in, and to compare to, more complex layouts. The combustion chamber in the C30 has now been replaced with a gas heater, meaning that there is now longer internal combustion, and that the turbine inlet temperature is set to 900ºC and a 100kW of heat input. To begin with we use the maximum temperature available to us, as this will likely result in the highest efficiency. For this purpose a simple model of the C30 with an additional export heat exchanger has been used. The system layout of this model is shown below in illustration 29.

For the export heat exchanger the mass flow was set as a variable, while the water inlet temperature was set at 55ºC and the water outlet at 75ºC. The Export heat exchanger effectiveness was set at 0.85; higher than the recuperator, since gas- to liquid heat transfer is more efficient. Initially no additional pressure losses where added. The system is simulated as being adiabatic, mechanical efficiencies for the compressor and generator has been set as 99% and the generators electrical efficiency as 100%. In addition dry air has been used as the medium in the cycle for this and the following simulations. The influence of these simplifications should not be large enough to significantly impact the conclusion we are able to draw from the results. In the case of a very detailed analysis and design calculations, theses values would have to be reevaluated.

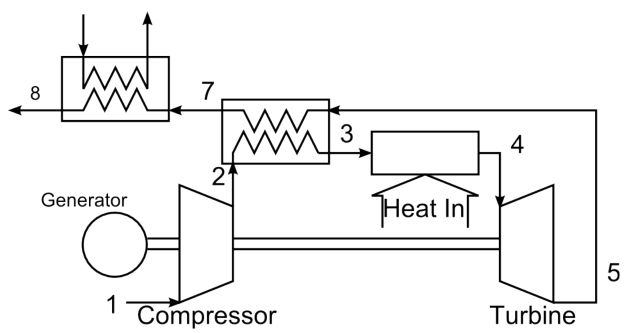

Illustration 29: System layout for the base case

This simulation produced the following key values shown in table 5:

Produced Electricity	28.2 kW
Cycle Electric Efficiency	28.2%
Exported Heat	51.9 kW
Total Efficiency	80.1 %
Power to heat ratio	0.54
mass flow at compressor inlet	0.31 kg/s

Table 9: Key values for unmodified base case.

Immediately apparent is the fact that we achieve higher efficiency th what the turbine is rated for. This might be caused by a combination of factors. Underestimation of he pressure losses inside the turbine or overestimation of the polytropic efficiencies might be factors, but most likely the main cause is the much higher TiT than in the manufacturers specification.

Changing the Turbine Inlet Temperature

If it should be apparent that the running temperature should have to be reduced for concerns over the materials limit, we should expect the efficiency to drop. The alternative would be to modify the GT to tolerate the temperatures. The base case simulation was done while varying the temperature, to look at the effects of changing the TiT.

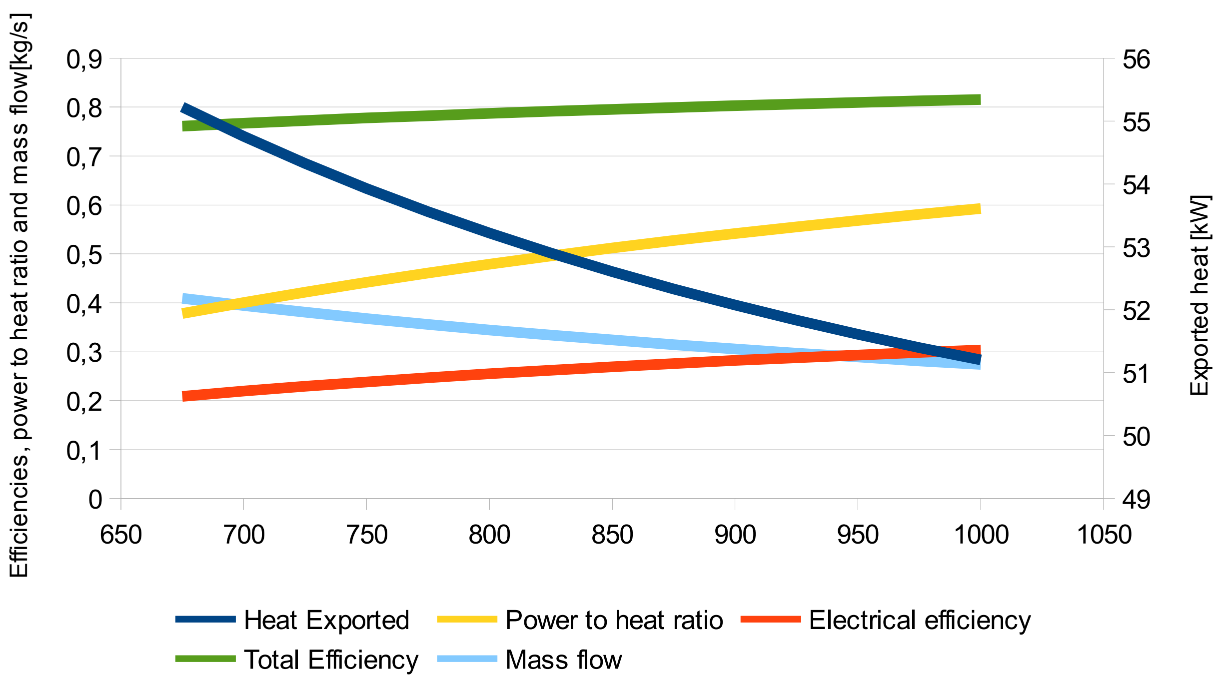

Chart 2: Power to heat ratio, mass flow, total- and electrical efficiency and heat exported, as function of the turbine inlet temperature for the base case.

The result of varying TiT can be seen above in chart 2. The heat produced is shown with a dark blue line, and plotted in kW on the right y-axis, while power to heat ratio, mass flow, electrical and total efficiency is plotted against the left y-axis. We should remember that for a given turbine or compressor there are efficiency changes as the mass flow changes. This means that while modifications might be made to the C30 to achieve one point on the TiT axis, a given gas turbine will not react according to the chart to changes in TiT. Put in other words, the chart above makes us able to consider what TiT should be the design point, and are not the results of changing the TiT in an running system.

The chart shows that as the temperature increases, the mass flow becomes less, as the heat input is constant. The temperature increase causes the electrical efficiency to rise, meaning that the turbine is extracting more work from the flow. This means that there is less heat available to the heat exchanger, and the heat exported decreases. The net effect is a slight increase of total efficiency, and a larger increase in power to heat ratio as the TiT increases.

Looking at the chart above we may be tempted to draw the conclusion that for this layout changing the TiT might be an attractive way to vary the power to heat ratio, with little cost in total efficiency. This would however result in kW for kW exchanging a unit of power for less than a unit of heat, something witch in most circumstances would be meaningless.

Pressure Drop Over the Gas Heater

Achieving a pressure drop of zero over the gas heater is very unlikely, and some additional pressure losses are to be expected. The base case has therefore been run while varying the pressure loss over the gas heater. While doing this the pressure ratio has been kept constant over the turbine, as the compressor works harder to make up for the pressure loss. The results are shown below in chart 3.

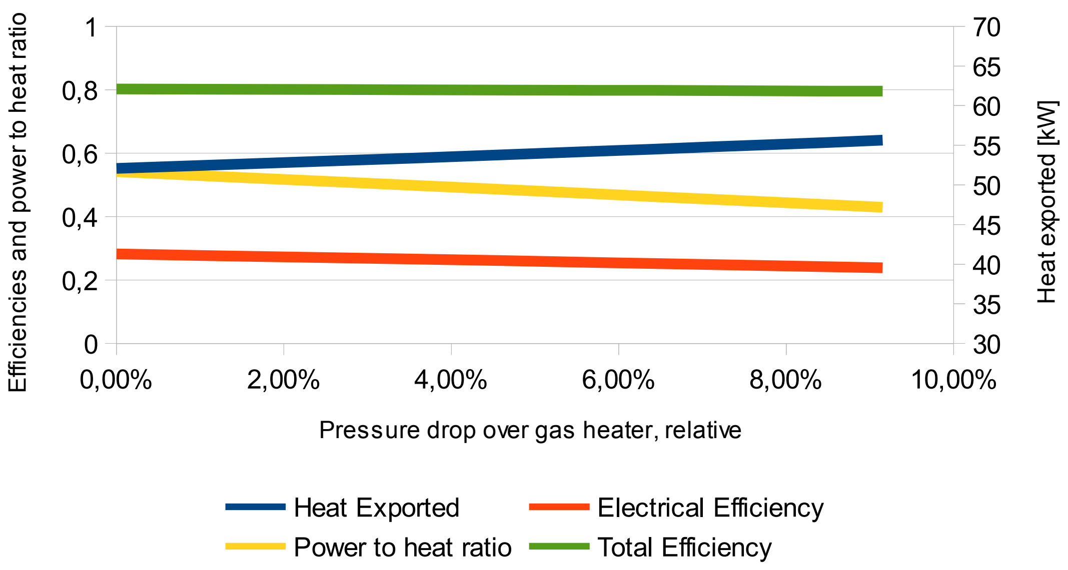

Chart 3: Power to heat ratio, total- and electrical efficiency and heat exported, as function of pressure losses over the gas heater. The heat exported is again plotted against the right axis.

As the pressure drops the compressor consumes more of the turbines energy to reach the target turbine inlet pressure. This results in the electrical efficiency going down as the pressure loss increases, since less energy is available to the generator. The increase in exported heat with the pressure drop is caused by the decreasing electrical efficiency, as less heat is converted to work in the turbine. The increase is smaller than the decrease in power produced in absolute terms, and the end effect is a slight decrease in total efficiency with increasing pressure losses.

Part Load Performance

As previously described, the part load performance is difficult to predict for several reasons, but general trends can be predicted. Part load has been simulated by reducing the heat input by way of the mass flow of fuel while TiT was kept constant, shown in chart 4, and while mass flow was kept constant, shown in chart 5.

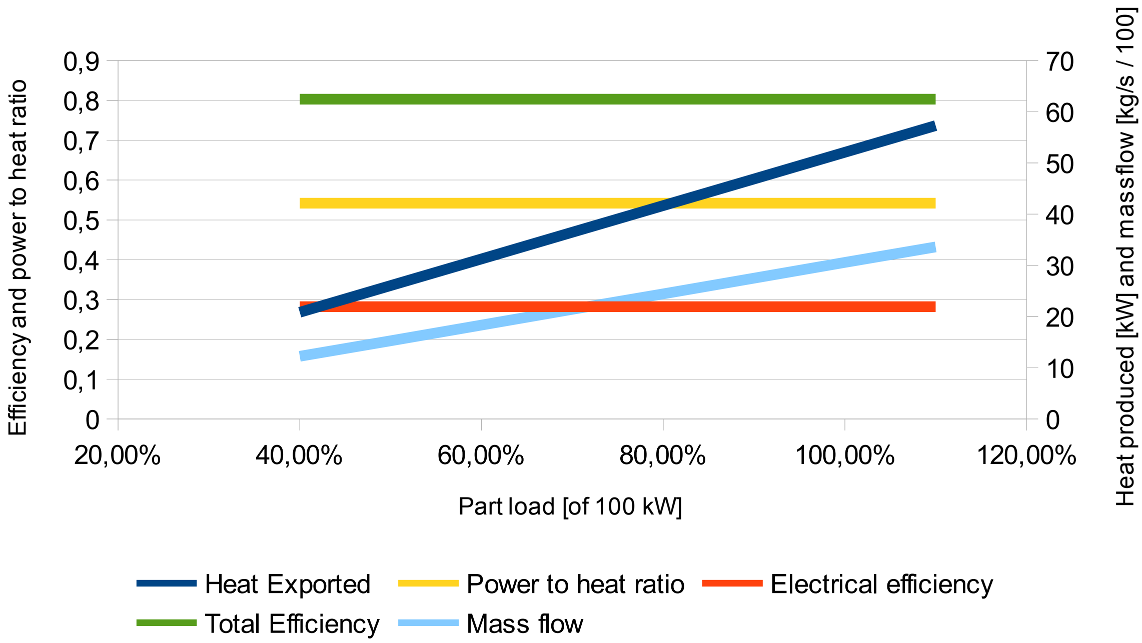

Chart 4: Part load performance of base case with constant TiT and varying mass flow

In the case of constant TiT, as seen in the above chart, electrical-, power-, and total efficiency is constant, while the heat and mass flow is dropping linearly. The mass flow in this chart is shown in 10^{-2} kg/s on the right axis. Since none of the factors affecting the turbines efficiency is varied, this is unsurprising. In reality the electrical efficiency would change, and likely drop, as the compressors and turbines efficiency would change caused by the volume flow and rotational speed deviating from the from design point. These straight lines are only a result of the lack of information about the compressor and turbine.

The total efficiency is in addition a function of the heat exchanger efficiency which do not change either, meaning that the total efficiency is also constant. Again this is caused by the simplifications done in our model. As previously discussed the heat transfer in a heat exchanger is a function of its size and its heat transfer coefficient, knowledge of witch would require sizing of the heat exchanger.

The main conclusion to be drawn from this chart is that proper prediction of part load performance based on varying the mass flow would require more detailed knowledge of the gas turbines components, as well as sizing of the heat exchangers.

Because of these limitations in our model, part load performance will be simulated by setting the mass flow as constant, and varying the TiT. This will provide a basis for a more qualitative comparison between our layouts. The results for this variation is shown below in chart 5.

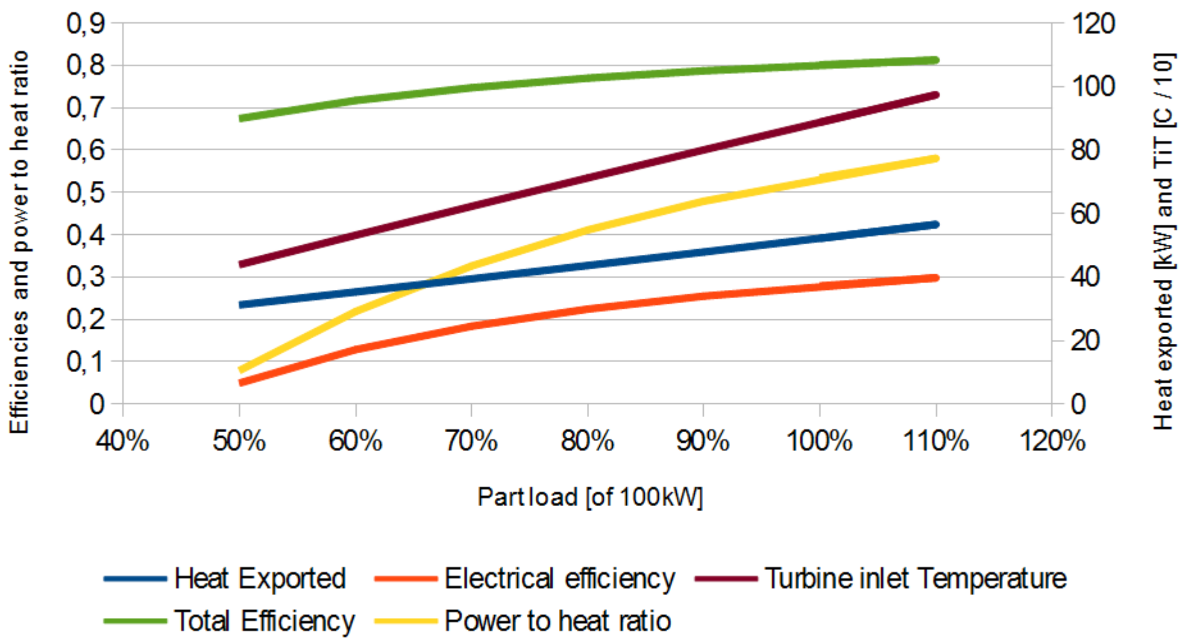

Chart 5: Part load performance of base case with varying mass flow and constant TiT

In chart 5 above, the TiT is plotted in 10^{-1} °C on the right axis. Again we see a linear variation in the heat exported, caused by our constant efficiency heat exchanger. The turbine inlet temperature is also linear, but this is function of the specific heat of the air changing little with the temperature over the ranges simulated, and not caused by any simplifications in our model.

More importantly we see the electric efficiency of the turbine dropping with increasing speed as the part load becomes lower. At some point before 40% part load it reaches zero, and the turbine is producing just enough power to drive the compressor. The total efficiency is dropping more slowly as the decrease in heat produced is not as rapid as the decrease in electrical efficiency.

16 Layouts

With regards to the systems layout the most obvious change is that from internal combustion to external combustion. In addition a export that exchanger needs to added to the system.

The two main points available to us for variation is the placement of the export heat exchanger, and the possibility of replacing the recuperator with simply feeding the hot air out of the turbine into the combustion chamber. Thus two layouts will be investigated, both with the turbine feeding the air into the combustion chamber but changing the placement of the export heat exchanger between them.

Heat Exchangers

Dispensing of the recuperator we are left with two heat exchangers. The export heat exchanger; a gas- liquid heat exchanger, and the gas heater; a gas-gas type. We retain the effectiveness of the recuperator to the gas heater which replaces it, as they should be very much alike. The gas heater will handle roughly the same mass flow, albeit at higher temperatures. The nature of the flue gases on the hot side of the gas heater will depend on the type of fuel burned, and we might expect some degradation in performance as deposits form on the outside of the gas heater. The effectiveness of the export heat exchanger is also retained in the further simulations.

Introducing the Combustion Chamber and Air Feedback.

As mentioned previously, since we have no flue gases out of the turbine, we have the option of using the hot air out of the cycle for combustion. This feedback of air can be viewed as a form of preheating the combustion air, and will in effect fulfill the same role as an recuperator. This layout is shown in illustration 30.

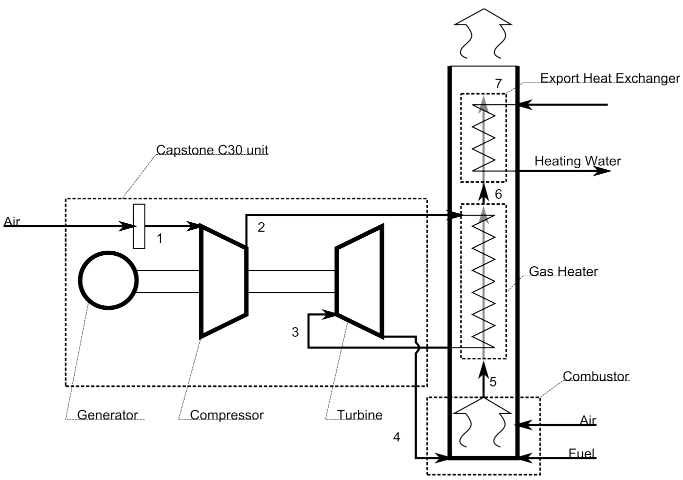

Illustration 30: Layout of CHP plant with air feedback into the combustor, and gas heate placed before export heat exchanger in the flue gases.

Shown below in illustration 31 is the cycle for this layout in a T-s diagram. The heat into the cycle is now added after the turbine, between point 4 and 5, and the heat addition between the compressor and turbine is now wholly an internal heat transfer in the cycle. This exchange happens from between point 5 and 6, and the heat removed between point 6 and 7 is to the export heat exchanger. The quantitative nature of the T-s diagram should be noted, as the air becomes flue gas between points 4 and 5, and changes properties.

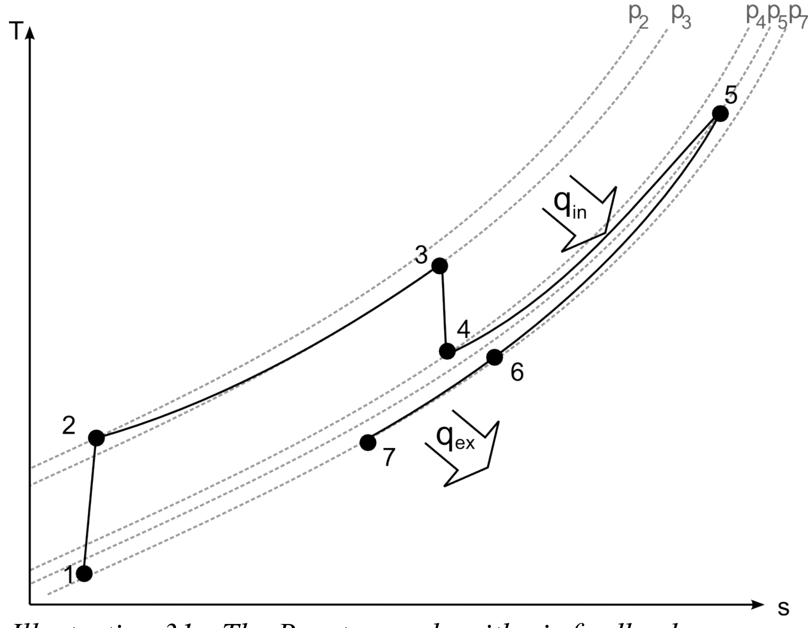

Illustration 31: The Brayton cycle with air feedback, gas heater and export heat exchanger

We have also now introduced the combustion chamber into the cycle, as shown between points 4 and 5 in the illustration. As we aim for fuel flexibility, we do not know what kind of fuel we are using. Additionally an air inlet has been introduced in the combustion chamber. The air inlet might be used to introduce cooling quickly, if the temperature of the flue gases exceed safe limits, as well as allowing the plant to run without the gas turbine, for pure heating.

From an efficiency point of view we would not want to add more air into the combustion chamber than is delivered trough the gas turbine. This is because the extra air would also need to be heated in the combustion chamber, lowering the temperature there, decreasing the efficiency of the turbine. Increasing the amount of air will also increase the amount of flue gases leaving the system, and with them increased amounts of lost heat.

To ensure that the air supplied is sufficient to ensure good combustion calculations where done on the basis of LHV and stoichiometric air fuel mixtures values from Energi Teknik by H. Alvarez, 2006. Looking at a worst case scenario air rate needed when wood with 20% water content was used as fuel was found. It was calculated that for 100 KW heat output 0.035 kg/s of air was needed, well below what is supplied by the gas turbine. This is not surprising as the amount of air that flows through a gas turbine with internal combustion is several times higher than what is needed for stoichiometric combustion of the fuel. The calculation for this is included in appendix B.

17 Layout 1: Gas Heater first

Modifying the gas turbine as described above, what will be called layout 1 is the case where we put the gas heater before the export heat exchanger in the flue gases. This is the layout and cycle shown in illustration 30 and 31. Turbine inlet temperature and pressure and will be kept the same as in the base case, but is it now the combustion chamber delivering the 100 kW of heat rather than the gas heater. For this and all following simulations fuel used is still pure methane.

Produced Electricity	19.2 kW
Electric Efficiency	19.2%
Exported Heat	59.4 kW
Total Efficiency	78.5 %
Power to heat ratio	0.32
mass flow at compressor inlet	0.22 kg/s

Table 10: Key values for layout 1

Comparing these values to the base case, we first see that we are producing less power. As the TiT remains the same, as does the pressures, this is caused by a reduction in mass flow. The reduction in mass flow is necessary to reach high enough temperatures in the flue gases to enable the 900ºC TiT. Perhaps more surprisingly this layout results in a marked increase in heat exported and total efficiency. This is caused by the increase in temperature at the inlet of the export heat exchanger and the higher specific heat of the flue gases.

The Turbine Inlet Temperature

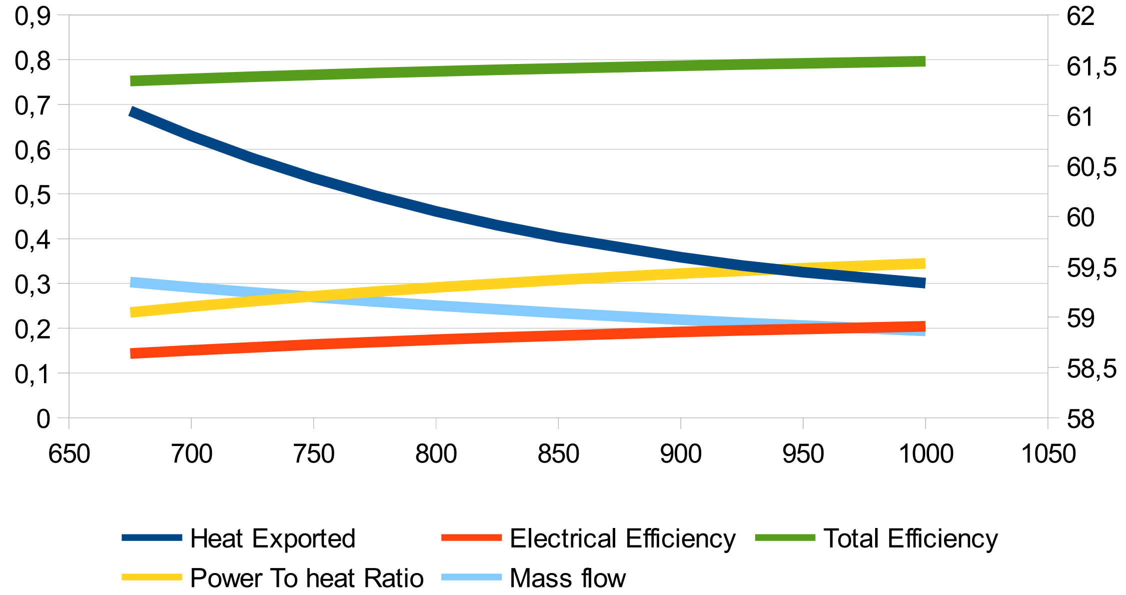

Chart 6: Power to heat ratio, mass flow, total- and electrical efficiency and heat exported, as function of the turbine inlet temperature for layout 1.

The results of changing TiT is a bit different for layout 1 than for the base case, and are shown above in chart 6. The general trends, i.e. increasing electrical efficiency, decreasing heat export and rising total efficiency, are present to differing degrees. The change in power to heat ratio is less, and is mainly caused by the drop in electrical efficiency, as the heat exported remains near constant. The conclusion is that a decrease in TiT from the original 900ºC would have a relatively small negative impact on the efficiency of system, and could be considered if needed.

Part Load Performance

The simulated part load performance of this layout is similar to that of the base case, and is shown in chart 7.

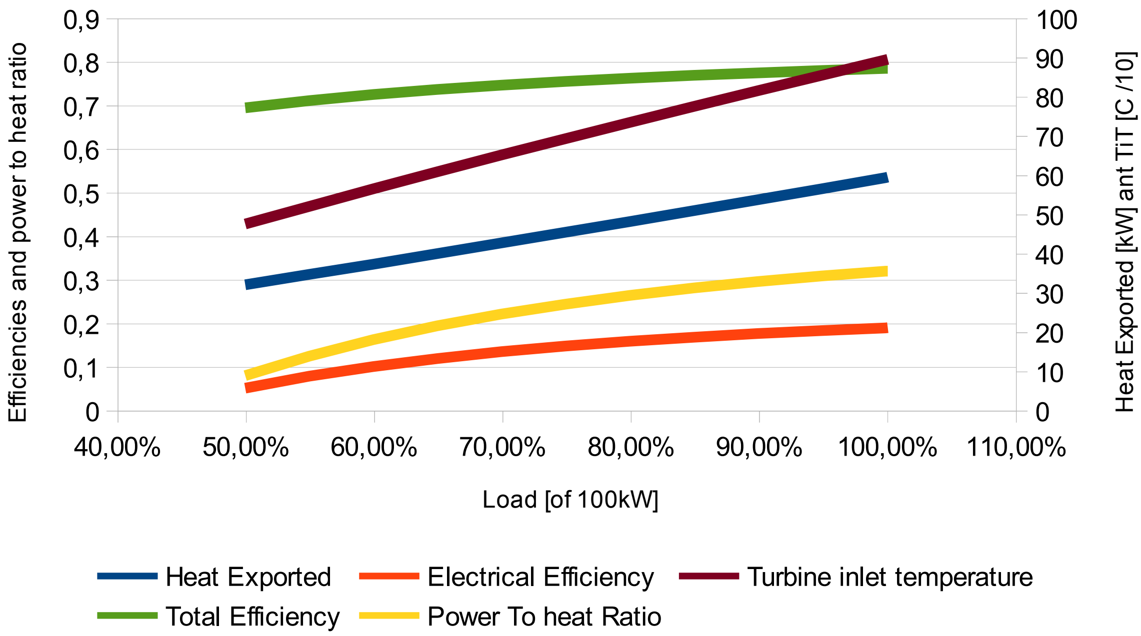

Chart 7: Part load performance with varying TiT and constant mass flow for Layout 1

All comments for the base case part load apply here, the main difference between layout 1 and the base case being that the decrease in total efficiency is less marked. This is caused by the drop in electrical efficiency being less marked, as in both cases the efficiency nears zero at 50% part load, but the starting point for this layout is lower. As the electrical efficiency goes below zero the generator will have to used as a engine, ensuring the compressor delivers enough mass flow to achieve good combustion, and avoiding the temperature of the flue gases becoming to high. Thus at this point the system will be drawing power and producing only heat.

18 Layout 2: Export heat exchanger first

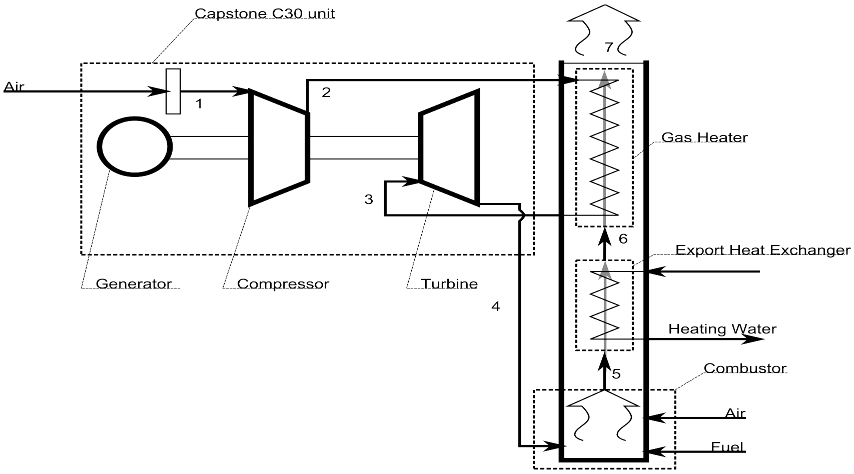

Illustration 32: Layout of CHP plant with air feedback, and export heat exchanger placed before gas heater in the flue gases.

In the case that high temperature is needed in the export heat exchanger, it could be placed in front of the gas heater in the flue gases. This layout is sohwn above in illustration 32 This will allow the export heat exchanger to exchange heat with the flue gases at much higher temperatures, useful for industrial or chemical processes. This layout will also let us control the heat input into the turbine by varying the amount of heat exported, as any heat not exported will reach the gas heater.

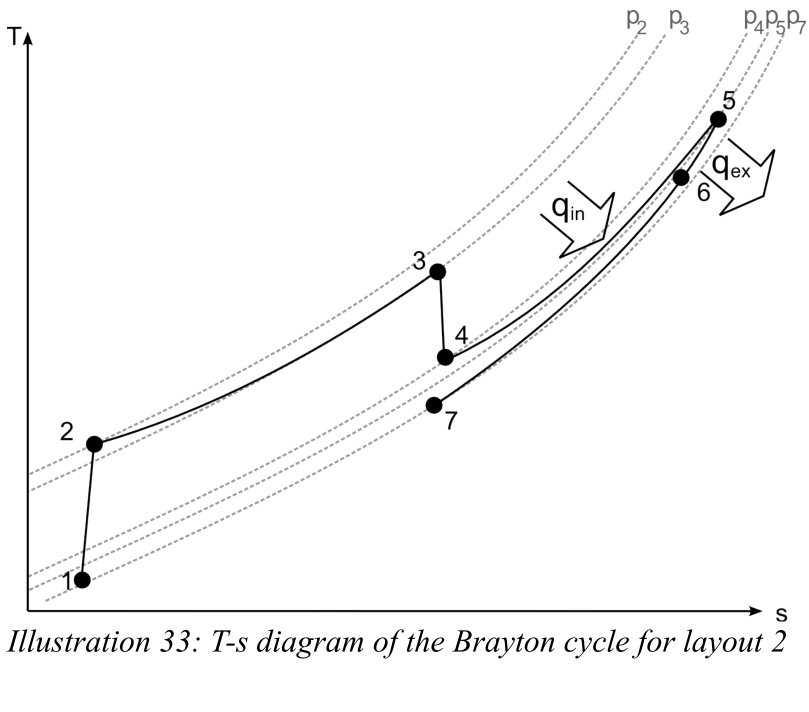

Illustration 33: T-s diagram of the Brayton cycle for layout 2

The cycle in the T-s diagram, shown in illustration 33, looks very similar, with the major change that the heat export now happens between point 5 and 6, and the internal heat transfer to the gas heater between 6 and 7. While not apparent in this diagram, points 5, 6 and 7 will be at a higher temperature in this layout than in the previous layout. That is the temperature out of the combustion chamber will be higher, as is the temperature of the flue gases between the heat exchangers and the temperature at the outlet of the cycle.

The Combustion Outlet Temperature

To simulate this layout some basic changes have to made to our simulation model in addition to the change in positioning of the export heat exchanger. Since the amount of heat into the gas heater is now dependent on the amount extracted beforehand in export heat exchanger, modeling the export heat exchanger with a set effectiveness is no longer possible, if we are to set the TiT. The model for the heat exchanger in the simulation was replaced with a simpler model, using only set temperature differences and not effectiveness. In this way we allow the heat exchanger to transfer all heat not needed to reach the target TiT. In addition, either the mass flow or the outlet temperature of the combustion chamber must be set. This is because the mass flow is no longer only a function of the temperature needed to reach the TiT in the heat exchange in the gas heater. To decide this the simulation was run with the mass flow being a function of the combustion chamber outlet temperature, and the results are shown below in chart 8.

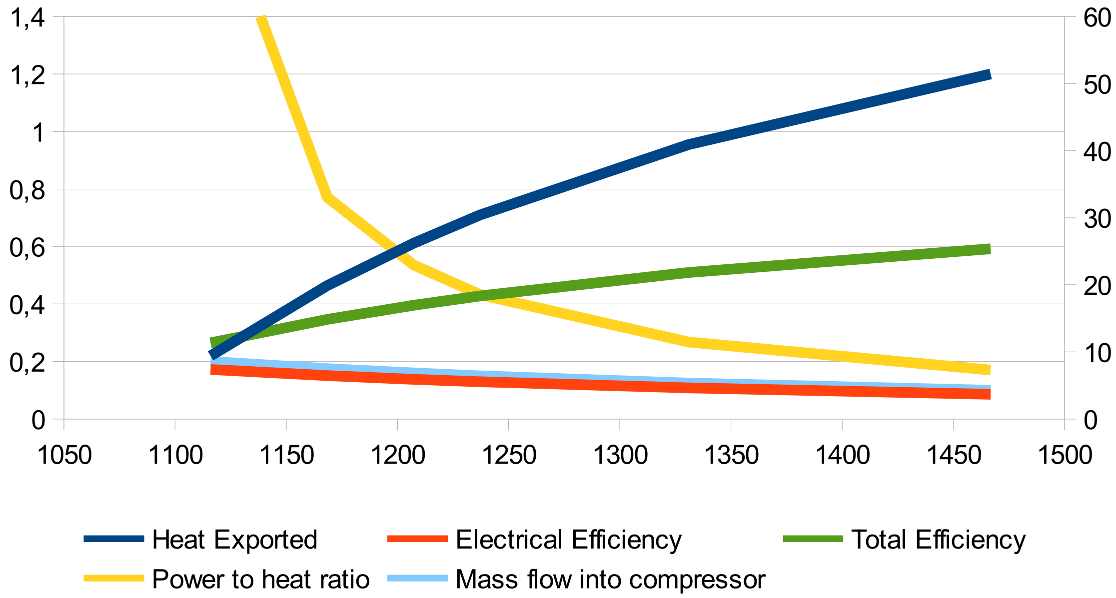

Chart 8: Efficiencies, heat exported, power to heat ratio and mass flow as function of comustion chamber outlet temperature

The cutoff of the lines at around 1125ºC is caused by the temperature of the flue gases no longer being high enough for the gas heater to reach the TiT. As the temperature increases the heat exported increases, since there is more heat available to extract, but the electrical efficiency goes down since the turbine can extract less work from the decreasing mass flow. The net effect is that the total efficiency rises with increasing temperature. For the further simulations the combustion chamber outlet temperature has been set at 1200ºC, deemed to be a compromise between total efficiency and the electrical efficiency. Running the simulation with a combustor outlet temperature of 1200ºC at 100kW load resultet in the data presented in table 11 below.

Produced Electricity	14.3 kW
Electric Efficiency	14.3 %
Exported Heat	25.1 kW
Total Efficiency	39.4%
Power to heat ratio	0.57
Mass flow into compressor	0.16 kg/s

Table 11: Key values for layout 2

What immediately becomes apparent is the poor performance of the system, both electric and total efficiency is significantly poorer than for layout 1. Because the heat exchanger that operates with the highest cold side temperature comes last, the flue gases leaving the system have higher temperatures than in the previous case, and thus more heat is lost to the environment. Costea et al. in 2012 simulated a gas turbine with internal combustion and no feedback, where gas turbine layouts where evaluated with regards to the relative positioning of the recuperator and export heat exchanger. These results are in accordance with what was found in that study; significantly lowered total and electric efficiency when heat export happened before heat transfer into the cycle. Thus such a layout should only be considered if there is a need for high temperature of the exported heat, and would need to evaluated against other alternative solutions for creating the heat. Whether this layout would be economical advantageous in comparison to a simple boiler is uncertain.

Again the effect of varying the TiT has been simulated and the results are shown below in chart 9.

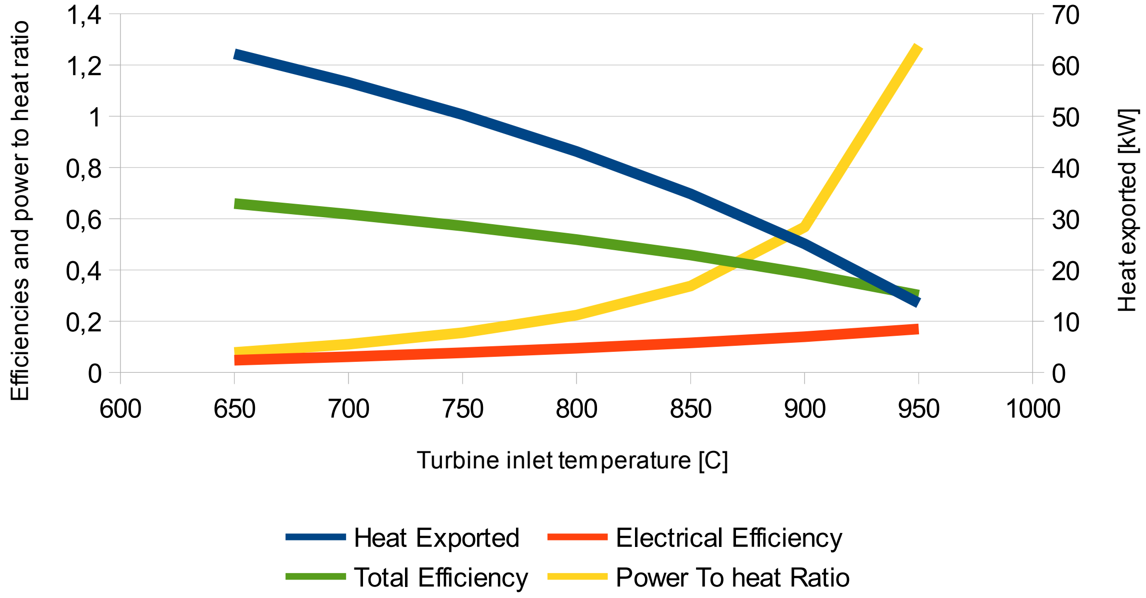

Chart 9: Power to heat ratio, total- and electrical efficiency and heat exported, as function of the turbine inlet temperature for layout 2. Heat exported is plotted against the left y-axis.

In contrast with layout 1, in this case the amount of heat exported and the systems total efficiency increases as the TiT lowers, and this has a drastic effect on the power to heat ratio. As metioned the simulation is set up in such a way that all heat not necessary to reach the TiT is removed from the flue gases before the gas heater. Since the heat input is set, this is what is causing the amount of heat exported to increase so rapidly with the lowering of the TiT.

It should be noted that even at an TiT as low as 650ºC, the amount of heat exported is no more than for layout 1 with a TiT of 900ºC, and the total efficiency is lower. The electrical efficiency is also so low, that one could question the inclusion of the gas turbine into the cycle at all. It is clear that in this layout a reduction in the TiT could be considered as a means to increase the heat exported.

Variation In the Exported Heat

One of the of advantages of this layout is that we can vary the power produced by changing the amount of heat exported. The result of lowering the amount of heat exported should result in larger amounts of heat transfered in the gas heater. This is an possibility not really available in the first layout, as there is no control over the amount of heat that gets transferred from the flue gases into the gas turbine, as long as the TiT and effectiveness of the gas heater is set.

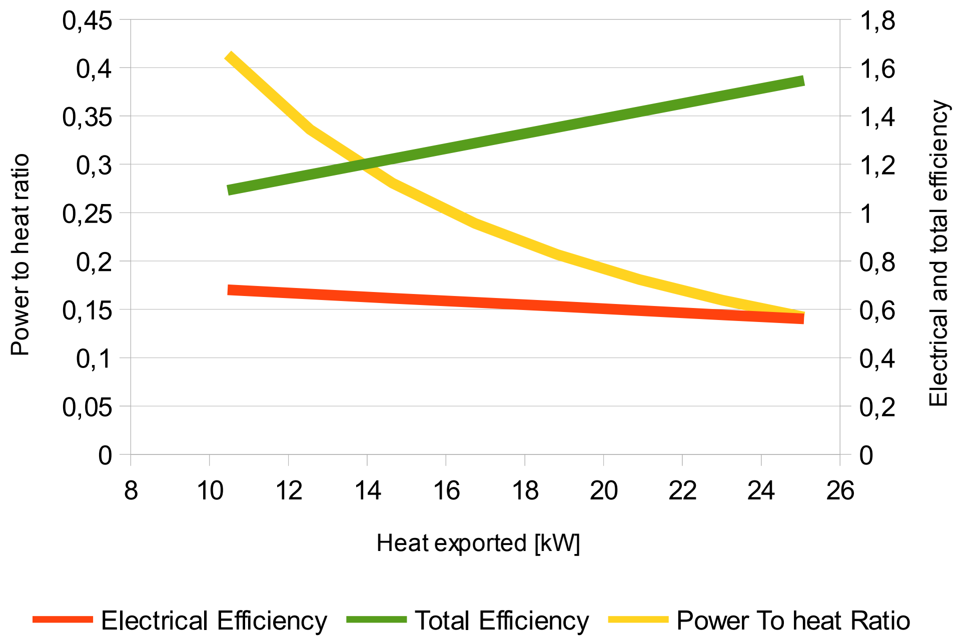

Chart 10: Electrical and total efficiency, and power to heat ratio as function of exported heat in layout 2

Simulating the variation in exported heat yields the results shown in chart 10, where efficiencies and power to heat ratio is plotted as a function of the amount of heat exported. As the amount of heat exported is reduced, the efficiency, and thus amount of electricity produced increases, as expected. The drawback of this being that the total efficiency is lowered, as flue gases leave the system at higher temperatures, and more heat is lost to the environment.

Part Load Performance

Since for part load performance we wish to have the TiT as a variable, we can no longer use the simplification of the heat exchanger extracting all heat not needed to reach TiT. For the purpose of simulating the part load performance the heat exchanger model was changed to include effectiveness. The effectiveness was then found to be 10.6%, by letting it be a variable while running the simulation for normal heat load. The part load performance was then simulated by letting the TiT be a variable and setting the mass flow constant. The outlet temperature of the flue gasses is now a variable, to allow for the setting of the mass flow. The result is shown below in chart 11.

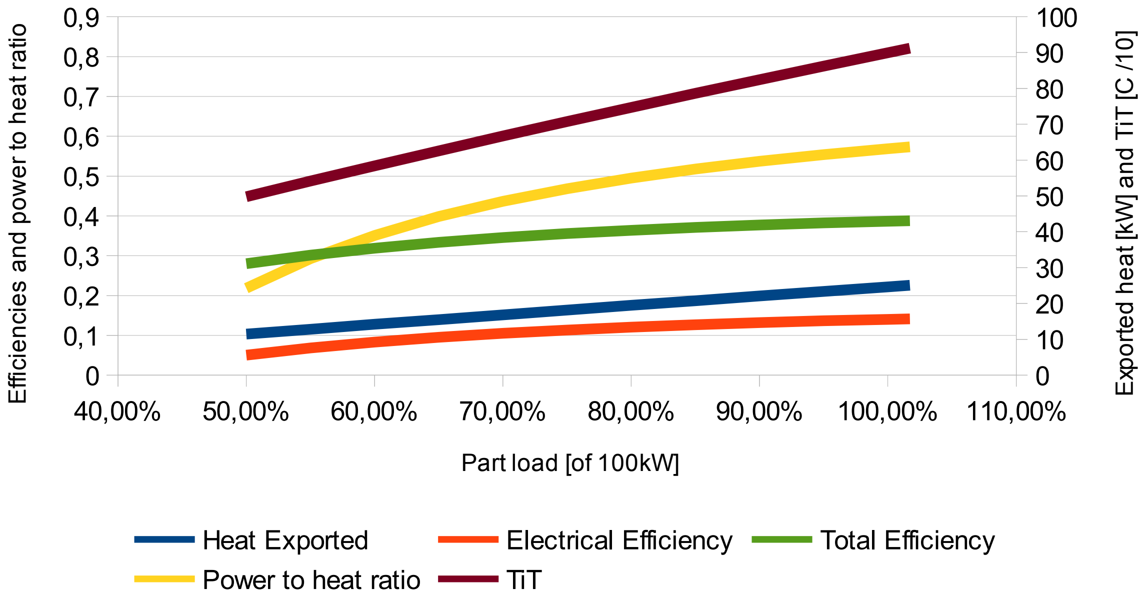

Chart 11: Part load performance with varying TiT and constant mass flow for Layout 2

In comparison to layout 1 the performance of layout 2 is similar, albeit with marked lower efficiencies overall. The drop in total efficiency at 50% load is around 10% which is comparable to layout 1. Interestingly the drop off in electrical efficiency is less than for layout 1, starting at lower efficiency and ending up at around the same place, around 5%. The conclusion to be drawn is that there is little difference in relative part load performance between the two systems.

19 Operational Control and Sensor Placement

The CHP plant will have to be connected by an energy control system able to adjust the systems output based on electrical and heat needs. Adjusting the systems outputs will be a function of the inputs, namely the flow of air and heat.

Controlling the amount of heat input into the system will be a function of the mass flow of fuel, assumed to be supplied to be an automatic feeding system. This feeding system will need to be controlled by a central control system. Since we wish fuel flexibility the heating value of the fuel will be unknown and variable, and the system will have to evaluate this continually. There is no simple way to measure the heating value and it will have to be calculated from other measured properties in the system. Thus we would need to calculate the fuel heating value, to be able to adjust the fuel flow according to the load. In short, we would wish to know be able to “solve” the system based on sensor input.

Knowledge of the heat output, electrical output and mass flows of air and fuel should be easily obtained, based on the C30s integrated control system, and knowledge of the water flow in the the export heat exchanger. Coupled with the export heat exchanger, gas heater, and the turbine units efficiency, only the outlet air temperature of the flue gases would need to be known to have full system knowledge. This temperature is around 140ºC for layout one, which should not be to high for a temperature sensor of reasonable price and life span, although this temperature might increase as the heat exchangers drop in effectiveness with time. For layout 2 however, this temperature is at around 380ºC, which would mean that the sensor would need to be considerably more expensive.

The efficiencies will change over time, as the turbine and compressor wears and the heat exchangers become dirty, meaning that a control system based upon extrapolation from these efficiencies will become more inaccurate over time. Thus regular maintenance of the will be of importance for the accuracy of the control system.

20 Conclusion to Part Two

The GT30 can, with modifications, serve as the base for a CHP plant with comparable total efficiency of larger plants, but with poor power to heat ratio. Had ORC or SRC been chosen, these problem of low power to heat ratio would likely have been exaggerated due to lower electrical efficiency, so the choice of a gas turbine for power generation seems validated.

Looking at the alternative layouts it is clear that the export heat exchanger should be positioned after the gas heater to achieve the highest electrical efficiency. If high temperatures are required, a CHP plant based upon layout 2 might be considered but the total efficiency is so low that a simple boiler might be preferable unless the need for power is critical.

When modifying the turbine the highest achievable TiT should be chosen as a design point for layout 1, as this results in the highest electrical efficiency. For layout 2 a lower TiT might be considered as a way to increase the amount of heat exported, but this has a negative affect on the already low electrical efficiency. Part load performance of layout 1 seems adequate achieving a total efficiency of 70% at half load, although this is likely an overestimation, since change in turbine and compressor efficiency was not considered. If one where to modify turbine and compressor to fit a new mass flow, the performance of these parts under different flow speeds should be taken into consideration, and the part load efficiencies re-evaluated.

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Appendix A: Simulation models

Models created in DNA take the form of code, written in a specific language for the program to runs. What follows is a short explanation of how these models should be read, and then the models themselves are presented. The models are preceded by a flowchart showing the components in the models and their connected nodes with numbers. There has been no additional formatting done to the simulation text so that they can be directly copied into a DNA file from a digital version of the text to recreate the simulations.

All illustrations and code made by the author.

How to read DNA text

The first thing to note is that all lines preceded by a "C" is commentary added by the author, and is not read by the simulation program, but is rather intended to keep the simulation file understandable, and to help the readability for someone not familiar with DNA. All simulations also start with **TITLE** and is simply the title of the simulation witch also does not affect the simulation itself.

DNA simulations are constructed as structures placed between nodes. These nodes are numbered in the single digits, i.e. 1,2,3.. ,for mass flows, in the hundreds for mechanical energy, two hundreds for electricity and three hundreds for heat flows.

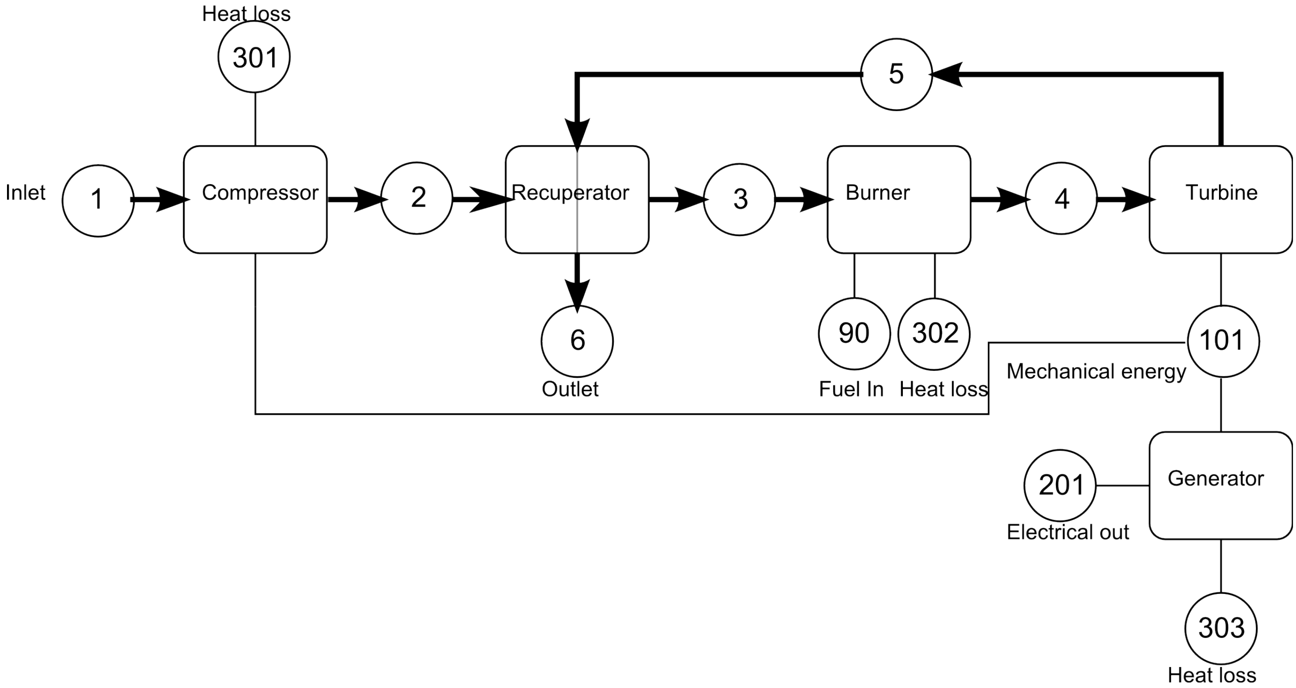
All lines starting with **STRUC** is an element between two or more nodes somehow affecting the flows(s). The word **STRUC** is followed by a name, chosen freely, used to refer to the element later, and the name is followed by the type of element. For example **STRUC** Compressor COMPRE_1 would add an element called Compressor of the type COMPRE_1 into the simulation. After the type of structure is written it is followed by a number of nodes and then by a number of parameters. The number of nodes and types of parameters varies greatly with the type of structure, and is explained roughly in the commentary.

Lines starting with the word **ADDCO** adds physical conditions to the flows in the nodes that are the basis for the simulation. The word **ADDCO** is followed by the property to be set, and follows standard nomenclature, with Q for heat flow in kW, T for temperature and so on. After the property is set it is usually followed by the name of the component that the flow has as a positive direction and the number of the node that the is from. The exception is pressure, as it only needs to set for the node.

Lines beginning with **START** are starting guesses for the simulation to run on, and are basically ballpark guesses.

The **VARPA** command frees a parameter for a structure as a variable. **VARPA** is followed by the name of an element and the number of the variable that is to be freed. E.g. **VARPA** Turbine 2 frees the second parameter of the structure «turbine» as a variable. The name and number needs to followed by an additional condition, as there is now an additional variable in the simulation.

Recreation of CU Gas Turbine




```
TITLE CU_GasTurbine

STRUC Compressor COMPRE_1 1 2 301 101 .75 0.99
C Compressor between nodes 1 and 2, heatloss through 301,
C mech. energy through 101. Last numbers are Isentropic efficiency
C and mech. efficiency

C STRUC Compressor COMPRE_3 1 2 301 101 .805 100 .99
C Compressor with polytropic efficiency, used when calculating the
C Polytropic efficiency by replacing the isentropic compressor.
C The three last numbers are respectively polytropic efficiency,
C numbers of steps in integration for the calculation of properties
C and the mechanical efficiency

MEDIA 1 STANDARD_AIR
C The media through node 1 is standard air, as defined by DNA and
C modeled as an ideal gas.

STRUC Turbine TURBIN_1 4 5 101 .82
C Turbine between nodes 4 and 5. Mechanical aenergy to node 101.
C Last number is isentropic efficiency


C STRUC Turbine TURBIN_3 4 5 101 .778 10
C Turbine with polytropic efficiency, used when calculating the
C Polytropic efficiency by replacing the isentropic turbine.
C The three last numbers are respectively polytropic efficiency,
C numbers of steps in integration for the calculation of properties
C and the mechanical efficiency
```

STRUC Generator SIM_GENE 201 303 101 0.98
C Generator. Recieves mechanical energy through node 101, and delivers
C power through node 201. Hetloss through node 303. Last number is efficiency

STRUC Recuperator HEATEX_4 5 6 2 3 304 .825 0.05 0.05
C Heat exchanger with hot side between nodes 5 and 6 and cold side between 2
C and 3. Heatloss through node 304. Last three numbers are effectiveness,
C and absolute pressure loss hot and cold side in bars.

STRUC Burner GASBUR_2 3 90 4 302 756 .99
C Burner between nodes 3 and 4 with fuel in through node 90. Heatloss
C through node 302. Last numbers are combustion temperature and the
c relative pressure loss over the burner.

MEDIA 90 METHANE 4 FlueGas
C the media through node4, after the burner, is flue gas.

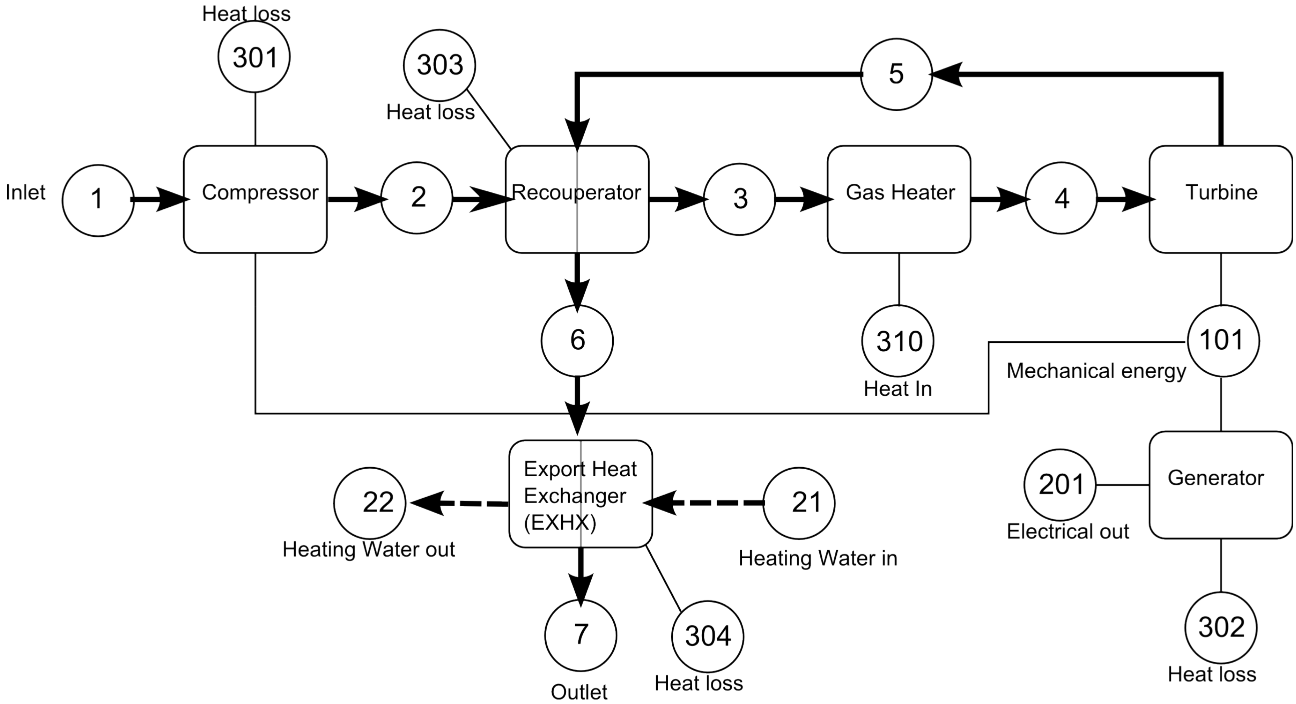
ADDCO Q Burner 302 0 T Burner 90 25 P 90 20
ADDCO Q Recuperator 304 0
ADDCO P 6 0.83
ADDCO P 1 0.83 T Compressor 1 20 M Compressor 1 0.17 P 2 1.992

START Y_J FlueGas O2 .1 Y_J FlueGas N2 .9
START M Burner 90 .1 T Burner 4 764
START P 5 1.0
START T Turbine 5 600
START T Compressor 2 130 Q Compressor 301 0
START T Recuperator 3 500
START Q Generator 303 -5

C VARPA Turbine 1 T Turbine 5 605.63
C This frees the first property of the turbine, the polytropic efficiency
C as a variable and sets the outlet temperature of the turbine as a
C constant instead. The temperature is gotten by running the simulation
C with isentropic efficiency set.

C VARPA Compressor 1 T Compressor 2 130.49
C This frees the first property of the compressor, the polytropic efficiency
C as a variable and sets the outlet temperature of the com pressor as a
C constant instead. The temperature is from a previuos simulation with
C isentropic efficiency set.

Base Case



TITLE BaseCase
STRUC Compressor COMPRE_3 1 2 301 101 0.778 100 0.99

C Inlet compressor. Mechanical energy in through point 301 which is shared

C between the compressor, turbine and generator.

C Excess heat out through 301.

C Last numbers are Plytropic eff, steps for iteration of pol. eff and Mech

C eff

STRUC Recup HEATEX_4 5 6 2 3 303 0.825 0.05 0.05

C Recouperator. Heat loss through 304. Hot side is between points 5 and 6

C Cold side between points 2 and 3

C Last three numbers are effectiveness, and absolute pressure loss hot

C and cold side

STRUC GasHeater HEATSRC0 3 4 310 0

C Heatsource where heat is added from point 302. Last number is pressure

C loss in bars

STRUC Turbine TURBIN_3 4 5 101 0.805 100

C Turbine. Mech energy out through 101 Last number is poly. eff. and steps

C for iteration for poly. eff.

STRUC Generator SIM_GENE 201 302 101 1
C Generator. El out 201. Heat out 303. Mech in 101. Last number is el eff.

STRUC EXHX HEATEX_4 6 7 21 22 304 0.85 0 0
C Export Heat Exchanger. The water on the cold side comes in at node 21 and
C out through node 22. Last numbers; effectiveness, and abs. pressure drop
C hot and cold side

MEDIA 1 STANDARD_AIR
C Medium in the cycle is standard air as defined by DNA.

MEDIA 21 STEAM
C Medium in the export heat exchanger cold side is water. STEAM is used as
C DNA has no WATER medium, but STEAM covers both the liquid and gas phas of water.

ADDCO Q Recup 303 0 Q GasHeater 310 100 Q EXHX 304 0
C The system is without heat loss, with generator as exception.

ADDCO T Compressor 1 21.7 T EXHX 22 75 T EXHX 21 55
C The temperatures in the system

ADDCO T Turbine 4 900
C Turbine inlet temperature

C ADDCO M Compressor 1 0.31
C Massflow into compressor. Used when simulating partial load

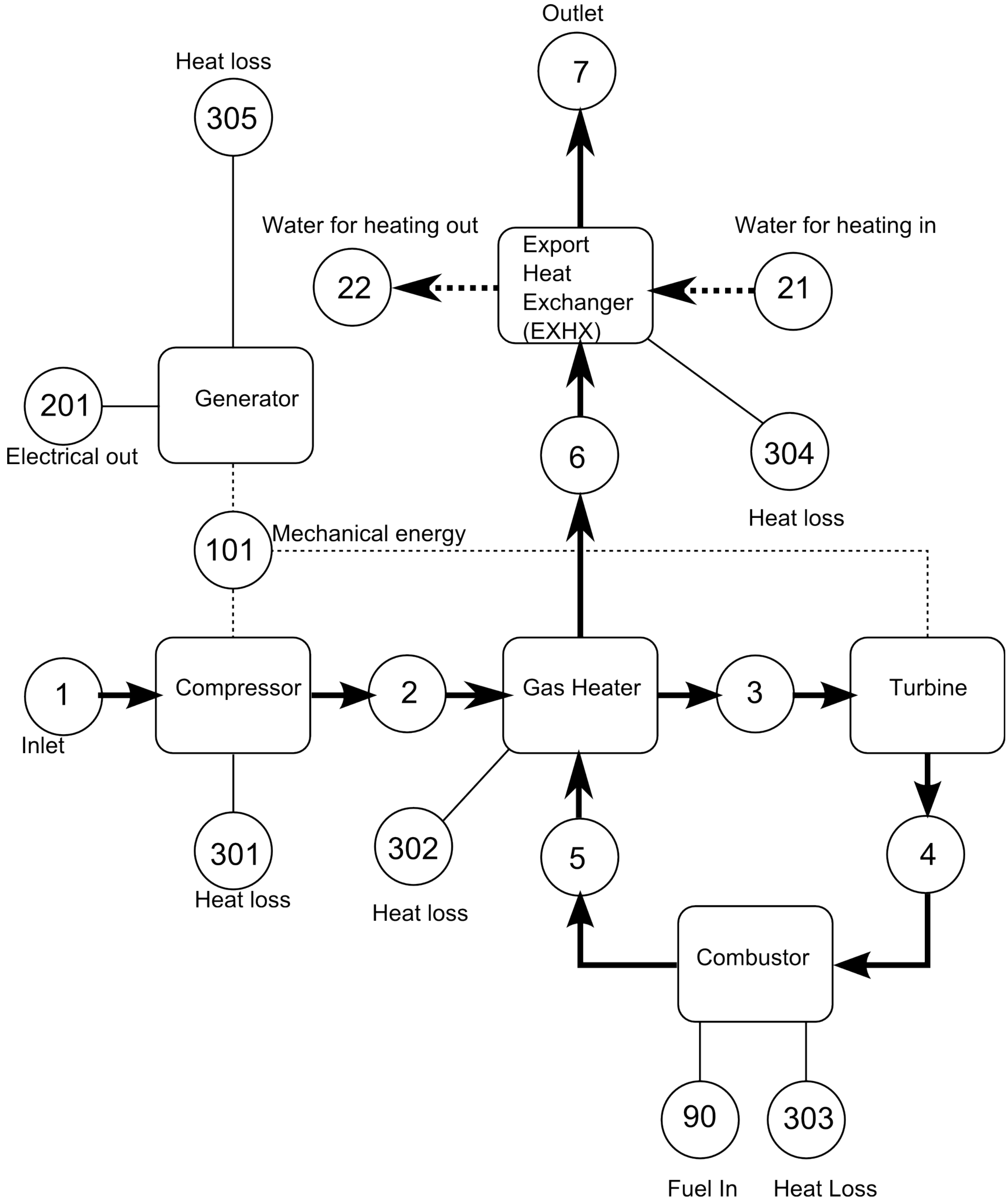
ADDCO P 1 1.0132 P 6 1.0132 P 4 2.4817 P 22 1
C Pressures. Note that the pressure ratio is simulated as being constant
C over the turbine.

START T Recup 5 725 W Compressor 101 35 W Generator 101 27
START E Generator 201 27 T Recup 6 250 M Compressor 1 0.31

START T EXHX 7 300 M EXHX 21 1

C Start values to get the simulation running.

Layout 1



TITLE Case1

STRUC Compressor COMPRE_3 1 2 301 101 0.778 100 0.99

C Inlet compressor. Mechanical energy in through point 301 which is shared

C between the compressor, turbine and generator.

C Excess heat out through 301.

C Last numbers are Isentropic eff and Mech eff

STRUC GasHeater HEATEX_4 5 6 2 3 302 0.81 0.05 0.05

C Gas heater. Heat loss to node 302. Last number is pressure

C loss in bars

STRUC Turbine TURBIN_3 3 4 101 0.805 100

C Turbine. Mech energy out through 101 Last number is isotropic eff

STRUC Combustor GASBUR_2 4 90 5 303 1200 .98

MEDIA 90 METHANE 5 FlueGas

C Combustor betwee nodes 4 and 5. Fuel in at node 90. Last numbers are

C Combustion temperature, and combustion efficiency.

STRUC Generator SIM_GENE 201 305 101 1

C Generator. El out 201. Heat out 303. Mech in 101. Last number is el eff.

MEDIA 1 STANDARD_AIR

C Medium in the cycle is standard air as defined by DNA

STRUC EXHX HEATEX_4 6 7 21 22 304 0.81 0 0

C Export Heat Exchanger. Water in at node 21 and out at 22. Last numbers

C are effectiveness and pressure losses hot and cold side

MEDIA 21 STEAM

C Water running throug export heat exchanger. Defined as steam, but DNA

c handles both phases when media is defined as STEAM

C ADDCO M Compressor 1 0.219

C Massflow into compressor

ADDCO M Combustor 90 0.0020034

C Massflow fuel

ADDCO T Compressor 1 21.7 T Combustor 90 25
C Temperatures

ADDCO P 1 1.0132 P 3 2.48168 P 7 1.0132 P 90 1
C Pressures

ADDCO Q Combustor 303 0 Q EXHX 304 0 Q GasHeater 302 0
C No het loss except for generator

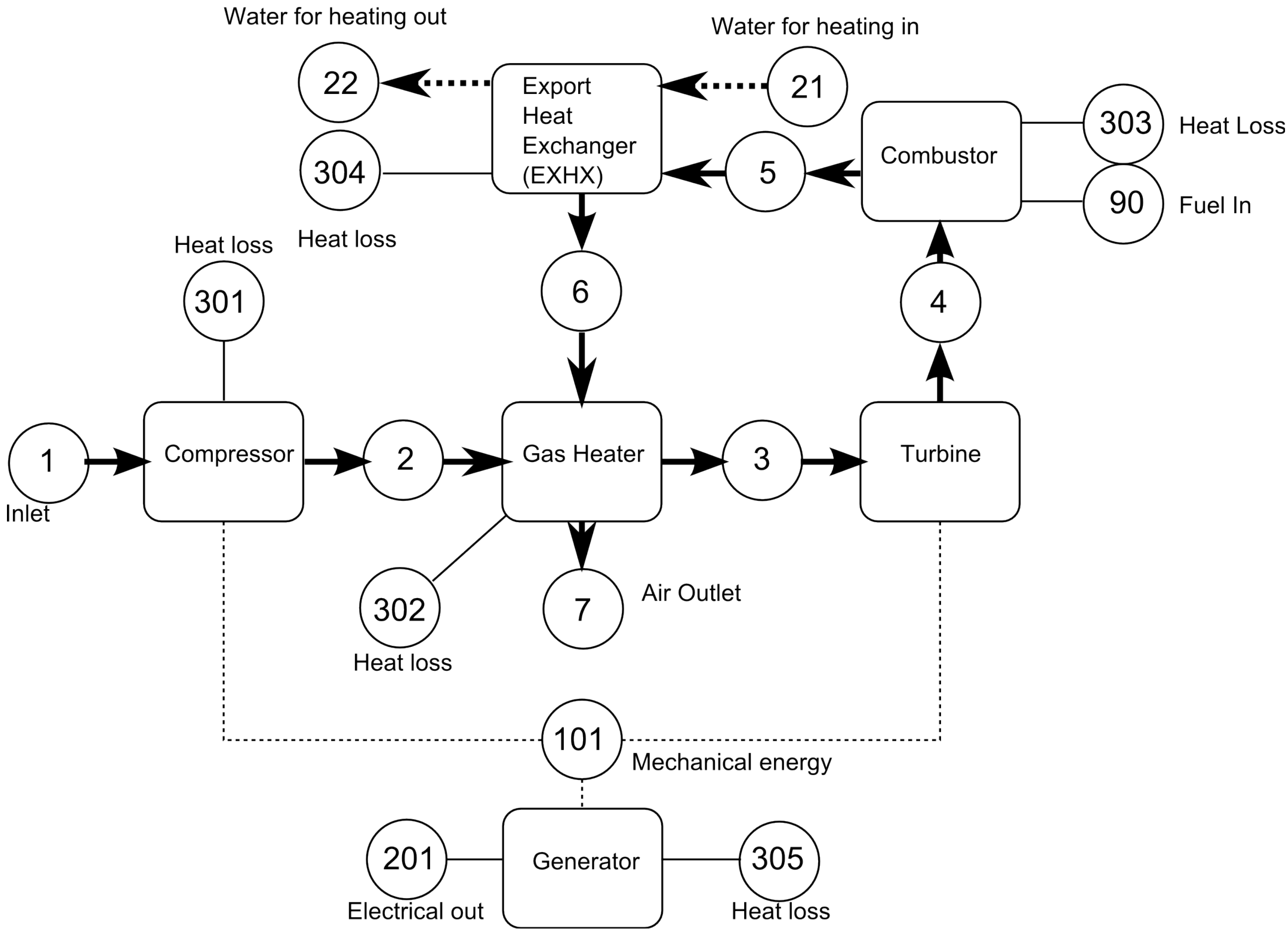
ADDCO T EXHX 22 75 T EXHX 21 55 P 21 1.1
C Settings for Export heat exchanger

START Y_J FlueGas O2 .1 Y_J FlueGas N2 .9 M Compressor 1 0.31
START M Combustor 90 .05 T Combustor 5 1500
START T Turbine 4 800 W Compressor 101 20 W Generator 101 30
START E Generator 201 20 T GasHeater 6 100 Q Compressor 301 0
START M EXHX 21 1 T EXHX 21 50
C Start values to get the simulation running.

VARPA Combustor 1 T Turbine 3 900
C releasing the combustion temperature as a variable, and setting the
C turbine inlet temperature.

C VARPA Combustor 1 M Compressor 1 0.22
C Releasing combustion temperature and setting massflow to what was found
C at 900C TiT. Used to simulate partial loads.

Layout 2



TITLE Case2

STRUC Compressor COMPRE_3 1 2 301 101 0.778 100 0.99

C Inlet compressor. Mechanical energy in through point 301 which is shared

C between the compressor, turbine and generator.

C Excess heat out through 301.

C Last numbers are Isentropic eff and Mech eff

STRUC GasHeater HEATEX_4 6 7 2 3 302 0.81 0.05 0.05

C Gas heater. Heat loss to node 302. Last number is pressure

C loss in bars

STRUC Turbine TURBIN_3 3 4 101 0.805 100

C Turbine. Mech energy out through 101 Last number is isotropic eff

STRUC Combustor GASBUR_2 4 90 5 303 1200 .98

MEDIA 90 METHANE 5 FlueGas
C Combustor betwee nodes 4 and 5. Fuel in at node 90. Last numbers are
C Combustion temperature, and combustion efficiency.

STRUC Generator SIM_GENE 201 305 101 1
C Generator. El out 201. Heat out 303. Mech in 101. Last number is el eff.
MEDIA 1 STANDARD_AIR
C Medium in the cycle is standard air as defined by DNA

STRUC EXHX HEATEX_1 5 6 21 22 304 0 0
C Simple heat exchanger model last numbers are pressure losses over hot and
C cold side.

C STRUC EXHX HEATEX_4 5 6 21 22 304 0.106 0 0
C Export Heat Exchanger with effectiveness. Last numbers are effectiveness, and pressure loss,
C hot and cold side. Used for part load simulation

MEDIA 21 STEAM
C Water running throug export heat exchanger. Defined as steam, but DNA
c handles both phases when media is defined as STEAM

ADDCO Q GasHeater 302 0 Q Combustor 303 0
C No heat losses in gas heater or combustor

ADDCO T Compressor 1 21.7 T Combustor 90 25
C Temperatures

ADDCO T Turbine 3 900
C TiT

C ADDCO M Compressor 1 0.16
C Massflow into compressor. Used for part load simulation

ADDCO M Combustor 90 0.002034
C Massflow of fuel to set heat input

```
ADDCO P 1 1.0132 P 3 2.48168 P 7 1.0132 P 90 20
C Pressures, last pressure is fuel pressure

ADDCO Q EXHX 304 0 T EXHX 22 75 T EXHX 21 55 P 21 1.1
C Export Heat Exchanger settings

START M EXHX 21 1 T EXHX 21 50
START Y_J FlueGas O2 .1 Y_J FlueGas N2 .9 M Compressor 1 0.2
START M Combustor 90 .1 T Combustor 5 1200
START T Turbine 4 700 W Compressor 101 20 W Generator 101 30
START E Generator 201 30 T GasHeater 7 600 Q Compressor 301 5
C Start values to get the simulation running.

C VARPA EXHX 1 M Compressor 1 0.16
C releasing the EXHX effectiveness as a varaible and setting the massflow
C Used for finding the EXHX effectiveness at nominal load

C VARPA Combustor 1 M Compressor 1 0.16
C Releasing the combustor outlet temperature and setting the compressor
C Inlet massflow. Used for simulating part load
```

Appendix B: Calculations

Calculations for thermoelectric generators and cells for table 2

Given a 5% efficiency, and 100 kW input, assuming all remaining heat is used for heating, and negligible losses elsewhere. Given the workings of the technology, the assumption is justified to a large degree.:

$$C_{CHP} = \frac{w_{\text{cycle,el}}}{q_{\text{in}}} = \frac{(\eta_{\text{cycle,el}} * q_{\text{in}})}{q_{\text{in}}} = \eta_{\text{cycle,el}}$$

TEG1-12611-6.0 14.7 kW Thermoelectric module from TECTEG mfg.

Power: 14.7 W_{el} Heat flux through the module: ~350W (q_{out})
Size: (Cell) 5.6 * 5.6 * 4.5 [cm]
Cost: $48.00 U.S. For purchases over 100 modules.

Specific Cost: ($48 * 0.76 €/$) / 0.0147 kW_{el} = 2482 €/kW_{el}

$$\eta_{\text{cell, el}} = \frac{w_{\text{cell,el}}}{q_{\text{in, cell}}} = \frac{w_{\text{cell,el}}}{q_{\text{out, cell}} + w_{\text{cell,el}}} = \frac{14{,}7\, W_{el}}{350\, W + 14{,}7\, W_{el}} = 4\%$$

Source: http://www.thermoelectric-generator.com/

Ultra high Temperature Thermo Electric Generator Cell from Goat Industries

Power: 20.9 W_{el} Nominal heat flux: 13.3 W/cm^2 (not specified if q_{in} or q_{out}, asuming q_{out})
Cost: £49.00

Specific cost: (£49*1.18 €/£) / 0,0209 kW_{el} = 2767 €/kW_{el}

$\eta_{cell,el}$ = (5.6 cm * 5.6 cm) * 13.3 W/cm^2 / 14,7 W_{el} = 5%

Source: http://www.thermoelectricgenerator.co.uk/

TEG integrated into fluid to fluid heat exchanger:

200 Watt Fluid to Fluid Heat Transfer TEG
Power: 200 W
Cost: $1519.99 U.S

Specific Cost: ($1519.99 * 0.76 €/$) / 0.2 kW_{el} = 5776 €/kW_{el}

Not enough data to calculate any $\eta_{\text{cycle,el}}$

Source: http://tegpower.com/

Calculations for Stirling engines for use in table 2

SOLO stirling engine
Power: $\eta_{cycle,el} = 24\%$ $\eta_{cycle,th} = 72\%$ $P_{el}= 2 – 9.5$ kW P_{th}=8-26kW
Size(L*W*H): 1.28*0.7*0.98 [m]
Cost: approx 25.000 €

Using the maximum values for thermal and electrical power, as the power input from our sytem is larger than required for one engine.

C_{CHP}= 9.5kW$_{el}$ / 26kW$_{th}$= 0.37
Specific footprint: (1.28m * 0.7m) / 9.5kW$_{el}$= 0.09 m^2 / kW$_{el}$
Specific cost: 25000 € / 9.5kW$_{el}$ = 2631 € / kW$_{eL}$

[based on a flow temperature of 50°C in the heating system]

Source: http://solarfreunde-moosburg.de/solarfreunde/mails/SOLO_STIRLING_g.pdf and http://www.chp-goes-green.info/sites/default/files/SOLO_Stirling_161.pdf

United Stirling SPS V-160
Power: $\eta_{cycle,el} = 33\%$ $\eta_{cycle,th} = 44.9\%$ $P_{el}= 11.4$ kW P_{th}=15.5 kW
Size(L*W*H): not found
Cost: not found

Mentioned in [Dentice d'Accadia et al., 2003] as market driver. Search for it online resulted in little information. Producer United Stirling was not found.

C_{CHP}= 11.4 kW$_{el}$ / 15.5 kW$_{th}$= 0.74
Specific footprint: n/a
Specific cost: n/a

Source: [Dentice d'Accadia et al., 2003]

Air need for stoichiometric combustion

LHV and l_t Values from H. Alvarez, 2006. Taking air density at 1 atm, 15°C to be 1.225 kg/m^3. The l_t is defined as the volume of standard air needed for stoichiometric combustion of1 kg of fuel.

For wood, 60% water content:

$$l_t = 2\, m^3/kg \qquad LHV = 7\, MJ/kg$$

$$\dot{m}_{fuel} = \frac{q}{LHV} = \frac{100\, kJ/s}{7000\, kJ/kg} = 0.0143\, kg/s$$

$$\dot{V}_{air} = \dot{m}_{fuel} * l_t = 0.0143\, kg/s * 2\, m^3/kg = 0.0286\, m^3/s$$

$$\dot{m}_{air} = \dot{V}_{air} * \rho_{air} = 0.0286\, m^3/s * 1.225\, kg/m^3 = 0{,}035\, kg/s$$