Ny gasturbinteknik 2009–2011

Gas Turbine Developments

Rapport 2011

Elforsk rapport 12:27

Magnus Genrup, LTH

Juni 2012

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Förord

Projektet är en direkt fortsättning på Elforsk-projekt 2329 som avrapporterats i slutrapporten 08:16 med målet att säkerställa beställarkompetens avseende moderna gasturbinkombianläggningar hos de i projektet deltagande parterna. Föreliggande rapport är slutrapporteringen inom ramen för det pågående fortsatta treårsprojektet (Elforsk-projekt 2537). Tidigare har två delrapporter utkommit; år 2010 utkom den första delrapporten ”Ny gasturbinteknik, Rapport 2009”, Elforsk rapport 10:63 och år 2011 utkom den andra delrapporten ”Ny gasturbinteknik, Rapport 2010”, Elforsk rapport 11:22.

Ansvarig för projektet har varit Magnus Genrup, Lunds Universitet. Treårsprojektet har finansierats av E.ON Värmekraft Sverige AB, Göteborg Energi AB, Svenska Kraftnät och Vattenfall AB.

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Juni 2012

Lars Wrangensten

Programområdesansvarig

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Executive Summary

The last three years have certainly been a game changer with respect to combined cycle efficiency and operational flexibility. All major manufacturers are able to offer plants with efficiencies around 61 percent. Siemens has a TÜV-certified performance of 60.75 percent at the Irsching site outside Berlin. The old paradigm that high performance meant advanced steam-cooled gas turbines and slow started bottoming cycles has definitely proven false. Both Siemens and General Electric are able to do a hot restart within 30 minutes to, more or less, full load. This is, by far, faster than possible with steam cooling and the only technology that is capable of meeting the future flexibility requirements due to high volatile renewable penetration.

All major manufacturers have developed air-cooled engines for combined cycles with 61 percent efficiency. Steam cooling will most likely only be used for 1,600°C firing level since there will be an air shortage for both dry low emission combustion and turbine cooling.

The increased combined cycle efficiency is a combination of better (or higher) performing gas turbines and improved bottoming cycles. The higher gas turbine performance has been achieved whilst maintaining a 600°C high pressure admission temperature – hence the gain in combined cycle performance. The mentioned requirements of both high gas turbine performance and sufficient exhaust temperature, should impose both an increase in pressure ratio and increased firing level.

The price level (2010) was on average 30-35 percent higher than the minimum level in 2004. The cost of ownership (or per produced unit of power) is strongly governed by the difference between the electricity and the fuel price. The importance of evaluating all factors (like degradation and de-icing operation) in the economic model cannot be stressed too much since it may have a profound impact on the analysis. The test code guarantee verification test is indeed an important verification that the plant fulfills the expectations. One important thing, however, is not to accept the test uncertainty as a test tolerance since it will provide the manufacturer overwhelming and unfair odds.

Siemens, General Electric, Alstom and Mitsubishi have all developed new versions of their combined cycle platforms. The key for 61 percent efficiency is high performing gas turbines, which includes components, pressure ratio and firing temperature. In addition, the exhaust temperature has to be at a level for maximum bottoming cycle performance. Today, most manufacturers have 600°C steam turbine admission temperature capability and the optimum exhaust gas temperature should therefore be on the order of 25-30°C higher. Both Siemens and General Electric have presented advanced admission data (170 bar/600°C and 165 bar/600°C) for their bottoming cycles. It is probably safe to assume that the other manufacturers are at the same level. The striking point is that both Siemens and General Electric appear to have no start-up time/ramp-rate penalty despite the advanced steam data.

There have also been several high-performing simple-cycle units presented during the project duration.

Nowadays, it is common to have a maintenance agreement at some level for risk mitigation. There are different levels of contractual services ranging from parts agreement to full coverage "bumper-to-bumper" LTSA services. One can chose to use either the OEM or another (third party) service provider. In many cases, the financing organs or insurer requires an LTSA (or better) for risk mitigation to level the insurance cost at a reasonable level. There are ways of potentially reducing the maintenance spending and one should always avoid lumped methods with equivalent hours. The word "lumped" is used in a sense that the two different ageing mechanisms (creep, oxidation, regular wear and tear and stresses related to thermal gradients during start and stop) are evaluated as equivalent time by e.g. assuming that a start consumes time rather being a low cycle. A competent monitoring system can be a good investment - even if only a single failure can be avoided.

The total world-wide gas turbine fleet is in the order of 47,000 units and the total value of the gas turbine aftermarket was 2009 13.8 B€ (13.8×10^9 €). The after-market is, indeed, valuable to the manufacturers since all 47,000 units requires maintenance on a regular basis. Certain in-house produced parts may be offered with several hundred percent's margin – in contrast to about ten percent for a complete new turn-key power plant. The reward for the user, by having a LTSA, is discounted parts and prioritized treatment by the supplier.

The combined cycle has about half the carbon dioxide emission compared to a coal fired plant. The large difference is driven by the higher efficiency and the higher hydrogen content in natural gas. This in combination with the flexibility makes combined cycles attractive for both flexible non-spinning and spinning reserve power – with comparably low emissions of greenhouse gas. The partial pressure of carbon dioxide is low when compared to coal firing. The lower partial pressure makes the sequestration process more difficult. There is also a much larger flue gas mass flow since a typical combined cycle has around 1.5 kg/s flue gas per MW (kg/MWs or kg/MJ) in contrast to approximately 0.95 for a coal fired plant. The low partial pressure can be increased by introducing recirculation of flue gases. In addition to the discussed post-combustion process, there are other technologies being developed based on e.g. oxyfuel and IGCC/H_2. All suggested technologies come with a significant efficiency penalty.

A gas turbine can be made carbon dioxide neutral by firing renewables. Most manufacturers have quite wide fuel capability ranges but no true omnivorous gas turbine exists yet. There are several issues related to the fuel system (valves and pressure drops) and combustor (fuel nozzles, vortex break-down, etc.). There is also a turbomachinery dimension related to stability, forced response and potential flutter problems. The latter is forces acting on the blading which are functions of the displacement, velocity, or acceleration of the blades – and these forces feed energy into the system.

The high penetration of volatile production like wind and solar (both CSP and PV) have been a game changer for the combined cycles. It is safe to assume that the role for the gas turbine based plants will change from base and mid-merit load to daily cycling and peakers.

Sammanfattning

Under projektets tre år har både verkningsgraden och flexibiliteten för kombianläggningar blivit avsevärt bättre. Idag erbjuder alla större tillverkare verkningsrader uppemot 61 procent. En lyftkyld kombi kunde tidigare nå maximalt runt 58-59 procents verkningsgrad medan ångkylda låg runt 60 procent. Vid EONs kraftverk i Irsching, utanför Berlin har TÜV certifierat verkningsgraden på Siemens SGT58000H anläggningen till 60,75 procent.

Förr trodde man att ångkylning var enda sättet att nå verkningsgrader över 60 procent. Det gav gasturbiner med avancerad ångkylning och svårstartade bottencykler pga. höga ångdata. Här har det skett ett paradigmskifte. Både Siemens och General Electrics gaskombianläggningar klarar idag en varmstart på mindre än 30 minuter, upp till full last. Det är betydligt snabbare än vad ångkylda maskiner klarar. Lyftkylda anläggningar är den enda teknik som klarar att möta de krav på flexibilitet som är kopplade till framtidens flyktiga elproduktion.

De större tillverkarna har idag utvecklat luftkylda gastubiner för kombiprocesser och nått 61 procents verkningsgrad. Ångkylning är idag förmodligen bara ett alternativ för maskiner med eldningstemperaturer över 1,600°C, där luften inte räcker till både för kylning och låga emissioner.

Att kombianläggningarnas verkningsgrad har ökat är ett resultat av att både gasturbinens- och ångcykelns prestanda har blivit bättre. Detta har skett med bibehållen admissionstemperatur för HP- och IP ånga på 600°C, vilket gör att hela kombianläggningens prestanda ökar. Kraven på både hög gasturbinprestanda och tillräcklig utloppstemperatur gör att tryckförhållande och eldningstemperatur ökar.

Prisnivån för den här typen av anläggningar ligger (2010) 30-35 procent högre än den låga nivån 2004. Ekonomin för anläggningen drivs till största delen av skillnaden mellan bränsle- och elpris. Det kan inte nog betonas hur viktigt det är att ta hänsyn till alla faktorer, även exempelvis degradering och de-icing i den ekonomiska modellen eftersom det påverkar analysen. Prestandaprov ger en viktig bekräftelse på att anläggningen uppfyller förväntningarna. Det gäller att inte acceptera provets onoggrannhet som provtolerans eftersom det ger leverantören orättvisa fördelar.

Siemens, General Electric, Alstom och Mitsubishi har alla utvecklat nya versioner av sina kombianläggningar. Nyckeln till 61 procents verkningsgrad är gasturbiner med hög prestanda, vilket inkluderar komponenter, tryckförhållande och eldningstemperaturer. Även utloppstemperaturen måste vara vid en sådan nivå att bottencykeln får maximal prestanda.

Idag har de flesta tillverkarna ångturbiner som är konstruerade för 600°C. För att nå maximal verkningsgrad i en kombiprocess är 625-630°C en lämplig rökgastemperatur. Både Siemens och General Electric har presenterat avancerade ångdata i sina kombianläggningar som ligger runt 165-170 bar och 600°C. Det borde betyda att de blir trögstartade men båda leverantörernas anläggningar kan startas under 30 minuter vid varmstart, trots avancerade admissionsdata.

Numera är det vanligt att ha någon form av underhållsavtal för att minska riskerna. Det finns olika nivåer av avtalsbaserade tjänster som sträcker sig från enskilda delar till hela systemlösningar av typen LTSA. Man kan välja att använda antingen OEM eller tredje parts tjänsteleverantör. I många fall kräver finansieringsorganen eller försäkringsgivare en LTSA eller bättre för att minska riskerna och för att få försäkringskostnaderna på en rimlig nivå.

Det finns cirka 47 000 körbara land- och fartygsbaserade gasturbiner i världen och eftermarknadens värde var 2009 13.8 miljarder € (13.8×10^9 €). Eftermarknaden för dessa enheter är mycket värdefull för tillverkaren då en tillverkare kan ha i storleksordningen hundratals procent nettomarginal för egentillverkade delar – medan t.ex. nettomarginalen för en komplett anläggning ligger runt 10 procent. Vinsten för användaren är rabatterade delar och prioriterad behandling av leverantören.

En modern kombianläggning släpper ungefär ut hälften så mycket koldioxid som en motsvarande koleldad anläggning. Det beror på att kombianläggningen har högre verkningsgrad och att naturgas har högre andel väte jämfört med kol. Att kombianläggningarna också är mycket flexibla gör dem attraktiva som reservkraft för att balansera t.ex. vindkraft. Partialtrycket för koldioxid i kombianläggningens avgaser är lågt i jämförelse med rökgaserna från ett kolkraftverk. Det lägre partialtrycket gör avskiljningsprocessen svårare. Rökgasflödet från kombianläggningar är också mycket större eftersom massflödet är ungefär 1,5 kg/s rökgas per MW (kg/MWs or kg/MJ) vilket kan jämföras med ett kolkraftverk som har ett rökgasflöde på ungefär 0,95 kg/s per MW (kg/MWs or kg/MJ). För att få upp det låga partialtrycket innan avskiljning kan man recirkulera rökgaserna.

Utöver post-combustion avskiljning finns andra tekniker som exempelvis oxyfuel och IGCC/H_2 men dessa tekniker påverkar verkningsgraden negativt.

Om gasturbinen eldas med förnyelsebart bränsle kan den bli koldioxidfri. De flesta tillverkare tillåter ganska breda bränslespecifikationer men det finns ingen gasturbin som klarar av större variationer i bränslekvalitet. Många av problemen rör inre- och yttre bränslesystem och brännkammaren, exempelvis aerodynamisk flamhållning.

Den stora ökningen av mycket intermittent kraftproduktion har ändrat förutsättningarna för kombianläggningar. Det troligt att gasturbinbaserade anläggningar kommer att gå från att vara baskraft till att bli peakers istället.

Chapter Summary

The purpose of the project is to increase the competence for optimal economy, environmental performance and availability of gas turbine based plants by following the development with emphasis on investment costs, operational availability, maintenance costs and fuel-flexibility for modern combined cycles.

Technology Trends and Roadmaps

The role of the combined cycle will change from being a natural gas fired mid-merit or base load plant to either a fuel-flexible base load or a plant for covering for daily variations. The introduction of high levels of volatile wind and solar power capacity will create a market for fast start and ramping production. A future, either economical incitement or legislation for carbon abatement will also call for special types of gas turbines. On top of fuel flexibility, operational flexibility and CO_2, the market will still require high efficient and reliable engines. Steam-cooled engines will not meet market expectations on operational flexibility.

Market Overview

There exists a diversity of engines ranging from a hundred kW to 460 MW. There are three major types of gas turbines namely: frames, industrial and aero-derivatives. Frames are normally heavy rugged machinery but ranges from 5 MW to the biggest of 460 MW. Industrial types are "lighter" than Frames and are either single- or multi-shaft. Their power range is typically up to 30-40 MW. Aeroderivatives are former aero-engines that have been adapted for land-based operation. The aeroderivatives typically offers the highest simple-cycle efficiency whilst the frames have highest combined cycle performance. All engines (regardless of type) are non-geared above approximately 100 MW, hence 3,000 or 3,600 min^{-1} for 50 and 60 Hz, respectably.

Economics

The 2010 combined cycle price (first cost) is on average 20 percent higher than 2000. The calculated economic metrics are based upon assumed fuel and electricity prices. There are unfortunately no general figures for assessing electricity and fuel prizing under all conditions over an assumed plant life cycle. Figures related to OEM spending are often proprietary since they are embedded in a flat rate (fixed costs and payment structure) maintenance contract. There are several third party organizations offering service and parts. The user has to decide upon the associated risk involved since a third-party doesn't necessarily have the full competence. A safe conclusion is there is a scale of size in terms of net present value and internal rate of return.

All power classes have their own features and price levels and it is hard to discuss in general terms.

Recent Developments

Most new heavy frames are approaching 40 percent efficiency in simple cycle and offers 60-61 percent in combined cycle mode. The 60 percent barrier was broken by high-performing gas turbines without steam cooling – hence no flexibility penalty.

There are several new engines in the medium and small power bracket that are exceeding 40 percent efficiency.

Some Aspects of Gas Turbine and Plant Maintenance

The service schedule of a gas turbine (and plant) follows a predefined set of cycles. There are several ways of arranging the cyclic maintenance through either simply buying parts to have a long term contractual agreement (LTSA and CSA) with a supplier. Each step between buying parts, classic LTSA, CSA, etc., increases the risk exposure for the contractor

The cost of a condition monitoring system can prove to be a good investment even if only a single hot path failure can be avoided. Any competent system should be able to detect minute changes to the exhaust temperature (EGT) pattern. Most hot-end failures have some kind of influence on the EGT-spread and should be treated with caution.

An old rule of thumb is that the maintenance cost is twice the initial (or first) cost during the plant life.

Gas Turbines and Carbon Emission

Gas turbine based plants will probably be fairly CO_2-neutral by either firing bio-fuels or removal. The latter could be either pre- or post-combustion based firing a large variation of fuels. Post-combustion technologies are not optimum for a normal gas turbine based cycle, since the partial pressure of CO_2 in the flue gas is low and further complicated by the higher specific mass flow. There exists no true capture-ready plant since the steam turbine has to be adapted for the massive extraction. A good rule of thumb seems to be twice the first cost and footprint.

Fuel Flexibility

A true off-the-shelf fully fuel flexible gas turbine does not exist and all OEMs have a suitable Wobbe-Index range for their specific engines. Most gas turbines could probably be fired with low-calorific fuels without major re-design of the turbomachinery. The necessary modification will probably be limited to the combustor and fuel system if the heating value is kept above 20 MJ/kg (compared to approximately 50 MJ/kg for methane). There are a few critical problems that any competent OEM has to address before commissioning of a low-LHV engine.

Operational Flexibility

The plant flexibility will certainly be of paramount importance when the amount of volatile power production is increased. A typical wind power plant starts producing at 4 m/s and increases in a cubic fashion to some 12 m/s where the rated output is reached. The cut-off speed is typically at 25 m/s where the production is abruptly stopped. Some features of handling flexibility are discussed together with the associated lifing penalty. Most simple cycle units will be able to start and be fully loaded within 10 minutes. A recent initiative by Siemens has resulted in the FlexPlant™ concept where the SGT6-5000 can reach 150 MW within 10 minutes. Full load is reached in another two minutes. The plant offers excellent turn-down to 40 percent load with single digit CO. The Flexplant™ concept is the only combined-cycle plant that could be certified for non-spinning reserve.

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1 Introduction

1.1 Background

Until the 90s Sweden had an ageing fleet of back-up units with low annual fired hours. The most common type of unit in Sweden is Power Pack™, based on Pratt & Whitney's JT3/FT3/GG3 and JT4/FT4/GG4, with a rugged power turbine from Stal-Laval (now Siemens). In the 90s, three plants were built with dry low NOx technology based on the GT10A unit from ABB Stal AB (now Siemens). During recent years, four SGT-800 units have been commissioned in Helsingborg and Gothenburg. The most recent and largest gas turbine is the 300 MW General Electric Frame 9 at Öresundsverket in Malmö.

The project is a continuation of the earlier ELFORSK project 2329.

1.2 Project execution

The project runs 2009-2011 at Lund University, Department of Energy Sciences. Project manager and responsible for the technical content is Associate Professor Magnus Genrup and Majed Sammak.

1.3 Limitations

All quoted performance and economic parameters are for cold condensing mode only. An adequate analysis of e.g. introducing district heating should involve detailed cycle modeling for each of the 60 different plants.

1.4 Common abbreviations and notations

AN^2 or ANSQ	Annulus area times blade speed squared – gives a gauge of e.g. root pull.
ANN	Artificial Neural Network
ASU	Air Separation Unit
BI	Boroscope Inspection
BLISK	Bladed Disc
BTMS	Blade Temperature Measurement System
CCS	Carbon Capture and Storage/Sequestration
CBM	Condition Based Maintenance
CI	Combustion Inspection
COT	Combustor Outlet Temperature[1]
CSA	Contractual Service Agreement
DCF	Discounted Cash
DLE	Dry Low Emission
DS	Directional Solified
EGT	Exhaust Gas Temperature
EIS	Engine/Entry in Service
FGR	Flue Gas Recirculation
FN	Turbine Flow Number or capacity
FOB	Free/Freight On-Board
HARP	Heater Above Reheat Point
HCF	High Cycle Fatigue
HGP	Hot Gas-Path Inspection
HPC	High Pressure Compressor
HPT	High Pressure Turbine
HTC	Heat Transfer Coefficient
HRSG	Heat Recovery Steam Generator
ICR	Inter-Cooled and Recuperated
IGV	Inlet Guide Vane
IPC	Intermediate Pressure Compressor
IPT	Intermediate Pressure Turbine
IRR	Internal Rate of Return

[1] Typically used synonymously with “firing”

MEA	Monoethanolamine (C_2H_7NO)
MI	Major Inspection
NDE	Non-Driving End
NPV	Net Present Value
O&M	Operation and Maintenance
OEM	Original Equipment/Engine Manufacturer
OPR	Over-all Pressure Ratio
OTDF	Overall Temperature Distribution Factor
LCC	Life Cycle Cost
LHV	Lower Heating Value
LMTD	Logarithmic Mean Temperature Difference
LPC	Low Pressure Compressor
LPT	Low Pressure Turbine (normally same as power turbine)
LTSA	Long Term Service Agreement
PT	Power Turbine
QFD	Quality Function Deployment
RAMD-S	Reliability, Availability, Maintainability, Durability and Safety
RH	Relative Humidity
RTDF	Radial Temperature Distribution Factor
SCOC	Semi-Closed Oxy-fuel Cycle
SCR	Selective Catalytic Reduction
SF	Scale Factor
SOT	Stator Outlet Temperature
TBC	Thermal Barrier Coating
TMF	Thermo Mechanical Fatigue
VSV	Variable stator Vane
WI	Wobbe-Index (see equation in section 9)
WLE	Wet Low Emission (cf. DLE)

2 Disclaimer

- The material is presented in bona fide and the material is solely based on open source material like trade press, ASME IGTI and PowerGen.
- The analysis represents the views of the author and not the individual manufacturers.
- The analysis is held on a basic level rather than in-depth for clarity reasons and maintaining a user/buyer focus. There is no claim to fully address all aspects of a certain issue.
- All figures used for economic analysis are estimates.

3 General trends

The role of the combined cycle will probably change from being a natural gas fired mid-merit or base load plant to either a fuel-flexible base load or a plant for covering the daily variations (i.e. operational flexibility). The introduction of high levels of volatile wind and solar power capacity will create a market for fast start and ramping production. The average capacity factor for wind production is certainly, on average, less than 40-50 percent. Wind power levels on the order of 20 percent installed capacity are present in some countries, hence a need for flexible production. A recent report by Pöyry[2], shows wind power prognosis for 2030 with 43 GW and 8 GW in the UK and Ireland, respectively. This level will call for some 41 GW flexible production capacities. A future, either economical incitement or legislation for carbon abatement will also call for special types of gas turbines. On top of fuel flexibility, operational flexibility and CO_2, the market will still require high efficient and reliable engines. The latter two requirements have historically not been conformal.

The advent of modern lateral drilling technologies will introduce shale gas as a complement to natural gas.

Customer focus/market pull	OEM focus	How
Low first cost	High specific power	Increased firing
Low fuel burn and LCC	High efficiency and dependability	Increased pressure ratio and firing + proven design (!)
Little maintenance	High maintainability	Design, CBM[3] and monitoring
Small environmental footprint	Low emissions (and high efficiency)	DLE for NOx and high effic'y and advanced cycles for CO_2
No surprises	Proven designs and structured development processes	Mature products. It takes time to discover all possible failure modes.
Fuel flexibility		Advanced combustors, flexible fuel systems and surge margin
Operational flexibility	Highly reliable designs	Proven designs

[2] Pöyry, Impact of Intermittency: How Wind Variability Could Change the Shape of the Brittish and Irish Electricity markets, Summary report, July 2009.

[3] Condition-based maintenance

The above requirements with e.g. fuel flexibility, high efficiency and high reliability introduce several issues in terms of available lifing. Bio-fuels may be corrosive and force the OEMs to develop exotic high temperature materials with both good oxidation- and corrosion resistance. Cyclic operation together with can-annular systems is another issue. Canned designs have a relative higher overall temperature distribution factor (OTDF) that may result in thermo-mechanical fatigue (TMF) problems. This type of problems typically manifests itself as cracks near the fillets in the first vane segments.

The prize trend has been a per annum drop 2000-2004 and an increase until 2009. The level has dropped since last year but it is too early to say whether this is a trend or not. The trend 2008-2009 shows a plateau that is probably driven by the recent regression in the world economy.

No OEM's besides General Electric, Rolls-Royce and Pratt & Whitney have yet developed engines for true flexible mid-size. The situation may change in the future depending on the development of flexible combined cycles. The lag in efficiency is on the order of 10 percentage points, hence advantageous to invest in combined cycles when fuel prices are high.

3.1 Technology trends and road-maps

The general technology trends will probably be:

- The heavy-frame firing[4] level will increase to 1,600 °C for combined cycle performance. The old limitations in cooling and material technology (i.e. lifing) will probably be replaced by the amount of air available for dry low emissions (DLE). Mid-size gas turbines will most likely not follow this trend and stay below 1400 °C. Higher frame firing levels will force the steam turbines to 600(+) °C admission temperatures, calling for usage of higher chromium alloys in the hot sections. Engines fired at levels of 1,600…1,700 °C will probably have little market penetration due to the necessity of steam cooling.
- Operational flexibility requirements with little or no RAMD-S[5] impact. Most OEMs are capable of 30 min hot-start and steep (35-50 MW/minute) ramp-rates.
- High-temperature engines will rely upon thermal barriers (TBCs). This feature is probably not accepted within the oil- and gas community and is one of the reasons for having lower firing level in the mid-size bracket. Both Siemens H-class and Mitsubishi F-class have reverted back to directionally solidified (DS) blades in contrast to single crystal blades (SX). This is probably driven by cost and the fact that DS-blades will do the job with proper cooling and TBCs.
- Higher engine efficiency requirements will force the OEMs towards higher engine pressure ratios. Both industrials and eventually frames will approach 25 with a difference in firing of 200 °C. Higher cycle

[4] The word firing is used synonymously for combustor outlet temperature (COT) throughout the text.

[5] Reliability, Availability, Maintainability, Durability and Safety

pressure ratios will also increase stage count and potentially longer rotors. Frames with three staged turbines will most likely be replaced by four stage designs. There is an efficiency potential associated with the fourth stage since the stage loading will be reduced and the possibility of having a larger exhaust.

- Steam cooled engines will not meet market requirement for rapid start- and ramping capability. The total 50- and 60 Hz sales (since introduction) of the Mitsubishi and General Electric steam cooled units are 66 and 5, respectively. This trend has been further established by the latest F-class units by General Electric and Mitsubishi, H-class by Siemens and GT26 by Alstom. The old paradigm that steam cooling was a requirement for 60 percent efficiency has definitely proven false.
- Specific flows will continue to increase and approach aero-engine technology. The latest Siemens 50 Hz engine has a flow of 820 kg/s@3000 rpm whilst the 9FB.05 by General Electric has a flow of 745 kg/s. The 50 Hz version of the new MHI J-class will most likely be the highest at approximately 860 kg/s. The absolute maximum of today is around 1000 kg/s@3000 rpm but this has only been achieved with multi-spools. A clear trend has been set by the latest GE and Alstom upgrades where aero-engine technology has replaced older designs.
- The trend with higher firing levels has to be accompanied by effective repair technologies (e.g. welding repair for exotic turbine materials and turbines) for reasonable cost of ownership.
- Better prediction capacity within the OEMs should mitigate issues related to dynamics/instabilities like forced response, flutter and combustor rumble (pressure pulsations). All major OEMs have full engine test capability ranging from semi-commercial operation to dedicated full load beds. It is also possible to introduce on-line compressor blade (on an individual blade level) vibration measurements and associated protection system – in situ.

Today, all major OEMs are capable of offering 61 percent and Siemens has a TÜV-certified efficiency of 60.75 percent – measured/demonstrated in-situ at the Irsching 4 site. The key factors here are principally the gas turbine and its components. An old saying is that one can only increase the efficiency of a combined cycle by increasing the gas turbine efficiency – without seriously affecting the bottoming cycle. In other words, one still needs a hot exhaust for good combined cycle and simultaneously high gas turbine efficiency. The main driver for high gas turbine efficiency is pressure ratio and hence the success factor is to achieve both. There are also conflicting requirements between bottoming cycle efficiency and flexibility. One can show that the steam turbine start-up time may increase by a factor of three by introducing advanced admission data. It is the combination between higher pressures and e.g. an increase in admission temperature from say 565°C to 620°C. This magnitude could very well increase the IP-cylinder thickness by a factor of three for maintaining a certain creep life requirement. Hence, for the same thermal stress, the time span increases on an equal basis (i.e. again a factor of three). An in-depth explanation is quite involved and outside the scope of the current report. There has also been a debate over the years whether the once-through HRSG technology should be better off than drum boilers in terms of cycling. The general perception is that there are other areas within the HRSGs that are more exposed / influenced – like the HP superheater and re-heater headers and HARP-attachments[6]. Hence, the once-through technology should generally not be superior in terms of flexibility.

A true leap frogging step was annunciated in 2010 with MHI's (Mitsubishi) revolutionary design concept where the entire first row of blades has been omitted. Their design was presented at the annual ASME IGTI conference in 2010. The design offers considerably lower part count and cooling consumption. No information has been published related to the production engine platform or market introduction.

Mitsubishi has carried out full scale testing of their new 60Hz[7] J-class with 1,600°C firing level in Takasago (T-Point test station) during 2011. The M501J engine will deliver 320 MW in simple cycle or 460 MW at >60 percent in combined cycle. The 50 Hz version will have an output of 460 MW and 670 MW in simple- and combined cycle, respectively. The efficiency is 61.0 and 61.2 for the 60 and 50 Hz versions – reflecting the scale of size on efficiency for the larger 50 Hz unit. This is the world's largest gas turbine at the time of writing. By virtue of the firing level, the new J-class uses steam-cooled combustor liners from the G-class. The reason for introducing steam is typically need for both effective liner- and blading cooling and still having sufficient air for dry low NOx (DLE) technologies. The emissions are guaranteed at / rated below 25 ppm(v) NOx and 9 ppm(v) CO. The ramp rate is 20 MW per minute between 100 percent and 50 percent load. The gas turbine start-up time is 25-30 minutes and the steam turbine will add another ten. This level will certainly require an auxiliary boiler for the steam cooling system.

Mitsubishi has also followed the flexibility trend by introducing a high-performing all air-cooled version of their F-class. The new engine is called F5 and is rated at 350 MW. The combined cycle power is above 500 MW with an efficiency of 61 percent. The design is based on combining features from F4, GAC (air-cooled G-class) and the latest J-class. The firing level is the same as for GAC and the guaranteed NOx level is 15 ppm(v).

Alstom has launched the latest GT26 unit with a guaranteed performance of 61 percent. This step has been possible by introducing aero-engine technology from Rolls-Royce into the GT26 platform. The compressor is redesigned with a higher mass flow. The engine has been tested in the Alstom test facility in Birr (Switzerland) since March 2011. The new low-pressure turbine has been in commercial operation (in-situ) for a full year before introduction. The unit can either be operated in performance optimized mode or lifing optimized mode. The latter is simply a reduction in the second burner firing level that prolongs the inspection interval. The Alstom flexibility concept is to park the plant at a minimum load with emission compliance. The

[6] Where the tubes are attached to the headers/manifolds.

[7] Japan has 60 Hz from Kyoto and westward whilst Tokyo and eastward has 50 Hz. This is due to historical reasons - Tokyo purchased German AEG equipment and Osaka American General Electric equipment at the end of the 19th century. Connection between the grids is by HVDC.

unit can be operated at low load with only the first set of burners in operation and fully closed IGV/VSVs.

General Electric has launched their FlexEfficiency™ 50 Combined Cycle Power Plant. The performance follows the trend of the other OEMs and the 50 Hz version is rated at 520 MW and 61 percent efficiency. The GE-plant has an impressive ramp-rate of more than 50 MW/min. The efficiency is kept above 60 percent down to 87 percent load and is emission compliant down to 40 percent. The gas turbine itself can be brought up to full load within 15 minutes and a hot-restart of the plant takes less than 30 minutes. The pressure ratio is slightly higher for the new (.05) version and the compressor is re-designed with 18 stages. The previous 9FB.03 compressor was actually a linear scaled and zero-staged E-class compressor. The word "zero-staging" is used when an additional front stage is attached to an existing design.

Pratt & Whitney has released their new FT4000 120 MW platform rated at above 41 percent efficiency. The product follows Pratt and Whitney's practice with two three-shaft gas turbines to a common generator. The turboset is a derivative from the flying PW4000 engine. The twin engine configuration offers, by virtue of having two engines, higher part load efficiency since one can be kept at base load. No information is available at the time of writing whether the unit can be operated in synchronous condensation mode (phase compensation) with or without a SSS-clutch.

In the 40 MW bracket, two new engines were launched in 2010 by Rolls-Royce (RB211-H63) and Siemens (SGT-750). Both engines have dry ratings around 37 MW and electrical efficiencies around 40 percent. Both engines are ideally suited for mechanical drive with power turbine speeds around 6000 min^{-1}. These new engines follow the recent efficiency trend set by the General Electric LM2500-G4 and the smaller Solar Turbines Titan 250. This efficiency level has previously only been offered with aero-type compound engines and is definitely a significant step in terms of reducing fuel burn. The main driver for high efficiency is mainly engine pressure ratio and high component efficiency. The Siemens SGT-750 is claimed to only require 17 days of maintenance in 17 years.

Siemens launched a new rating of the SGT-400 (formerly Cyclone) in 2010. The new rating is at 15 MW with an efficiency of 37.2 percent. This level of efficiency is remarkable in this power segment. Smaller engines always, by virtue of their size, lags in efficiency when compared with larger sizes. The uprate is a traditional "high flowing" where the front of the compressor is redesigned for higher capacity. The firing level is unchanged, hence a relative low risk for improved performance. The engine has been tested and the first commercial unit was available in 2011.

A safe conclusion is that steam cooling technology wasn't a prerequisite for breaking the 60 percent barrier.

4 Overview of selected gas turbines

This overview is far from complete in terms of available engines. The overview will be in terms of micro turbines, small units, mid-size and large units. The impact on a gas turbine from having a low-LHV fuel will be discussed in Chapter 9.

A more technical view of gas turbine aspects are presented in Appendix I.

4.1 Engine configurations

A single-shaft unit is not an optimum solution for emergency power since the produced power drops steeply with load speed (e.g. grid code emergency operation). Low speed operation may also render a single shaft unit into surge due to high front compressor loading at high firing levels. Another drawback is high starting power requirement. Both issues are effectively avoided with a multi-shaft since the gas generator operates independently from the load turbine (to a first order) and the starting power is significantly smaller. Single shaft units cannot be operated in continuous synchronous condensation mode without a SSS-clutch (or similar clutch that makes independent operation of the alternator possible). The same probably holds for compound engines like General Electric LM6000 and Rolls-Royce Trent. It is not possible to generalize in terms of normal twin-shafts since the limiting factors are rotor dynamics and temperature rise due to power turbine windage.

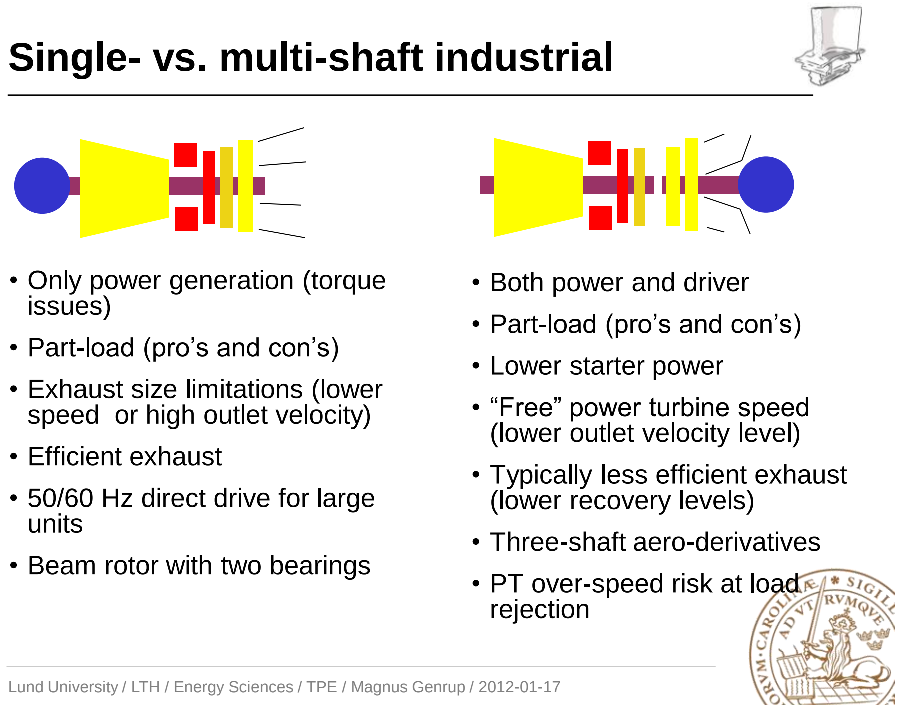


Figure 1. Single vs. multi-shaft industrial unit

4.2 Micro turbines 20-200 kW

There are a handful of small units available from manufacturers like Turbec and Capstone. There engines are typically recuperated and have efficiencies around 30 percent. Driven by size, radial components are generally used since the volumetric flows are small.

	Turbec T-100	**Capstone C65**	**Capstone C200**	**Comments**
Power	100 kW	65	200	
Efficiency	33 %			
Exhaust heat				
First cost				
O&M costs				
Fuel flex	yes	yes	yes	
Fuel spec.	Unlimited with external firing capability	Wide range	Wide range	
Turn-down	Very wide operating range due to the variable speed.			
Emissions				
Cooling	N/A	N/A	N/A	
Shaft config	Single	Single	Single	High-speed generator

The previous list is incomplete because the limited amount of available data.

The Capstone product range accepts a wide variety of fuels like low-LHV (landfill, wastewater treatment centers, anaerobic, etc.) and flare gas.

The Turbec T100 also accepts a wide variety of fuels and can also be externally fired. An externally fired unit has a totally separate and atmospheric firing system; hence any fuel could potentially be used.

By virtue of its size, a typical micro-size unit is un-cooled with an approximate firing level of 900…1000 °C. The cycle pressure ratio is typically on the order of 4…5. The rather low level is a consequence of having a recuperated process and the compressor size.

Micro turbines compete with stationary piston engines over the entire application range.

4.3 Small units 1-15 MW

There are several OEM's in this range covering most applications within the power range. The gas turbine competes with medium speed diesel engines up to approximately 10 MW. The market is dominated by Solar Turbines which has more than 13,300 gas turbine delivered. Their product portfolio covers 1…23 MW mechanical drive and gen-sets. Other OEMs in this range are Siemens, General Electric, Pratt & Whitney, Kawasaki and Rolls-Royce (among others). There is probably a limited combined cycle market and the dominating products are either simple cycle or cogeneration. The latter is typically supplying a downstream plant with process steam or hot water.

	Solar Taurus 70	Siemens SGT-400[8]	Solar Titan 130	Comments
Power, kW	7,520	15,038	15,000	
Efficiency, %	33.8	37.2	35.2	
Pressure ratio, -	16.1:1	18.8:1	17.0:1	
Exhaust temp., °C	485	539	496	
Exhaust flow, kg/s	26.9	43.7	49.8	
First cost €/kW 2010	378	331	320	
O&M costs	----------- See chapter 5 ----------			
Fuel flex (WI)	Yes	>25 MJ/m^3	Yes	
Turn-down	?	?	?	
Emissions NOx/CO	?	<15ppm	?	
Cooling	Air	Air	Air	
Shaft config.	1, 1+1	1+1	1, 1+1	

The selected engines shows have similar pressure ratio levels but different levels of firing.

The detailed architecture of each engine is not possible to discuss in detail. The reader is referred to Appendix I for a more in-depth analysis related to different engine types. It is quite common for the manufacturers to have both singe- and multi shaft versions of the same engine.

Fuel flexibility is indeed important in this range since a fuel plant[9] of relevant size should be feasible for supplying the unit. One OEM (Solar) has shown that their combustor should be able to fire most low-LHV fuels. Fuel flexibility is, loosely stated, solving burner issues, fuel system issues, engine matching, engine handling and torque issues.

[8] New rating presented at Power-Gen 2010.
[9] E.g. gasifier.

There are recent examples of using organic Rankine cycle (ORC) for heat recovery. The ORC-process can produce electricity from low-grade heat sources because the working media is “tailor-made” for each application. For gas turbine applications in this size range, pentane (cyclopentane) seems suitable due to its properties. It is also common practice to have an intermediate thermal oil circuit.

4.4 Mid-size units

The range 20…60 MW is normally referred to as the "mid-size" market. The market is dominated by General Electric's (aero-derivatives and frames), Siemens and Rolls-Royce. The market in this segment can be divided into power generation (simple, cogen and combined cycle) and mechanical drive. Gas turbines are found both upstream (e.g. off-shore compression) and downstream (e.g. pipe compression[10]) in the oil and gas industry. The driver market is typically from the lower power bracket up to some 40 MW.

	GE LM2500 G4	Siemens SGT-800	Rolls-Royce Trent 60	Comments
Power SC, kW	33,165	47,000	51,685	
Power CC, kW	44,331…48,935	66,500	64,232	
Efficiency SC, %	38.9	37.5	42.1	
Efficiency CC, %	48.5…53.6	53.7	52.7	
Pressure ratio, -	23	20	33	
Exhaust temp., °C	525	544	444	
Exhaust flow, kg/s	91.2	131.5	151.5	
First cost SC, €/kW	268	243	242	
First cost CC, €/kW	?	539	596 (2009)	
O&M costs, €/MWh	----------- See chapter 5 ----------			
Fuel flex	20…25 % N_2 or CO_2. WI>40	50 % N_2 and high C3	?	
Emissions NOx/CO	25/25	15/15	25/25	
Cooling	Air	Air	Air	
Shaft config.	1+1	1	3	

The Siemens SGT-800 and Rolls-Royce Trent 60 are available as combined cycles at 66.5 and 64.6 MW, respectively. The SGT-800 has a combined cycle efficiency of 53.7 percent, whilst the Trent has 52.7 percent. The almost five

[10] A typical pipeline has a compression station each 15…20 km.

point's higher Trent simple cycle efficiency has turned into one point lower efficiency in combined cycle. This example shows the impact from pressure ratio (and firing level) – and the difficulties in addressing markets. The specific price for the Trent-based plant is 790 USD/kW, whilst the SGT-800 plant is slightly higher at 800 USD/kW. The level of complexity is indeed much lower for the SGT-800 with its single shaft compared to the three shafts of the Trent.

4.5 Large units

The market above 100 MW is covered by direct drive “Frames” because there are no available gears over this level. There are no aero-derivatives covering this area since all parent aero-engines are at a lower rating. The market is dominated by General Electric, Siemens, Alstom and Mitsubishi.

	GE PG9371FB.03	MHI M701G2	Alstom GT26	Comments
Power SC, MW	284.2	334.0	289.1	
Power CC, MW	437.2	498.0	424.0	
Efficiency SC, %	37.8	39.5	39.6	
Efficiency CC, %	58.5	59.3	58.3	
Pressure ratio, -	18.3	21	33.4	
Exhaust temp., °C	628.9	587.2	615	
Exhaust flow, kg/s	655	737	640	
First cost SC, €/kW	174	169	175	
First cost CC, €/kW	373	364	375	
O&M costs, €/MWh	----------- See chapter 5 ----------			
Fuel flex - WI	?	?	±10%	
Emissions NOx/CO ppmv	25/?	?	25/?	
Cooling	Air	Steam	Air	
Shaft config.	1	1	1	

4.6 Sales trends 2005-2014

The total gas turbine sales during 2005-2014 are estimated to 7,750 units. The number may at, a first glance, seem very high but still only represents some 16 percent of the world's total gas turbine fleet of 46,500 engines in 2009. The distribution between the OEMs and share of each power bracket are shown in the figure below. The presented numbers are based on Forecast International and were compiled before the recent recession. Figure 2 shows the total number of units for each OEM.

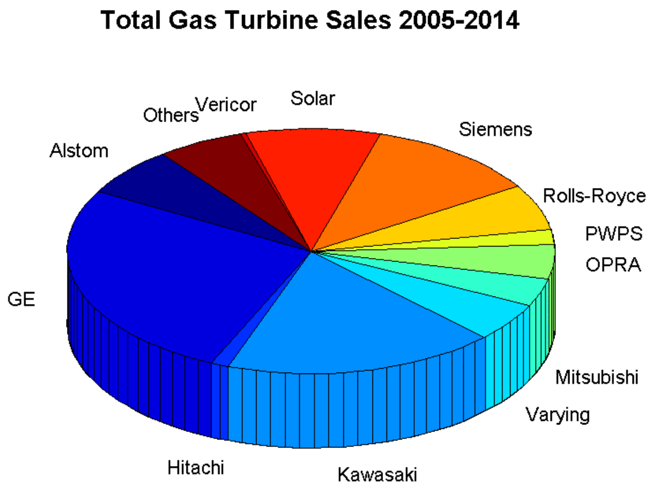


Figure 2. Land-based gas turbine sales 2005-2014

The total number of units is 7,750 covering all size ranges and does not reflect on the individual companies turn-over or installed power. The next figure (3) shows filtered (>180MW) for a better comparison of large units.

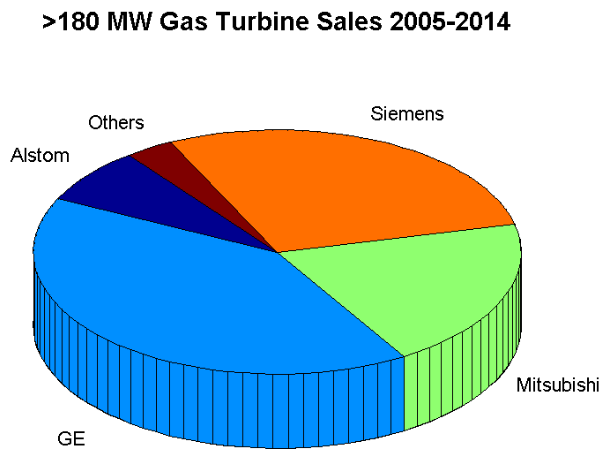


Figure 3. Large-size sales 2005-2014

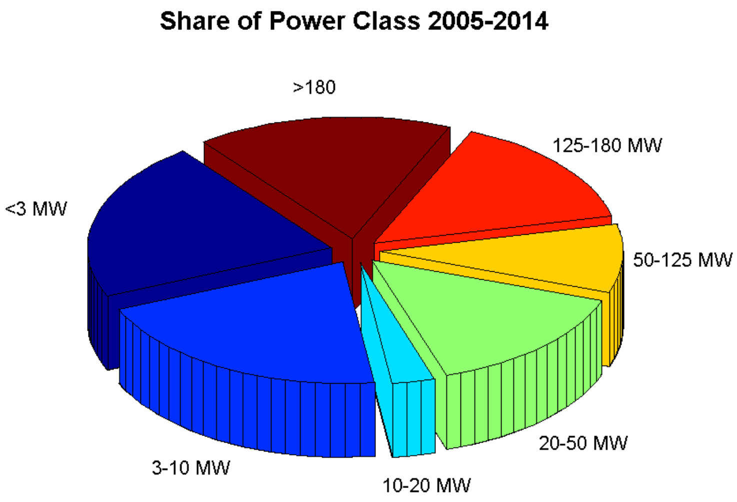


Figure 4. Size distribution 2005-2014

Figure 4 shows how the total sale is shared among the power brackets.

5 Aspects of plant life-cycle economic analysis

The value of efficiency over time is indeed high since fuel spend may be about 70 percent of the total cost. The value of two percent efficiency may be on the order of five percent fuel burn.

5.1 Price trends 2000-2010

The evolution of gas turbine and combined cycle prices has been according to the graph below. The trend has been a per annum drop 2000-2004 and then an increase until 2009. The level has dropped since last year but one cannot say whether this is a trend or not. The trend 2008-2009 shows a plateau that is probably driven by the recent recession in the world economy.

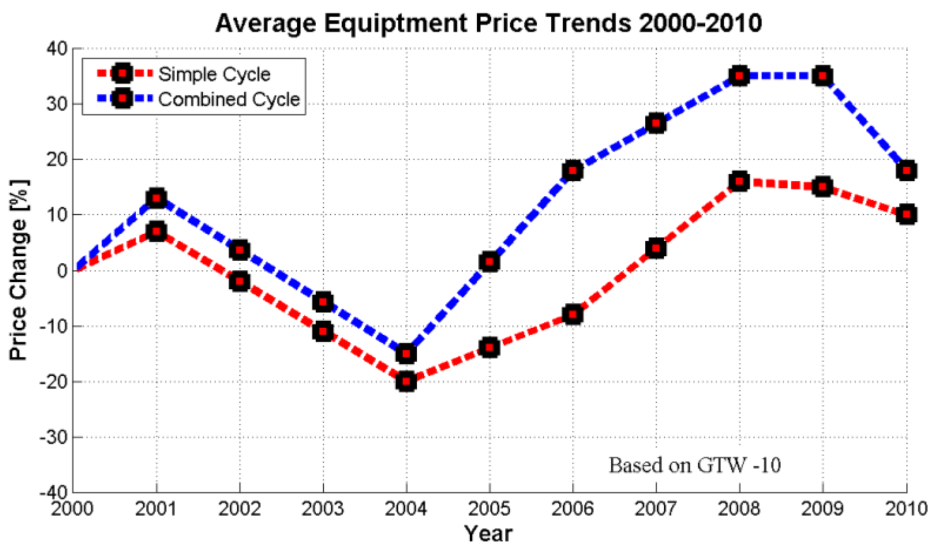


Figure 5. Gas Turbine World equipment prices 2000-2010

5.2 2010 Price level

The presented figures are valid for 50 Hz combined plants and are based on Gas Turbine World 2010 GTW Handbook. The presented turnkey budget numbers in GTW are equipment only and FOB[11] factory in 2010. The standard scope of supply includes gas turbine(s), recovery boiler with adequate number of pressure levels, steam turbine, generator(s) and associated balance of plant equipment. The gas turbine is skid-mounted in an acoustically treated enclosure for outdoor installation with standard control and starting system. An outdoor installation is probably not relevant for our climate, since the

[11] Free On Board or Freight On Board as per INCO-terms.

preferred choice is within a heated building. It is very hard to assess the impact on cost since the gas turbine anyway requires an enclosure for cooling and fire restraining features. The steam turbine is a standard sub-critical with relevant number of pressure admissions. The heat recovery steam generator (HRSG) is a standard unfired boiler including ducts but no dampers. Selective catalytic section (SCR) is not included in the scope of supply. The generators are either air- or hydrogen cooled (depending on size / power) and step-up transformer equipment is included. Cost of compressor wash system and unit is excluded. The exact details are found in Gas Turbine World 2010 GTW Handbook.

The calculated internal rate of return (IRR) and net present value (NPV) are strongly dependent on the difference between the prices of electricity and fuel. IRR is used in the report for avoidance of very large numbers. It is the interest rate that would give a NPV of zero and gives a good gauge whether the investment is sound. The presented figure is only valid for certain assumptions and should be treated with nuanced caution. One can see a rather large spread in cycle efficiency for similar sizes. The impact from fuel burn is indeed very large and fuel cost may very well be on the order of 70 percent of the total life cycle cost. Hence, an indeed strong incitement for low fuel burn when fuel prices are high.

The calculated figures are turn-key and exclude inflation and are only valid for condensing plants, i.e. no other revenue than electricity sales.

The calculated figures are based on the following assumptions (N.B. 2010 €):

- Electricity price: 66 €/MWh
- Fuel cost: 26.20 €/MWh >150 MW
  33.53 €/MWh <150 MW
- O&M Cost: f(power) – See equation
- Plant economical life cycle 25 years
- Discounted cash flow rate (DCFR): 6 %
- Utilization factor: 1.0
- Fired hours per annum: 5000 h/year

The numbers above are used to evaluate internal rate of return (IRR) in the graph below.

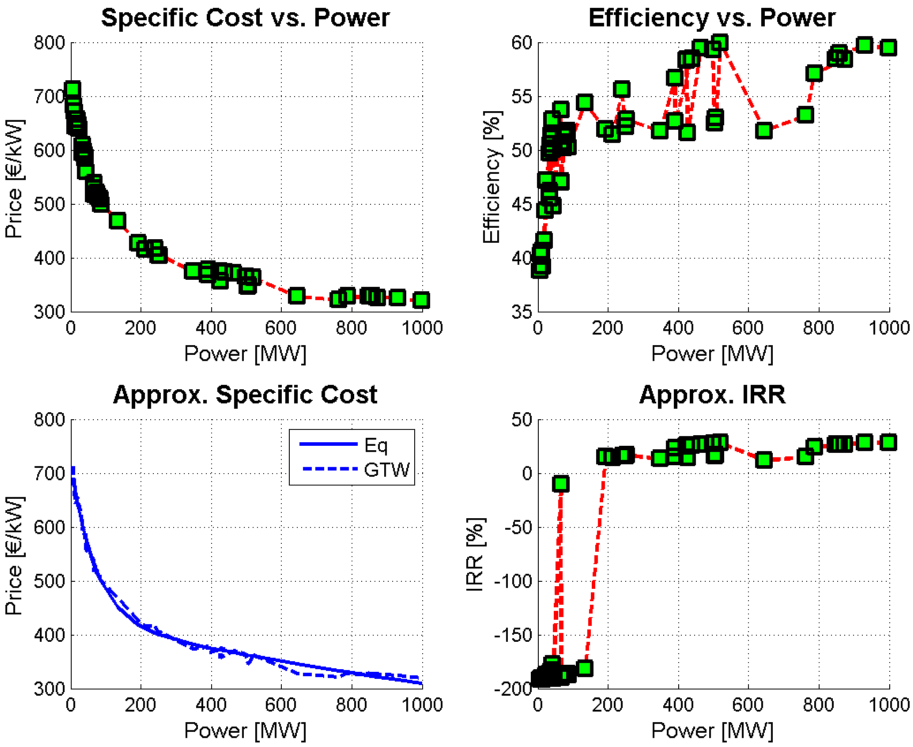


Figure 6. Power plant economics.

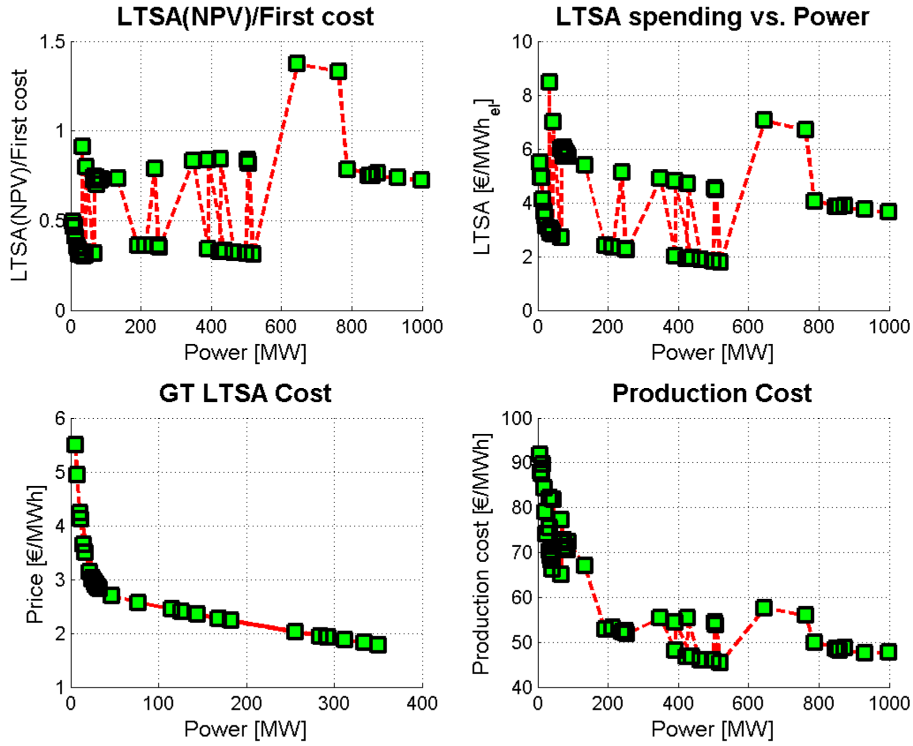


Figure 7. Power production costs

The analysis shows that most of the smaller plants has poor or even negative internal rate of return. The smaller units are heavily penalized by the very high natural gas prize below 150 MW. The gas price figures are harmonized with ”11:26 El från nya och framtida anläggningar 2011”

The plant specific cost (in 2010 €/kW@0.7551€/$) can for analytical purposes be approximated with:

$$\text{Spec.cost} = 295.8 \cdot e^{-0.01404 P_{el}} + 424.2 \cdot e^{-0.0003165 P_{el}}$$

The preceding equation is plotted in figure 6 (bottom left), showing a R^2-value close to unity (i.e. good fit). For hand calculation the following expressions provides relevant accuracy:

$$\text{Spec.cost} \approx 1072 \cdot P_{el}^{-0.1730} \quad \because \frac{\Delta \text{Spec.cost}}{\text{Spec.cost}} \approx -0.1730 \frac{\Delta P_{el}}{P_{el}}$$

The other cost data are not amendable for regression analysis since the spread is significant due to a mixture of engine generations[12].

All quoted data is valid for ISO-condition (15°C, 1.013 and 60 percent RH) and specific information has to be sought from the manufacturers for each case. The fuel composition, mainly the ratio between carbon and hydrogen, may have a significant impact on power level (and to some extent efficiency) due to the change in mass flow through the turbine and the expansion quality.

The direct O&M cost of a turn-key plant is often embedded either in a flat-rate maintenance contract or a fix cost per produced unit. The direct O&M cost does not scale with plant size (a fix rate per MWh does not give a fair comparison between sizes). Twice the size doesn’t mean twice the absolute O&M spending – instead probably limited to 10…20 percent increase. *A good rule of thumb is that the maintenance cost is twice the initial cost during the plant life.* The running profile has a profound impact on the O&M cost. As an operator, one should try to avoid solutions based on “equivalent” hours where creep/oxidation and LCF (low cycle fatigue) related issues sums up to time. Instead, the preferred method is a separate count and whichever (i.e. either time or number of events) first reaches a certain value set of the appropriate maintenance action. To further illustrate this, one could consider two cases where the engine is operated 4,000 hours with 300 starts and a second case with 8,000 fired hours and 160 starts. The latter could be questioned since it leaves little room for routine maintenance and regular service actions. All numbers are valid on a per annum base. These numbers represents daily cycling and mid-merit production. The first maintenance is scheduled for either 24,000 hours or 1,200 starts (whichever occurs first). The resulting maintenance intervals for the cases are four and three years, respectively. The lumped (or equivalent) hour’s method would have set of maintenance after 2.4 and 2.1 years, respectively. The first case would have some 700

[12] The finding is quite interesting, that an older and less performing units is offered at the same specific cost as for a high performer.

starts and 10,000 hours whilst the second would have some 300 starts and 18,000 hours. Both cases render in premature replacement of engine parts and higher O&M spending.

An attempt to develop an approximate function for analytical work on maintenance spending resulted in (€/MWh):

$$\text{GT LTSA} \approx 5.313 \cdot e^{-0.1219 P_{el}} + 2.852 \cdot e^{-0.00134 P_{el}}$$

It should be noted that the preceding equation is an approximation and real data has to be used for evaluation of budget bids.

A caveat is in place for de-icing operation. De-icing is required when the ambient condition is between -5…5 °C and above 80 percent relative humidity. The range is set to reflect bellmouth depression and the absolute content of water in the ambient air. The bellmouth is probably the most critical point in terms of costly compressor failures whereas an iced filter renders a trip (or engine surge and subsequent trip). The exact range is, in principal, governed by the compressor inlet Mach number. The impact on performance can be significant since the air has to be heated approximately 5°C (again a function of inlet Mach number), with an associated drop in engine mass flow. On top of the increased temperature, the used air is sometimes extracted from the compressor with an extra associated performance penalty. These effects have to be included for a realistic evaluation of bids.

5.3 Guarantees and verification

All projects have contractual guaranteed performance figures that, from a customer perspective, should be verified. The process of testing typically follows either an ASME standard or an ISO standard. The probably most important thing as a buyer is not to accept the test uncertainty turning into tolerance for modifying the test result (i.e. a kind of “deadband” for comparison with given guarantees). There is no logical basis for this and one can easily show that a standard 95 percent (or $1.96 \cdot \sigma$ ~ “two sigma”) confidence test uncertainty with ± 1 percent test uncertainty has a 95 percent chance of the true value to fall within the 1 percent of the measured value. If the measured value shows that the performance is a percent short (with a 1 percent uncertainty), then the result only has a 2.5 percent chance of meeting the guarantee. The likelihood of being worse is 97.5 percent (!). *The buyer should be aware of the overwhelmingly odds that are awarded the seller, if uncertainty is accepted as tolerance.*

Most OEMs want to run the acceptance test with an unnatural mode of operation, with e.g. root valves closed. The reason is that it is not un-common to have leaky steam traps and automatic by-pass valves for start-up drain and heat up. As a customer, one can choose to accept this but one should also realize the performance penalty for having steam bypassing the turbine and there is a potential loss of highly processed hot water / steam. Any competent OEM should, for a turn-key plant, be able to erect and commission the plant to a status where leaks do not have an impact on performance. Some OEMs even apply a highly questionable and unreasonable correction factor (or even a curve) for having steam leaks for turn-key deliveries. There is no logical reason for awarding the supplier this possibility of delivering a leaky plant. The normal operation, however, will eventually cause leaks and it should be a natural part of the daily operation to perform checks.

5.4 Performance degradation

All quoted performance numbers are valid for a new and clean engine. All engines will experience a drop in performance over time, but most of it is recoverable. One can distinguish between easily recoverable, part change and restoration and non-recoverable. The first is typically compressor fouling caused by airborne particles. The particles stick to the compressor blading mainly because of bellmouth condensation (or more correct depression driven condensation and sufficient), or by far worse, by bearing #1 oil leak. There are excellent descriptions in the open literature and the reader is referred to e.g. Stalders of Turbotect publications. A few words are in place for completeness: too little humidity gives too little water for gluing the particles whilst very high levels might result in some kind of spontaneous on-line wash. Hence, there exists a temperature and humidity region with higher compressor fouling rate. Fouling typically reduces the compressor capacity two- or three times the drop in efficiency. A twin-shaft unit will compensate for this by maintaining more or less the same flow at a higher speed level[13], resulting in a performance drop mainly driven by the drop in efficiency. The reason for this does not lend itself for a brief analysis since an in-depth reasoning in terms of engine matching is required. A single shaft unit does not have the capability to increase speed; hence the drop will be two-fold. The cure to fouling is in order of suitability: High efficiency particulate air filter (HEPA) technology, soak washing and on-line washing. The latter does not replace soak washing and only prolongs the intervals between soak washes. The on-line wash does not cost a standstill of the engine for cooling down and wash, but introduces a risk of getting dirt particles into the secondary air system. Degradation that requires changes of parts or repair are typically rubbed compressor blades, oxidized turbine blade tips, etc. Non-recoverable are typically distorted casings etc.

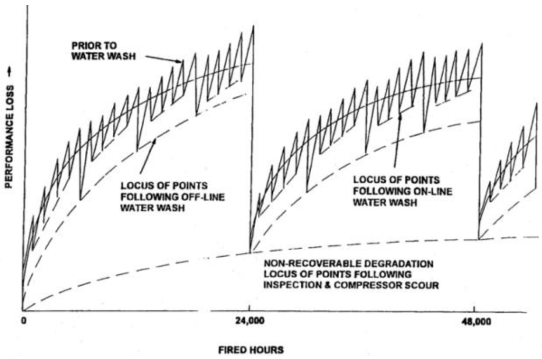


Figure 8. Typical engine degradation pattern (large leaps are maintenance events – see next section for details)

[13] An intuitive fouling detection method was developed in the 70s, using this relation. Most advanced fouling detection systems of today are either data driven or based on analytical models but don't add any insight and physics beyond this level of tool.

Another issue related to bellmouth condensation is that the condensed water acts like a scrubber for various impurities. The droplets and the wet blading and end-walls become acidic due to the scrubbed pollutants (such as CO_2, SO_2, NOx, HCl and Cl_2) – Hence, an elevated corrosion risk and need for protective coatings.

The need for proper filtration cannot be stretched too much since it will have a significant effect on the operational costs. One should always strive to have HEPA technology (or E11-class) for trouble free operation.

The filtration system is also of absolute importance for the entire engine (i.e. beyond the compressor); one good example is the littoral plant with airborne carryover of water droplets. Sea-water contains sulfur and sodium that together with oxygen will form sulfate ($2Na+S+2O_2 \rightarrow Na_2SO_4$). The combination of NaCl (and other alkali) and Na_2SO_4 is particularly pernicious since it produces a molten salt mixture already at 600°C. Hot corrosion is an extremely rapid process when an alkali metal like sodium reacts with sulfur to form molten sulfates. The principal damage mechanism is that the molten salts deplete/destroy the protective Al_2O_3 and CrO_3 oxide layers (from substrate diffusion). The situation gets even worse if other metallic salts are present containing V, Pb, Ca, K, Li, Mg as either fuel- or air-borne pollutants.

Fouling, corrosion and filtration issues will be addressed in a later section.

6 Recent developments

The engines presented in this chapter are both the identified units in the project specification and additional recently launched gas turbines.

6.1 Siemens SGT5-8000H

This is the latest engine in the Siemens 50 Hz portfolio and offers certified 60.75 percent efficiency in combined cycle. The gas turbine is H-class without steam cooling rated at 375 MW, or 570 MW in combined cycle operation. The design was initialed in 2000 and prototype operation started at the Irsching Block 4 (E.ON) in Germany in 2007. The gas turbine prototype test ended during summer 2009 and the plant is now turned into a single-shaft combined cycle with the highest efficiency (in-situ) in the world currently.

The engine is the world's largest at 13 times 5 meters, weighing 440 tones.

The engine is air-cooled which gives high operational flexibility and a short starting time. This was (when introduced) a deviation from current steam cooled G/H-technology used by General Electric and Mitsubishi. Rapid starting and ramping is gaining in importance with volatile electricity prices and security of supply by other CO_2-neutral production.

The engine is the first common design since the merger of Siemens KWU and Westinghouse. The intention was to combine the best practice from both companies' existing portfolios with advanced technology.

The entire first turbine stage and the fourth rotor blade can be removed and replaced without lifting the cover. This design feature is unique for this engine and offers a great time and cost saving when a replacement becomes necessary.

6.1.1 Performance data

Simple cycle data:

Power output	375 MW
Simple cycle effi'y	40 percent
Pressure ratio	19.2 –
Exhaust temperature	625 °C
Exhaust flow	820 kg/s
NOx	25 ppm_v@15% O_2
CO	10 ppm_v@15% O_2

Combined cycle data:

Power output	570 MW
Efficiency	60.75 % (certified by TÜV)
Steam data	HP: 170 bar/600°C, IP: 35 bar/600°C, LP: No info

6.1.2 Design features

The engine is a single shaft unit with a new twelve stage compressor with a specific flow of 820 kg/s @3000 rpm. The compressor is built from a single tie bolt and Hirth type serration for torque transfer, connecting the complete rotor and turbine. The compressor has four variable stages for flow control and low-speed stall avoidance. The compressor uses the latest blading technology and 3D design features. The unit has a can annular (or "cannular") combustor section, probably for easy 60 Hz scaling (lower count) and family concept. The lean premix system was scaled and optimized from the previous 60 Hz product range. The turbine has four stages with the first three un-shrouded and active clearance control. The active control is achieved by pushing the rotor inwards with a hydraulic system. This feature on a single shaft unit results in the necessity of cylindrical compressor blades. This gives both high turbine efficiency and rapid start capability. The fourth stage offers the possibility for a large exhaust and the AN^2 is assumed to be on the order of $55\text{-}60\times10^6$ for optimum performance.

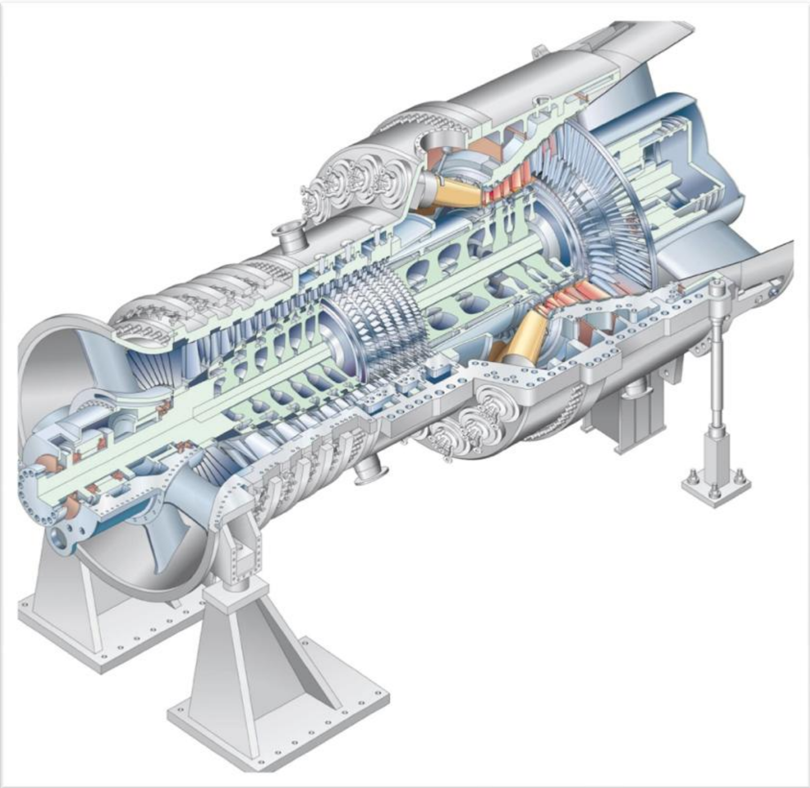


Figure 9. Siemens SGT5-8000H (Courtesy of Siemens press service)

The engine has three extractions at stage 5, 8 and 11 and one internal at the hub of stage 5. The outer extractions feed turbine stage 2, 3 and 4 whilst the internal is used for rotor thermal conditioning and purging of stage 4 rotor blade attachments. The design philosophy is an innovative "multi-use", where the cooling air from vane #2 is used to cool rotor #2 (same holds for stage #3). This approach saves cooling air and is ideally suited for stages with low cooling effectiveness requirement. Stage one has a normal full charge cooling concept. Stages 2-4 have modular cooling with valves to provide the right amount at all operating modes. This feature gives less losses when full cooling isn't required and the possibility to meet all cooling needs. Pre-swirlers are used in stages 1-3 for lowering the relative coolant temperature and minimizing the pumping power. The first three stages are directionally solidified (DS) with "angel wings" for good rim sealing. The blading of stage one and two uses a thermal barrier coating (TBC). The first two rotors have cast-in impingement cooling of the leading edge and pressure side cut-back.

The tests at the Irsching Block 4 site revealed no published issues.

6.2 Mitsubishi M701 G/G2/J/F5

Mitsubishi Heavy Industries (MHI) is a major player in the heavy frame segment with a total fleet of more than 535 units. MHI launched their latest J-class unit 2011, offering 61 percent efficiency (LHV) at 670 MW. The engine is fired some 100 °C hotter than current H-technology at 1500 °C. The simple cycle output is 460 MW at an efficiency of 40 percent. Within frame nomenclature, MHI has F, G, H and J covering 58-61 percent efficiency in a rather large power bracket. Their second generation G-class (M701G2) uses technology developed for the later H-class. MHI has introduced variants of steam cooling from G-class, except for the recent 60 Hz M501GAC. There is a strong driver for steam cooling in DLE-technology – since the flame temperature should be in the range of 1,500-1,600 °C for low emissions. A normal film-cooled combustor liner and transition piece requires large cooling flows to keep the metal temperature at a reasonable level. This air is typically mixed into the combustor post combustion and could be of better use in the front end providing more lean combustion.

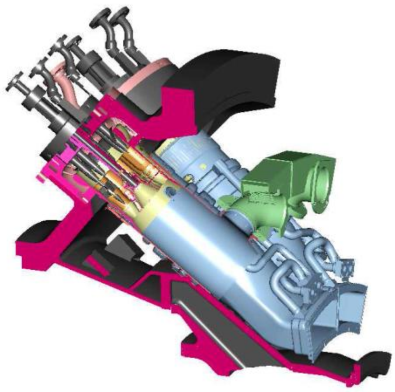


Figure 10. MHI steam-cooled liner and transition piece (Courtesy of MHI)

The G2-version also has an increased mass flow by 17 percent and an increased pressure ratio (21:1) for higher performance. The H-class engine uses a high level of steam cooling both in the combustor and turbine. The "back-flow" from H-technology to the G2-technology is the liner cooling.

6.2.1 G-series performance

	M701G	M701G2
	Simple cycle	
Power output [MW]	271	334
Efficiency [%]	38.7	39.5
Air flow [kg/s]		737
Exhaust temperature [°C]		587
	Combined cycle	
Power output [MW]	405	489
Efficiency [%]	57.0	58.7

6.2.2 G-series design features

The compressor has 14 stages with a pressure ratio of 21:1 resulting in an average stage pressure ratio of 1.24. The specific flow is 737 kg/s @ 3000 rpm. The rotor has 12 bolts both in the compressor and turbine. In addition to the bolts, additional radial pins and curvic couplings carry the torque. The turbine has four stages where the first two have cylindrical tip contour. This feature results is high levels of axial velocity ratios, but introduces the possibility to have minimum running clearances since the rotor is insensitive to the radial position. Row one and two also have an advanced clearance control. The third and fourth stages are shrouded for minimum leakage loss and good mechanical properties. The turbine outer- and inner wall (or hade) angles are within normal turbine practice. The cooling for the first stage is taken from the compressor discharge level, whilst the second, third and fourth stages are fed from the bleed system. The turbine disks are cooled with externally cooled air, taken from the compressor discharge. The stage three shroud is cooled since there is no combustor temperature profiling with a cooled liner and transition piece (aka smiley).

6.2.3 Air cooled G-class engine (M501GAC)

The steam cooling benefits in terms of DLE capability are eclipsed by the inherent limitation in the down-stream process. The steam is supplied from the HRSG and hence not available until a certain load is reached. Both the gas turbine and heat recovery units have their own sets of suitable loading gradients. Hence, steam cooling introduces an additional coupling causing significant longer starting and ramping. The driver for introducing steam cooling is a combination of available air for DLE, cooling efficiency and cycle performance. The air-cooled GAC engine offers G-class level of performance at the same firing level as for the steam cooled units of 1,500 °C.

The TBC technology for the GAC has been derived from the Japanese National Project for developing a 1,700 °C gas turbine. The aim of the project is to develop a combined cycle plant with 62…65 percent efficiency. The air-cooled combustor is derived from MHIs F-class.

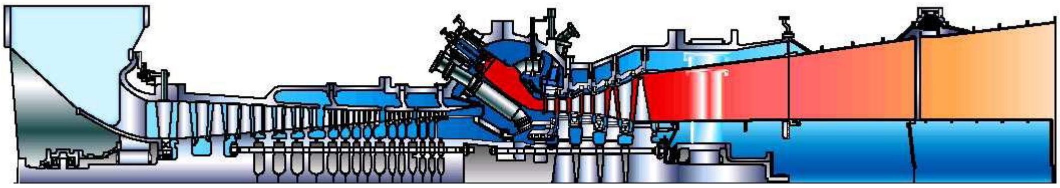


Figure 11. MHI M501GAC

A future 50 Hz version should have a power scale factor of 1.44, maintaining the same efficiency.

Tests at MHI T-point show that the GAC plant is capable to reach 59.2 percent efficiency.

6.2.4 The 1,600°C J-class

The latest commercially available engine from Mitsubishi is the 1,600°C J-class engine. The platform will be commercially available in 2011 and 2014 for 60 and 50 Hz, respectively. The engine has an output of 460 MW and is the largest gas turbine in production. The combined cycle offers efficiency in excess of 61 percent and follows the recent trend by the major OEMs.

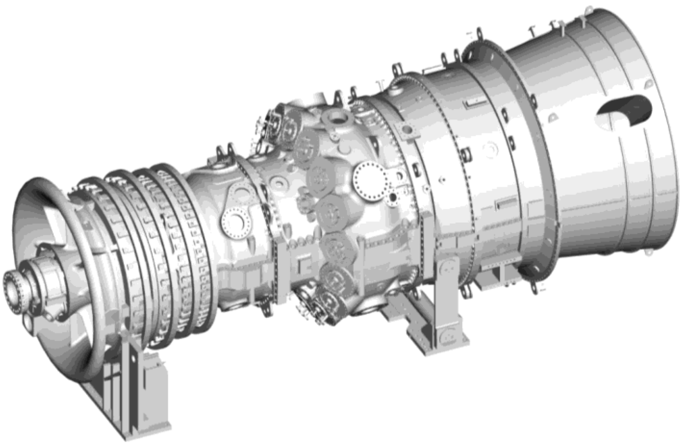


Figure 12. The latest MHI 1,600°C J-class engine

The engine firing level is the highest for a non-flying unit and the pressure ratio is 23:1. The latter figure is higher than the current G-class and it is safe to assume a redesigned compressor rather than just zero-staging or high-flowing. The J-class is an intermediate step before the Japanese National Project 1,700°C class engine. Some technical features from the 1,700°C class are carried into the J-class – like cooling technology and thermal barriers for maintaining the life span of the hot parts. There is a departure from current practice with single crystal (SX) material in the first turbine stage, where MHI has reverted back to directionally solidified (DS) blades. This has been driven by the indeed high costs for SX-blades and the fact that DS-alloys may be sufficient. The cooling air for stage one to three is pre-cooled by an external cooler. The usage of TBS's (thermal barrier coating) provides about 100°C "free" gas temperature in the cooling effectiveness balance.

The compressor in the J-class engine has 15 stages for a pressure ratio of 23:1. The 50 Hz mass flow is approximately 862 kg/s @ 3000 min^{-1} (1320 [lb/s]·0.436 [kg/lb]·$(60/50)^2$=862 kg/s) and is probably the highest in the business. This level of specific flow implies very high tip Mach numbers and this has been addressed by MHI by introducing three-dimensional blades. There are some indications of a forward-swept design for the first rotor for reduction of shock loss. The first four stages are multiple circular arc (MCA) blades – again indicating high velocities. The downstream stages are controlled diffusion airfoil (DCA) designs. The compressor has four variable stages for effective flow turndown and low speed stall margin. The design is derived from a MHI H-class steam-cooled rotor design with a pressure ratio of 25:1.

The J-class follows the practice by Mitsubishi with steam-cooled combustor liners. The combustor liner steam is also used to cool certain turbine stator parts, for clearance control. There are other rationales for introducing a steam cooled liner like the actual reduction in firing temperature for a certain combustor outlet temperature. A typical design may have a drop on the order of 100°C, whilst the steam cooled may be about half that value – hence less firing with less NOx etc. The engine is capable of 25 ppmv NOx and 9 ppmv CO, despite the high firing level.

The turbine follows current MHI practice with four stages. The two first stages are un-shrouded and most likely cylindrical for good clearance control. A cylindrical design is beneficial since one can design with lower clearances because its inherent insensitivity to the rotor axial position. The two last stages have shrouds for conventional leakage control and mechanical coupling. One can argue whether an early turbine blade should have a shroud or not. The general perception is that there exists a threshold firing value when the cooling of the shroud becomes too intricate and lossy. The 1,600 °C level is certainly above that level for a non-flying engine. The situation for a flying engine is quite the opposite, but the number of fired hours at 1,600°C in not even close to a land-based. The increased firing level capacity is a combination of advanced TBC (and bond coating) and cooling technology. The turbine has advanced three-dimensional end-wall contouring for optimum performance.

The engine was brought on-line at the T-point station in June 2011 and had accumulated (per December 2011) 1,700 actual running hours and 45 starts. The MHI T-point station was commissioned in 2007 and is a commercial plant where the power is dispatched by the local utility. The T-point is in the 60 Hz part of Japan so the size of the tested units is 1.2 times smaller than for 50 Hz units. The T-point station was prior to the J-class test a G-class plant/test platform. The possibility of having a fully instrumented engine on-line on your own backyard is indeed valuable when it comes to validation of a product. The shear fuel burn would have rendered extended non-grid operation impossible without balancing the economics by selling power to the grid.

MHI has published results from the test operation during 2011 at the PowerGen conference 2011. Their presentation revealed no issues related to the firing level or DLE pressure pulsations.

6.2.5 High performing air-cooled F-class (F5)

Mitsubishi has also followed the flexibility trend by introducing a high-performing all air-cooled version of their F-class. The new engine is called F4 and is rated at 350 MW. The combined cycle power is above 500 MW with an efficiency of 61 percent. The design is based on combining features from F4, GAC (air-cooled G-class) and the latest J-class. The firing level is the same as for GAC and the guaranteed NOx level is 15 ppmv.

Gas turbine	M701F4	M701F5
Gas Turbine Power [MW]	324	350
Gas Turbine Efficiency [%]	39.9	>40
GTCC Power [MW]	478	525
GTCC Efficiency [MW]	60.0	>61

The compressor is basically a re-bladed F4 compressor where the mass flow and meridional contour is kept at the original level. The F4-compressor has NACA-65C series in the middle and rear part (stage 7-16) of the compressor. The 65-series of blades were designed in the late 50s and more recent CDA-blades (Controlled Diffusion Airfoils) offers higher efficiency and incidence range.

The combustion section is derived from the 60 Hz air-cooled G-class engine (GAC). The firing level is the same as the GAC-unit and has been tested previously at the T-point.

The turbine is based on the J-class and the GAC-engine, carrying more recent technology into the F-platform. The firing level is lower than for the J-class and the cooling system for the two first stages has been adapted. The two first stages are based on the J-class (and GAC) with its advanced TBC-and film cooling technology. The last two rows are based on the GAC-engine and the last stage is un-cooled. The F5-unit uses the same materials throughout the turbine as the GAC.

6.3 Alstom GT26

Alstom released their upgraded GT24/GT26 platform in 2011. The upgraded engine follows the current trend with 61 percent combined cycle efficiency at 500 MW power. One unique feature is that the improvements can be retrofitted into the bulk of the fleet (all units after the 2002 compressor upgrade). This is currently not the case for any other OEM – since they have changed the architecture significantly with e.g. different stage counts in the compressors and turbines.

The type of unit was introduced in the 90s and has been upgraded three times before the current version. The first (or B-version) was introduced in 1999 for solving hot end issues. The 2002 and 2006 incorporated (among others) compressor modifications within the meridional contour by re-staggering and profile modifications – for higher power output. The 2006 upgrade also included increased firing level for the second burner.

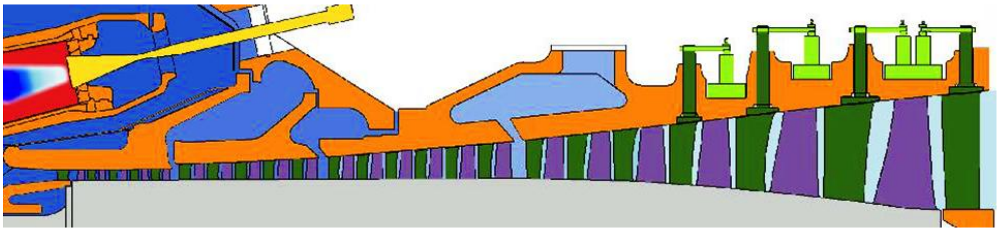


Figure 13. Alstom GT26 compressor section X-section[14].

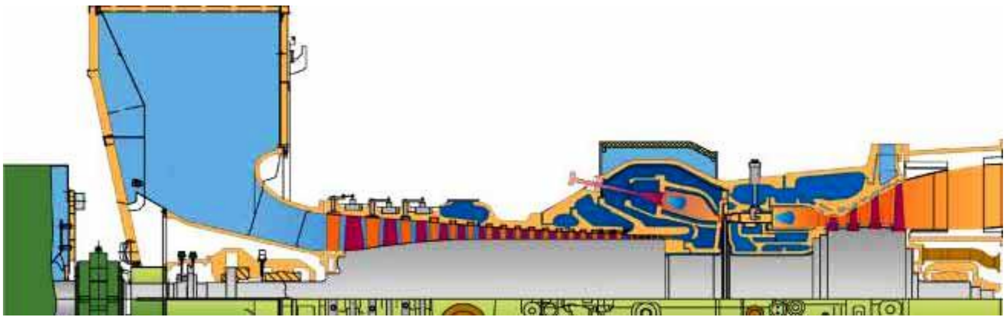


Figure 14. Alstom GT26 X-section.

The KA26 (kombi anlage) can be restarted (eight hours shutdown) within 30 minutes and reaches 350 MW in 15 minutes. The sequential firing system offers the possibility of low-load parking at a minimum load. The minimum load is most likely dictated by the steam turbine exhaust ventilation / turn-up.

6.4 General Electric 9FB.05

The latest FlexEfficiency™ plant by General Electric follows the trend with its high efficiency and flexibility. The power output is 510 MW with an efficiency of 61 percent. The start-up time from hot-overnight is 28 minutes to base load and the ramp rate is 50 MW/min. The 338 MW engine is based on a

[14] The Next Generation KA24/GT24 From Alstom, The Pioneer In Operational Flexibility Sasha Savic, Karin Lindvall, Tilemachos Papadopoulos and Michael Ladwig

scaled 7FA.05 compressor and a new four stage turbine section. The 7FA.05 is the newest high performing 60 Hz frame.

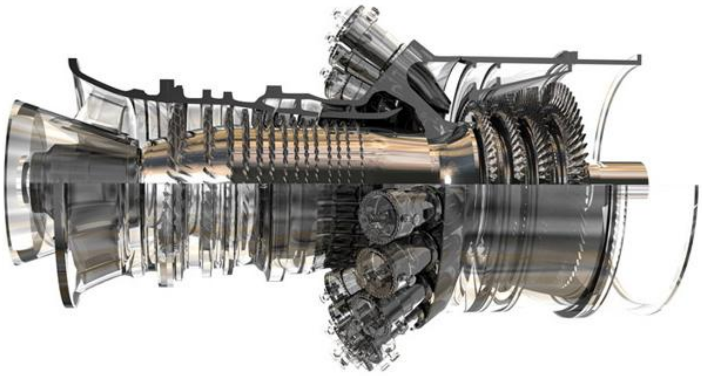


Figure 15. General Electric 9FB.05 (Courtesy of General Electric[15])

The latest version is a radical improvement from the previous version. Both the compressor and turbine have been redesigned. The former was actually an up-scaled and zero-staged E-frame compressor. The new design has a higher pressure ratio of 19.7:1 with 14 stages (approximately 1.24 on average per stage). The previous 9FB.03 had a pressure ratio of 18.3 with 18 stages (approximately 1.18 on average per stage). Its clear heritage from aero-engine technology follows the current trend – like the new GT26. The firing level appears to be slightly higher than the previous and should be around 1,465°C. General Electric always quotes the firing level as rotor inlet temperature and the actual combustor outlet is typically on the order of 100°C higher. The turbine has four stages in contrast to the previous version that had three stages. The driver for the new four stage design is most likely the increased pressure ratio, increased fining, increased mass flow and efficiency. The four stage design probably offers higher ANSQ-capacity by virtue of the lower temperature.

The combustion system is the DLE 2.6+ and the turn-down, with guaranteed emission, is down to 40 percent plant load. The guaranteed levels are 50 mg/Nm^3 and 30 mg/Nm^3 for NOx and CO, respectively. These levels are, loosely stated, about half in ppmv@15%O_2.

The heat recovery steam generator (HRSG) is an all-drum technology – despite the indeed fast start-up time. It appears that the most critical components, with respect to LCF, are the header attachments (HARP) for the superheater and the reheater.

The gas turbine will be tested at full load at the General Electric test site in Grenville in the US during 2012.

[15] FlexEfficiency* 50 Combined Cycle Power Plant

	Current GE Fr 9FB.03 plant	Flex 50 GE Fr 9FB.05 plant
Introduction year	2002	2014-2015
Power@ISO [kW]	284,000	338,000
Gas Turbine Heat Rate (LHV) [kJ/kWh]	9,512	<9,002
GTCC Heat Rate (LHV) [kJ/kWh]	5,829	<5,595
Gas Turbine Efficiency [-]	37.9	>40
GTCC Efficiency [-]	58.6	>61
Pressure ratio [-]	18:1	19.7:1
Exhaust flow [kg/s]	655	745
Exhaust temperature [°C]	642	623
Admission data [bar(a)/°C]		165/600 X/600 X/X
Steam Turbine Power [kW]	162,100	180,000

6.5 Pratt & Whitney FT4000

Pratt & Whitney has released their new FT4000 120 MW platform rated at above 41 percent simple-cycle efficiency. The product follows Pratt and Whitney's practice with two three-shaft gas turbines to a common generator. The turboset is a derivative from the flying PW4000 engine. The twin engine configuration offers per se higher part load efficiency since one engine can be working at full load. No information is available at the time of writing whether the unit can be operated in synchronous condensation mode with or without a SSS-clutch.

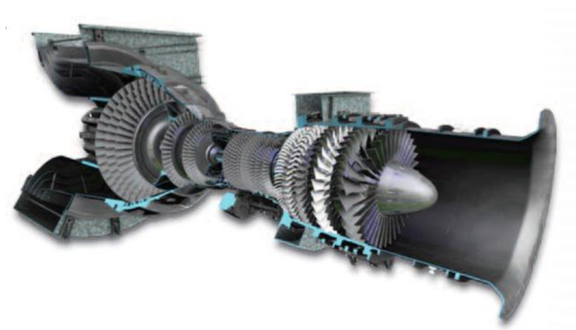


Figure 16. Pratt & Whitney FT4000

The FT4000 plant can be started and loaded within 10 minutes without maintenance penalty. The efficiency is claimed to be "above" 41 percent and the plant lags four points when compared to the LMS100 by General Electric. The P&W FT4000 is a traditional simple cycle unit and the "lag" in efficiency is most likely due to a lower pressure ratio. The General Electric LMS100 is offered at 45 percent efficiency with a cycle pressure ratio on the order of 42. This high level requires a bulky (about the size of a city bus) compressor intercooler for optimized performance. The LMS100 inter-cooler duty is within the range of 20-25 MW. Inter-cooling will only increase the efficiency at very high pressure ratios. A detailed discussion whether to inter-cool or not is outside the scope of this report. A few words will, however, be given for completeness. Inter-cooling will reduce the compressor work (directly proportional to the inlet temperature in degrees Kelvin) – but the lower discharge temperature requires more fuel burn for the same firing temperature. Above a certain combination of pressure ratio (discharge temperature) and firing level, the trade-off is positive and the efficiency improves by introducing inter-cooling.

6.6 Solar Titan 250

The newest Solar™ turbine was launched in 2006 and offers 22 MW at an efficiency of 40 percent. The unit is a twin-shaft and is suitable for both power generation and mechanical drive. The power turbine maximum speed is 7,000 min^{-1}. This engine is certainly a game changer in terms of efficiency in the lower 20 MW power bracket. The new Titan 250 has the same footprint as the older Titan 130, but delivers 50 percent more power. Solar has sold 17 units since the introduction and the fleet leader has (per December 2011) 15,000 operating hours.

6.6.1 Compressor

The pressure ratio is 24:1 with 16 stages resulting on an average stage pressure ratio of 1.22:1 per stage. The speed is 10,500 rpm, resulting in a compressor flow at 3000 rpm[16] of 820 kg/s. The level suggests a moderate to high tip Mach number level. The level is actually the same as for the 375 MW Siemens SGT-8000H. The level seems too high for a geared single shaft unit since the normal range of AN^2 would cause a too high exit Mach number from the turbine. Hence, one cannot expect a single shaft version of the Titan 250.

6.6.2 Combustion system

The firing temperature has not been published, but the engine power density suggests a firing level of 1,300°C (±50°C).

[16] The 3000 rpm value is defined as: and is a convenient measurement of the technology level when comparing axial flow compressors. It tells what flow one would have for a 3000 rpm design with the same Mach-number based velocity triangle – hence a linear scaling to 3000 rpm maintaining all components of the triangle.

6.6.3 Turbines

The compressor turbine has two cooled stages. The rotor blades are cylindrical for minimum running clearance.

The power turbine has three stages and the design speed is 7,000 min^{-1}. A three-stage design at this speed levels indicates a high efficiency potential over a large speed range. All three stages are shrouded for optimum performance and range without frequency issues.

6.7 Rolls-Royce RB211-H63

The newest RB211 engine was presented in 2010 and offers a wet rating of 44 MW at 41.5 percent efficiency. The dry rating is around 37 MW with similar efficiency level. The engine has the same architecture as the rest of the RB211 range, i.e. traditional three-spool concept (LPC/IPC–HPC–CC–HPT–IPT–PT).

The engine follows the Rolls-Royce common concept of high commonality with their flying parent engines.

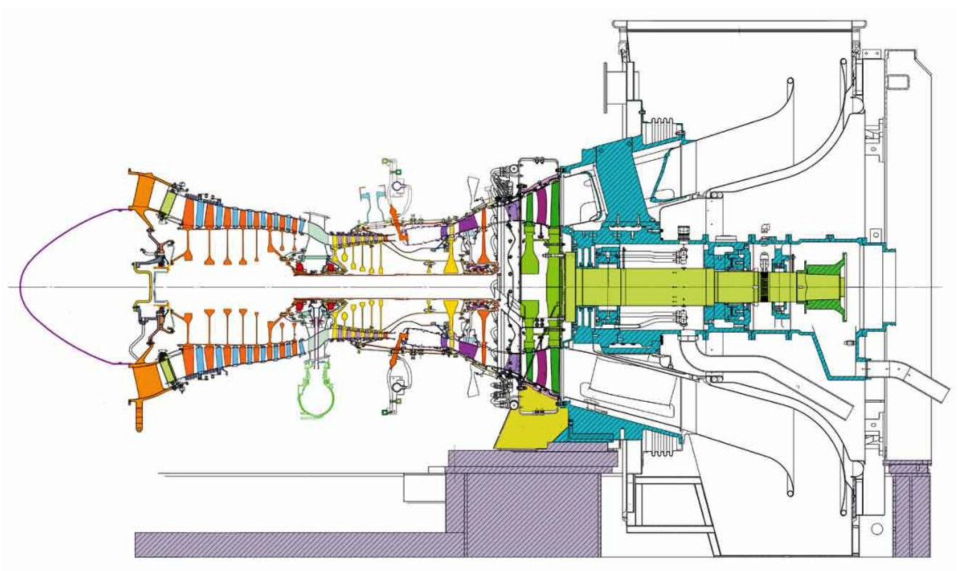


Figure 17. Roll-Royce RB211-H63 X-section[17]

6.7.1 Compressor section

The low pressure compressor has 8 stages and is derived from the industrial Trent engine. The casing has been taken from the aero-engine Trent 800 and has been redesigned/simplified to fit the needs of a land based engine. The casing has a horizontal split for easy maintenance.

The high pressure compressor has 6 stages and is also derived from the industrial Trent (and Trent 800) engine. The pressure ratio is 25:1

[17] Rolls-Royce, The RB211- H63, The winning formula for increased power and efficiency

6.7.2 Combustion system

The engine was launched with a wet low emission (WLE) system in 2010 but will have dry low emission (DLE) system available in 2013. The current WLE-system is the same as for Trent 60 and aero Trent 800.

6.7.3 Turbine section

The turbine section follows the traditional three-spool concept with a single stage high turbine, single stage low turbine and a two-stage power turbine. The power turbine speed is 6000 min^{-1}, this level is quite typical for pipeline compressors in this power range. No information related to the low and high spool speed has been presented. A reasonable guess is on the order of 7,500 and 10,000 min^{-1}, respectively.

7 Some Aspects of Gas Turbine and Plant Maintenance

The second largest cost over a plants life cycle is the operation and maintenance (O&M) spending. The O&M cost over 25 years of operation may be twice the first cost of the equipment. There are several ways of handling scheduled and un-scheduled maintenance costs and associated risks. There are "typical" levels of contractual services offered by the OEM/contractor ranging from parts agreement and technical advisory to full "bumper-to-bumper" services regardless of the occasion.

There are third party companies' also offering the same type of parts and long term service agreements (LTSA) – but one should be aware of the strengths of an OEM. In some cases a third party part and associated services (e.g. reverse engineering of parts) may be a superior and cheaper choice. This especially holds for older units, but it is not possible to draw firm conclusions. The OEMs dominate the aftermarket with a share of 57% (2009).

The duration of an LTSA is typically two major maintenance cycles or 80,000 to 100,000 hours. There are other constructions suitable for e.g. peakers where the number of events may replace time.

In some cases like IPP's, financing organs or insurer may require a LTSA (or higher) for risk mitigation or leveling the insurance cost at a reasonable level.

One should also bear in mind that the aftermarket is, indeed, valuable to the OEM. The margin for certain key parts may (manufactured in-house) very well be on the order of several hundred of percents in contrast to typically ten for a complete new turn-key plant. *The total aftermarket spending was 13.82 B€ (13.82×10^9) or 18.3 BUSD (18.4×10^9) in 2009*[18]. The total world gas turbine fleet is on the order of 46,500 units – all requiring maintenance at certain points.

The reward, from having contractual services, is discounted parts and prioritized treatment by the supplier – hence quid pro quo.

7.1 Level of provided contractual services

The lowest level is parts agreement, where the OEM/contractor is the exclusive provider of parts and perhaps rejuvenation – with or without technical advisory. The number of parts and refurbishments are fixed according to the predictive maintenance plan, but may extend to unforeseen events. The plant provides staffing and the contractor typically has an advisor present during overhaul. The utility typically carries the risk for unplanned events and collateral damage. The direct OEM/contractor warranty is typically limited to the replaced- or rejuvenated parts itself. In some cases, the actual refurbishment of the used parts is a separate issue.

[18] The Industrial Gas Turbine Global Maintenance Market, Aero Strategy.

The next level is the classic long-term service agreement (LTSA) where the OEM/contractor has further contractual obligations. The LTSA typically covers spare- and refurbished parts for all planned (or scheduled) maintenance events. The OEM/contractor provides relevant staff and supervision. The plant owner probably still have to carry the risk associated with unplanned events (or forced outages) and collateral damage. The typical OEM/contractor exposure for collateral damage is limited to the excess clause in the insurance cover.

The next level is the contractual service agreement (CSA) or term warranty, where the OEM/contractor provides all parts for maintenance. The word *all* is used in a sense that parts for both planned and unplanned (forced) maintenance are included. The OEM/contractor also carries a significant part of the risk for collateral damage. This type of contract is an incentive for introducing various levels of condition monitoring systems. The risk for the OEM/contractor can be mitigated and there is also an increased possibility to avoid collateral damage. Most engines have sufficient instrumentation and the associated true hardware costs are quite small.

The range of contractual services is quite wide and it is not possible to give firm recommendations. More of the hardware failure risk is transferred to the OEM/contractor for each level. Some users have competent maintenance organizations and culture, and are capable of major engine work. In this case, the parts-only contract may provide a sufficient level of OEM/contractor support. For another organization, where the maintenance organization is limited to daily routine work, the CSA may be the right choice. Again, the funding institution or the insurance company may require a certain level of OEM/contractor services for risk mitigation or insurance cost, respectively.

7.1.1 LTSA

Two articles in 2003 (February and March) in Power-Gen Worldwide lists ten LTSA contractual pitfalls and instructions of how to avoid them. The reader is referred to the two articles by Thompson and Yost for a complete cover. A list of their ten points will, however, be given here for completeness:

1. Clearly Defining Scheduled Maintenance
2. Clearly Defining Extra Work
3. Appropriately Allocating Prolonged Start-up Risks
4. Protecting Owner Interests in the Absence of Performance Guarantees
5. Clarifying Responsibilities Between Unscheduled Maintenance and Warranty Obligations
6. Early LTSA Cancellation
7. Extended End-of-Term Parts Life Warranty
8. The Absurdly Long LTSA
9. Liquidated Damages for Termination
10. Limitations of Liability and Exceptions

A couple of caveats and comments are in place related to the previous list. The first point may seem obvious but the risk exposure from such a lack of clarity may indeed be very costly. The whole point of having a LTSA is that the contractor supplies all parts and carries the associated risks – simply to having to avoid providing small bits and pieces. It is simply not desirable to delay the outage over a missing consumables like a small hydraulic filter, solenoid, seal, fuse,… The list can be long but the striking point is that it should be clear who carries the responsibility prior to the events. The second item of the list is "extra work" and to give an example one could consider the following case – what if the owner/user wants to replace a part prematurely than stipulated in the LTSA? A clear definition of "extra work" could provide a framework for not having to go through new negotiations every time that this occurs.

A further extension could be that the OEM/contractor operates the plant and carries out daily maintenance work for the entire plant. Daily maintenance typically includes consumables like filters and wash detergents. This concept may be attractive to merchant plants since the plant can be operated without own staff. The figure below shows different concepts from General Electric – ranging from "loose parts" and advisory, classic LTSA/CSA, operation and maintenance to include full engine performance.

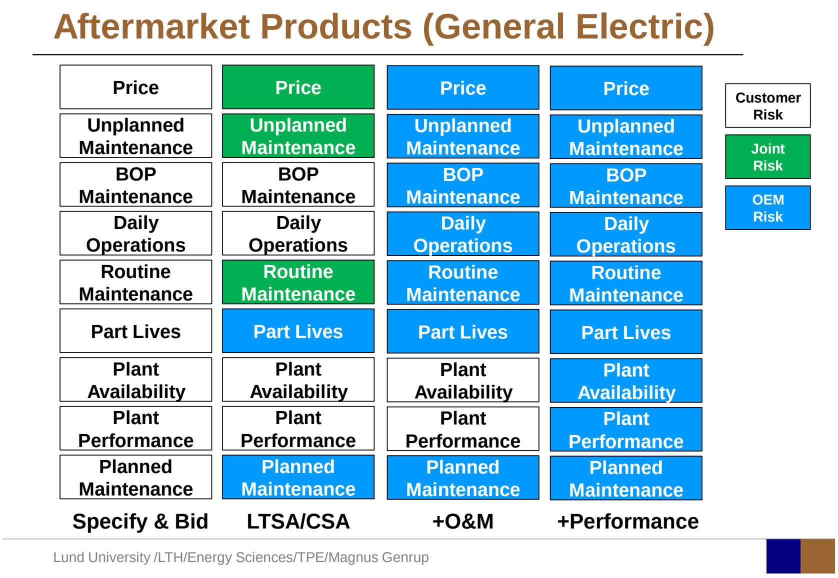


Figure 18. Different levels of contractual services.

Another, indeed important thing is the type of engine since there is a huge difference in complexity involved with different generations and types of engines. For example, a combustor liner replacement is more or less straight forward on an engine with flame tubes. The situation is quite different for an engine with an annular combustion system, where the turbine section has to be removed. Removal of the turbine section certainly requires OEM/contractor staff in situ and probably field balancing before start.

Most of the previous descriptions are for heavy frame types for which the engine has to stay at the plant. Lighter units, like aero-derivatives and light industrial, may be transported to a dedicated service shop. Some owners have an own spare engine for swift replacement. This luxury comes with a certain price and is probably only feasible for an operator with a large fleet of a certain engine – or if the engine is a part of a critical system. The latter is typically found in the oil and gas industry, where gas turbines are used for either electricity production or mechanical drive. The cost of a spare engine may be small compared to a loss of production revenues, during sometimes even a short duration. Some LTSA contracts offer a replacement engine during overhaul, hence providing the same level of availability. The concept of engine swap will certainly offer advantages in availability since most engines can be replaced within 24 hours after arriving at site. The latest Siemens engine (SGT-750) is claimed to offer as low as 17 days of maintenance during 17 years.

If the OEM/contractor has full control of both replacement parts and staff, then OEM/contractor should be able to guarantee both time and duration for planned events.

7.1.2 End of term

Any level of contractual parts- or service agreement has to have an end of term agreement. This could state that the OEM/contractor provides parts to the next planned inspection event, or even, a brand new set of hot parts.

An unclear situation may be indeed very costly since major refurbishments or even replacements may be required.

7.1.3 Open vs. closed pool

Open and closed pool parts are another choice for an operator. Having an open pool contract means that the operator only owns the parts when they are in their engine. This is in contrast to the closed pool, where the operator uses their own set of parts throughout the useful life of the engine. There is a cost saving potential associated with the open pool concept for the user if one is willing to use "someone else's" refurbished parts. The pedigree of the parts is important since one can introduce new life ending mechanisms into the engine. On example is a base load machine with very low number of starts per fired hours, where creep and oxidation is the predominate mechanism for fall out. A peaker, on the other hand, has a high number of starts per fired hours and the fall out mechanism is typically HCF-driven cracks.

7.1.4 Condition-based maintenance

The concept of condition-based maintenance (CBM) is an extension of condition monitoring where the actual end of life of a certain component sets of a certain maintenance event. This is in contrast to the normal time-driven, where parts are changed/refurbished in certain time intervals. The word time is used in a sense that it reflects consumed life time and not necessarily clock time. When designing a part e.g. a rotor blade, one has to make a tradeoff between material cost and cooling technology. The minimum acceptable creep and oxidation life is somewhere between 24,000 to 50,000 hours. The OEM has to make sure that the blade maintains a suitable temperature so that the expectations can be met. Temperature gradients are the prime source for

LCF-fatigue and a totally different mechanism. All lumped methods where cycles are transformed into time should therefore be avoided. The concept of just bookkeeping some kind of factored hours and events gives a rather conservative, but blunt approach. Instead, a system is set up to evaluate the components actual environment and evaluate the consumed life.

The obvious reason for an OEM for introducing conditions-based maintenance within LTSA- and CSA contracts, is less parts during the contract term. The advantage for the user could be more availability and perhaps less O&M spending. The risks associated with this more aggressive way of assessing remaining life are forced outages and collateral damage.

CBM together with direct blade temperature measurement offers an additional level of improvement – especially since the hottest blade can be identified in an on-line fashion.

7.2 Engine maintenance

All gas turbines have some kind of cyclic maintenance intervals. There are typical structures for heavy- and light units. The common approach is to set of appropriate maintenance actions after certain time intervals.

7.2.1 Example of definition of separate time and number of events

There are different ways of calculating the consumed life of an engine depending on the operational profile. The examples below are based on a publication by GE[19]. The interval for hot gas path inspection (HGP) is nominally 24,000 hours or 900 starts.

Factored hours for evaluating maintenance intervals

$$\text{Maintenance interval} = \frac{24000}{\text{Factored Hours}/\text{Actual Hours}} \text{ hours}$$

Factored Hours = $(K + M \cdot I) \cdot (G + 1.5 \cdot D + A_f \cdot H + 10 \cdot P)$

Actual Hours = $(G + D + H + P)$

G = Annual Base Load Operating hours on Gas Fuel

D = Annual Base Load Operating hours on Distillate Fuel

H = Annual Operating Hours on Heavy Fuel

A_f = Heavy Fuel Severity Factor (Residual = 3 to 4, Crude = 2 to 3)

P = Annual Peak Load Operating Hours

I = Percent Water/Steam Injection Referenced to Inlet Air Flow

M&K = Water/Steam Injection Constants (see GE documentation)

[19] Balevic et al.,Heavy-Duty Gas Turbine Operating and Maintenance Considerations, General Electric, GER-3620K (12/04).

The preceding example shows that the maintenance interval is 24,000 hours or 2.7 years (two years and eight months), when continuously operated at base load on natural gas without injection. The penalty due to peak load is 10 factored hours per actual hour.

The equation has no credit for part load and the minimum factored hours is one per true fired hour. The actual life consumption should follow an Arrhenius type of expression[20]. If one assumes that the factor of ten is valid for a 56°C increase in firing temperature and the nominal firing has a value of unity – then a reduction of 56°C should give a number on the order of 0.5. Hence, one cannot "balance" one hour of 56°C peak with on hour of 56°C (reduction from nominal) part load for compensating the consumed life. With the current assumptions, the hours for balancing should be on the order of 20. In earlier work at LTH, where the factor is set to 6:1 at 35°C and 36:1 at 111°C, the resulting equation was:

$$MF=e^{0.0323 \times \Delta COT}$$

Any competent condition-based maintenance system should be able to keep track of such phenomena. Most combined cycle plants maintain the nominal firing down to a maximum exhaust temperature and the load level is instead controlled by the compressor variable geometry. The exhaust is actually *hotter* at part load than at nominal load. This means that the engine is running at the same firing level at a higher exhaust temperature at part load. The last blade has the highest P/A stress level and should be relative more sensitive to an elevated temperature level. Fortunately, the last rows relative inlet total temperature stays the same since the pressure level is decreased. The exact underlying aerothermal principles are outside the scope of this report.

Lund University has worked on another approach where the primary control is the admission steam temperature rather than EGT and firing level[21]. This offers significant life improvement at the expense of only minute changes in cycle efficiency. The reason is the steam turbine, which typically has a maximum admission temperature of e.g. 565 °C and the associated admission cooling with sprays. The undelaying thermodynamics can be analyzed on first-law principles but no details are given here.

When a unit is operated on syngas or with any "non-conventional" fuel, then an appropriate factor has to be introduced into the preceding factored hours equation. The magnitude of such factor is not possible to discuss in general terms since, more or less, the complete periodic system can be present. There are factors for crude oil and residuals of 2-4 and 3-4, respectively.

[20] Creep properties are normally correlated with the Larson-Miller parameter (LMP) as: $LMP=(C+Log_{10}Life)\times(T+273.5)\times10^{-3}=f(P/A\ stress)$, where C is a constant (typically on the order of 20-22). This Arrhenius type of expression is the base of the exponential behavior of lifing vs. firing for creep.

[21] Jonshagen, Modern Thermal Power Plants – Aspects on Modeling and Evaluation.

Factored events for evaluating maintenance intervals

$$\text{Maintenance interval} = \frac{900}{\text{Factored Starts}/\text{Actual Starts}} \left[\text{Starts}\right]$$

$$\text{Factored Starts} = 0.5 \cdot N_A + N_B + 1.6 \cdot N_P + 20 \cdot E + 2 \cdot F + \sum_{i=1}^{\eta} \left(a_{Ti} - 1\right) T_i$$

Actual Starts = $(N_A + N_B + N_P)$

S = Maximum Starts-Based Maintenance Interval (Model Size Dependent)

N_A = Annual Number of Part Load Start/Stop Cycles (<60% Load)

N_B = Annual Number of Base Load Start/Stop Cycles

N_P = Annual Number of Peak Load Start/Stop Cycles (>100% Load)

E = Annual Number of Emergency Starts

F = Annual Number of Fast Load Starts

T = Annual Number of Trips

a_{Ti} = Trip Severity Factor = fcn(Load, Trip during accel. = 2, Peak = 10)

η = Number of Trip Categories (i.e. Full Load, Part Load, etc.)

The preceding set of equations shows that a normal start counts for one cycle whilst a fast start counts for two. The algorithm also takes into account the final load (or firing) level, where less than 60 percent gives a factor of half and peak at 1.6. Both numbers in concert, gives a gauge of the overall temperature gradient and level. The most severe events, by far, are trips that give as a minimum two events during start-up and 20 at peak load.

There is a separate set for rotor inspections. The typical intervals are 6 times less than the HGP for factored hours and starts.

The reasoning can be visualized by the figure below. The previous set of equations is used to calculate how fast one is moving along the ordinate and the abscissa, respectively. The three “typical” running profiles (peaker, mid-merit and base load) are plotted into the figure. The duration between each event is set by the previous set of equations.

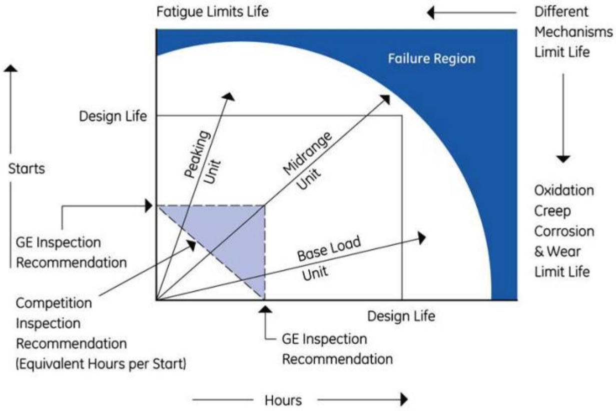


Figure 19. Lifing diagram for a heavy frame (Courtesy of GE).

7.2.2 Payment for LTSA and CSA

The payment is typically coupled to either the number of factored hours or factored starts plus a fix monthly fee. This introduces issues if the operational profile changes from e.g. base load to peaker, and vice versa. Another option is payment at each planned maintenance event plus a monthly fee. The former offers a “levelized” cash flow, whilst the latter gives lumped payments each 8,000 hours.

7.2.3 Example of equivalent operational hours

The equivalent or lumped operational hours gives a single figure for determining when to take appropriate maintenance acts.

$$EOH = \underbrace{\sum_{1}^{OH} 1 \times F_{fuel} \times F_{firing}}_{creep and oxidation} + \underbrace{\sum_{1}^{n_{starts}} 1 \times F_{starts} \times F_{load rate} + \sum_{1}^{n_{trip}} 1 \times F_{trip}}_{LCF}$$

Where:

EOH	equivalent operating hours
OH	actual operating hours
F_{fuel}	factor depending on fuel
F_{firing}	factor depending on firing level
n_{starts}	number of “fired” starts
F_{starts}	number of hours per start
$F_{load\ rate}$	load rate factor

As already mentioned, the method of equivalent hours should be avoided since the two life consuming mechanisms do not rigorously behave in an additive or accumulative fashion.

7.2.4 Inspections and intervals

The scope and intervals vary between the manufacturers and types of engine. The large- or major inspection typically comes each 40,000 to 50,000 hours. The terminology varies between the OEMs and even between their engines.

The *typical* heavy frame sequence is depending on operational profile:

Hours-based: **CI – CI – HGP – CI – CI – MI – CI – …**

Starts-based: **CI – HGP – CI – HGP – CI – MI – …**

Where:

CI	Combustion inspections
HGP	Hot Gas Path Inspection **N.B. per calculations above**
MI	Major Inspection

If the unit is operating as a base load engine then there is a maintenance event (or CI) each 8,000 factored hours and a hot gas path each third time. The second hot gas path (HGP) inspection is referred to as a major inspection (MI). There is a similar structure in the start-based, where each 450^{th} factored start sets of a CI or HGP. The major inspection is carried out every second cycle or 1,800 starts.

There is a more detailed method for calculating the combustion inspection (CI) interval for different types of equipment, but it follows the previous structure.

A light unit or aero-derivative *typically* has a similar structure:

Hours-based: **BI – BI – HGP – BI – BI – MI – BI – …**

Where:

BI	Boroscope inspections
HGP	Hot Gas Path Inspection
MI	Major Inspection

Most aero-derivatives have no maintenance requirements due to cycles.

7.3 Maintenance scope

Having established the time frames associated with planned maintenance, the following section will give a brief description of the involved scopes. Each engine type has its own inspection and service program. This section does not claim to be exhaustive in terms of scope or methods.

7.3.1 Combustion inspection (CI)

The combustion inspection is the lowest frame level and typically includes:

- Inspection of all combustion chambers, cross-fire tubes and transition pieces (a.k.a smilies).
- Inspection of thermal barrier coatings (TBCs) for spallation, wear and cracks.
- Visual inspections of the first turbine guide vane and boroscope inspection of the first rotor.
- Compressor boroscope inspection.

The word inspection is used rather loosely and should typically include: abnormal wear, foreign objects, cracks, TBC issues, loss of material[22], hot spots, etc.

Replaced parts are either scrapped or refurbished for the next maintenance cycle. The typical refurbishment for a combustor part is weld repair and TBC stripping/replacement.

7.3.2 Hot Gas Path Inspection (HGP)

The second level is the Hot Gas Path Inspection where it is normal to change or rejuvenate parts in the hot flow path. Again, the exact scope is unique for each engine type. The HGP typically includes:

- Same as CI, plus:
- Replacing or repairing the first turbine stage guide vanes and blades. This requires either lifting the turbine casing or removing the turbine section for smaller engines. The newest Siemens H-class engine offers the possibility of replacing the first stage from the transition ducts – hence no need for lifting the cover to replace a blade.

Typical refurbishment/rejuvenation for turbine blades are: welding, general machining (e.g. blade profiles, new squiler tips, shrouds), stripping and recoating. This level of work has to be performed in a qualified shop, capable of e.g. metallurgical tests and flow testing of cooling passages etc. Stripping and recoating requires heat treatment that potentially can affect the single-crystal base material in a negative way. Weld repair could also be applicable on highly structurally loaded parts of blades (in contrast to e.g. a squiler tip). This, however, requires a full blade stress analysis. A caveat is that, the creep properties of a weld is quite often much worse than the base material. A

[22] Corrosion, oxidation and erosion

"qualified shop" may certainly be a dedicated third party supplier/contractor. One operator in the US always takes one of the refurbished blades into destructive testing before returning the set into the engine. This provides additional confidence in third party work, at the expense of a new blade. In some cases, the contractor may be another prominent competing OEM (or market companion). On example is Pratt & Whitney (P&W), which manufacturers' new parts for the General Electric frames and aero-engine fleets. P&W have also been successful in rejuvenation of GE F-class blades. Another is Mitsubishi that manufactures parts for the old 60 Hz Westinghouse (now Siemens) fleet. The list of competent OEMs and refurbishment contractors can be made very long.

This level of inspection for an aero-derivative or light industrial typically means transportation of unit or the gas generator to a dedicated shop. The cycle for the power turbine typically follows the one for the rotor (i.e. 100,000 to 120,000 hours). As earlier mentioned, the availability can be improved significantly with a lease engine. This is normally referred to as "engine swap" and offers downtimes down to 24 hours per event. The latest Siemens product SGT-750 offers a low number as 17 days in 17 years for maintenance.

7.3.3 Major Inspection (MI)

The second large inspection is the Major Inspection (MI), which could be seen as an extended HGP. The major inspection is an inspection/overhaul from the inlet to the outlet. The major inspection typically includes:

- Same as HGP, plus:
- Complete turbine overhaul
- Compressor blades overhaul
- Bearings and seals
- Rotor inspection

7.3.4 Boroscope Inspection (BI)

The term Boroscope Inspection is normally used on aero-derivatives and light industrials. The duration of such inspection is often quite short – on the order of a day.

7.4 How can an operator influence maintenance spending?

The question of "how to control your maintenance spending?" is not straightforward to address. One extreme would be not to consume any life but the revenue for direct power production would also be zero. Most gas turbines before the advent in the 80s of large quantities of comparably cheap and relative environmentally friendly natural gas were almost never operated. The bulk of the Swedish gas turbine fleet has very few operational hours on a per annum base. Instead, these units serve the non-spinning reserve and are typically used for synchronous condensation and emergency back-up. The situation is quite different on a worldwide base, where gas turbines stand for significant parts of the power production. In the US, in both Texas and California, the gas turbines produce about half of the electricity. The advent of high levels of volatile power will also be a game changer.

On a more practical level, the user has some control of the life consumption. One could chose not to dispatch the maximum rating of the unit at the expense of a proportional loss of revenue.

One example is from Alstom where the operator can chose between 28,000 or 32,000 HGP-intervals. The price in terms of lost production is on the order of 2…3 percent power. This concept has recently been introduced on the GT24 and 26 by lower the temperature level after the second burner. Alstom have significant experience from the smaller GT13E2 units.

Another example is a ship with two 17 MW units where the chief engineer controls the rating between 120,000…4,000 hours (for 40,000 equivalent hours). As usual, the captain drives the ship but has to negotiate the maximum power setting with the chief engineer (speed scales on power cubed). Here the operator has the option to balance maintenance spending with the necessity of maintaining a certain speed. Most high-speed service ships have rather short voyage distances and therefore typically a high number of cycles per fired hours. The obvious cure would be to operate the engines at idle when moored – but the berth would experience severe water erosion from the jets. The trick here is to have large brakes on the power turbine shafts. The gas generator can still be in operation whilst the power turbine is stopped – avoiding a full low-cycle for the gas generator.

7.4.1 Condition monitoring

The cost of a condition monitoring system can prove to be a good investment even if only a single hot path failure can be avoided – prevention through prediction**. As a minimum, any competent system should be able to detect minute changes to the exhaust temperature (EGT) pattern**. Most hot-end failures have some kind of influence on the EGT-spread and should be treated with caution. The word "some" is used since both a too low or too high may indicate severe engine problems. Beside the pattern itself, the firing level itself is important. A too high firing value will consume life, whilst a too low will result in lost production revenues.

The firing level cannot be measured directly and the OEMs have typically used indirect methods. Direct blade temperature measurements should be ideal since the unit could be operated at a certain *metal* temperature. The word metal is highlighted because the normal method gives an average or bulk temperature of the gas rather than a gauge for the metal temperature. The latter is strongly influence by the function of the cooling system and hot spots from e.g. burners. The main issue with direct blade temperature measurement systems (BTMS) is ruggedness. This normally renders the BTMS impractical and one has to revert to the normal practice. The normal method is to base the indirect method based on a relationship between the firing- and exhaust temperature vs. expansion ratio. This relation should follow some kind of polytrophic process, where the polytrophic exponent (n) is a function e.g. the ambient condition[23]. The control equation is typically linearized through logarithmic differentiation, or similar. Other advanced control algorithms also take into account the effect from coolant temperature(s). One could either use the function to calculate the target exhaust temperature for a certain firing level, or vice versa. No method is, however, superior to the other with regard to this choice. A detailed description of a typical control algorithm is outside the scope of this report. A few words on how engine faults affect the firing level are included for completeness. Most issues related to turbine capacity (e.g. burnt vanes, central seal leakage, etc.) results in a lower cycle pressure. This naturally (thermodynamically) gives a higher EGT for a certain firing level. The net result is typically a reduction in firing level on top of the lost power due to the associated efficiency drop. The striking point here is that any condition monitoring or expert system has to incorporate the control algorithm. There are several publications and even research papers on various levels of condition monitoring where the firing level has been missed. Any competent system should calculate the actual firing level before any further analysis. Every measured parameter that goes into the firing controller will, to some extent, influence the firing level. For example - a faulty compressor discharge pressure (CDP) measurement will have a profound impact on the firing level. If the expert system has been trained without the control algorithm, the system will indicate a too low power level and a low CDP. The absolute key factor here is to be able to discover the low firing level – and that it is a natural fact due to the control function. An inexperienced operator may have interpreted the low power as a turbine failure (or similar).

Compressor blade vibrations can be monitored on-line with "tip-timing" systems. There are two principal problems, namely; forced response and flutter. Both are per se indeed complicated and only a brief discussion will be given here for completeness. The former is related to excitations from neighboring components and is independent of the displacement of the component itself. Flutter is more intricate and could be seen as forces acting on e.g. a compressor blade that are functions of displacement, velocity or acceleration and these forces feed energy into the system. The tip-timing measurement system is a non-intrusive inductive technology for measuring when the individual blades pass certain positions. This method can together with advanced algorithms be used for analysis of individual blade vibratory behavior. There are examples of commercial tip-timing systems on large heavy frames. General Electric introduced their rotor 1, 2 and 3 tip-timing system for 9FA and 9FB in 2010 and has more than 20 in commercial operation (per 2011).

[23] Air composition – but strictly speaking of the flue gas composition within the turbine section.

7.4.2 Inlet filtration

The inlet air filtration system and fuel quality is one of the absolute most important factors related to maintenance spending. Beside the drop in direct compressor performance, impurities within the working media may potentially cause both cold- and hot corrosion. The former is typically due to airborne pollutants that get scrubbed out from the air flow in the front stages in the compressor. The saturation line approach is driven by the depression due to the inlet acceleration. A state-of-the-art compressor may very well have acceleration from about 20 m/s in the inlet system to about a Mach number of 0.6…0.7 – Hence a significant drop in static properties! The air flow contains sufficient amount of condensation nuclei or cloud seeds. They typically are about 0.2 μm in size and certainly pass any reasonable inlet filtration system. The table below shows how rural impurities like SO_2 and HCL may introduce compressor corrosion by introducing an acid environment.

Acidity of Ambient Gases		
Sulfurous Acid – SO_2		
Ambient SO_2 in ppb (weight)	Dissolved SO_2 in ppm (weight)	pH
1	0.20	5.5
10	0.64	5.0
100	2.0	4.5
1,000	6.4	4.0
10,000	19.8	3.5
Hydrochloric Acid - HCL		
Ambient HCL in ppb (weight)	Dissolved HCL in ppm (weight)	pH
1	1,600	1.44
10	5,500	0.94
100	17,600	0.44

Based on Haskell, Gas Turbine Compressor Operating Environment and Material Evalution, GER-3601.

The filtration system is indeed important when it comes to the degradation rate of the unit. Compressor fouling is the combined effect of particles and a "glue" effect causing the dirt to stick to the blades. As a gas turbine operator one should have the highest possible level of filtration for trouble free operation. The table below shows typical filtration levels for two- and three-step filtration, respectively. The striking point is that the E11 (or HEPA) technology has indeed high filtration quality.

Typical filtration levels

Inlet		Two-step (F6+F8)		Three-step (F6+F9+E11)	
Size (µm)	Particle count per m^3	Filtration efficiency (%)	Particle count per m^3	Filtration efficiency (%)	Particle count per m^3
0.3-0.5	20,000,000	≈ 64	7,200,000	≈ 98,9	220,000
0.5-1.0	4,000,000	≈ 80	800,000	≈ 99.9	4,000
1.0-2.0	300,000	≈ 95	15,000	≈ 99.999	3

The figure below shows a compressor inlet after 28,000 operating hours with three-step filtration (F6+F9+E11). It is indeed hard to draw firm conclusions from a picture but it is not uncommon to find much worse after only a few months of operation. One can argue whether it is even possible to draw any conclusions because local conditions prevail.

Figure 20. Compressor IGV and rotor 1 after 28,000 hours of operation with HEPA-level (courtesy of VGB)

One illustrative example could be to compare the amount of air that goes into a GE 9FB in terms of an air-column over a normal soccer pitch. A standard pitch is 105×68 meters – equivalent of 7150 square meters. A massflow of 640 kg/s translates into 522 m^3/s or about 2,300 km air column (with the base of a football pitch and constant density) per annum. If one assumes 1 ppmw of foulant then the total mass is just below 20 ton.

Operation at extended periods at high ambient relative humidity may result in a phenomenon called “filter saturation”. Filter saturation is a critical problem since the filter can release large parts of the captured matter into the compressor in seconds. There is a potential immediate surge risk on top of corrosion issues. A surge is not an axisymmetric phenomenon and there is a significant risk of blade rubbing and compressor damage. The engine trust balance is severely affected with associated high bearing loadings.

7.4.3 Liquid fuels

Hot corrosion is an extremely rapid process when an alkali metal like sodium reacts with sulfur to form molten sulfates ($2Na+S+2O_2 \rightarrow Na_2SO_4$). The principal damage mechanism is that the molten salts deplete/destroy the protective Al_2O_3 and CrO_3 oxide layers from substrate diffusion. The "direct" corrosion is oxidation of the naked material once the protective oxide has been removed. The situation gets even worse if other metallic salts are present containing V, Pb, Ca, K, Li, Mg as either fuel- or air borne pollutants. The detailed chemistry kinetics is outside the scope of the report and the reader is referred to standard texts[24] on hot corrosion. This type is normally referred to Type I hot corrosion and features substrate depletion, inter-granular attack and sulfide particles. The type I of corrosion is typically kinetically restricted to temperatures above 900°C.

The combination of NaCl (and other alkali) and Na_2SO_4 is particular pernicious since it produces a molten salt mixture already at 600°C. I.e. if the metal temperature is less than 600°C, then condensation will occurs on the surface. This "lower temperature" corrosion is normally referred to as Type II corrosion.

Proper liquid fuel sampling and quality processes are crucial and prudent practice when operating on liquid fuels. There are several possibilities for contamination within the logistics – especially during sea transportation. Assume a small coastal tanker carrying **3,800 m^3** of fuel:

Sea-water contamination **in Liters**:	Resulting Na+K in ppm:
30	0.11
150	0.54
300	1.08

The specification of most OEMs typically falls within 0.1-0.5 ppm, i.e. little room for contamination. A caveat is in place here since all manufacturers have their own limitations.

One can show similar issues with lead contamination, but there is no real obvious source since we abandoned leaded fuels for automotive purposes. One can show that a small contamination of 10 liters of leaded car fuel would run a 40 m^3 liquid fuel batch out of specification.

The requirement of proper fuel quality sampling cannot be stretched too much!

[24] E.g. Roger Reed, Superalloys.

8 Gas Turbines and Carbon Emission

The role of gas turbines has changed from either a special application or stand-by mode to combined cycle plants in either mid-merit or base load. The reason for this is the availability of natural gas in combination with high efficiency potential. The high efficiency combined with natural gas high hydrogen content result in relatively low levels of specific carbon dioxide emission. Unfortunately, the relative lower carbon content in the flue gas makes the separation process more difficult, and may render in high separation tower heights to provide for sufficient residence time. Another issue is the flue gas flow which is on the order of 1.5 kg/MWs, compared to 0.95 kg/MWs for an advanced steam plant. The cross section of the separation tower should provide for a velocity around five meters per second. Hence, a higher and wider (bigger footprint) tower for a combined cycle plant capture plant compared to a coal fired. A normal state-of-the-art gas turbine based plant has about half the CO_2-emission per unit produced power compared to a hard coal fired plant.

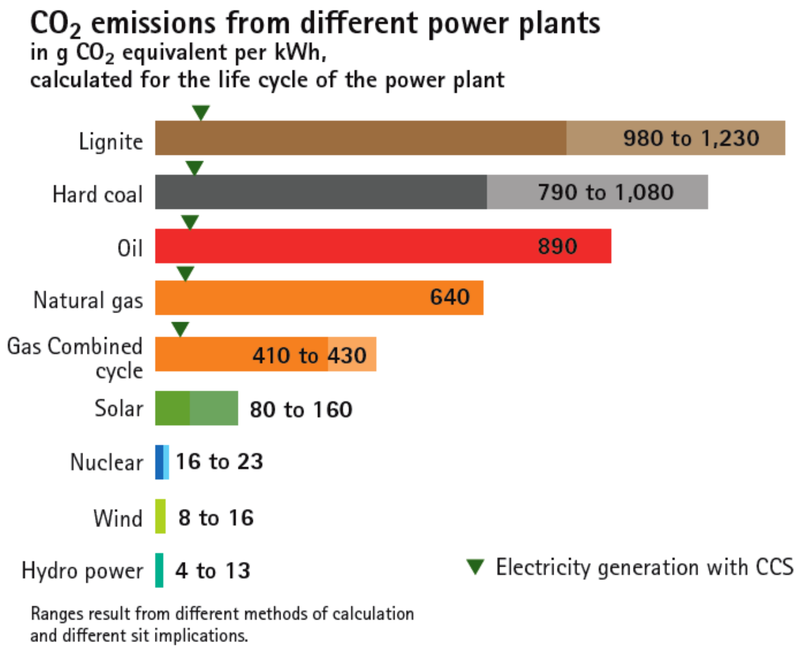


Figure 21. Carbon emission from different technologies (VGB[25]).

No commercial full-scale technology for CO_2-capture exists today and the road-maps towards feasible solution are still not clear. The text in this section will address post-combustion technology but there are other technologies being developed. Emerging technologies like Oxyfuel, IGCC/H_2 and chemical looping are currently being researched and developed.

[25] VGB, Facts and Figures 2010/2011 – Electricity Genration

It is probably safe to assume significant rise in first cost and the size of each plant. We would probably not see plants like Edisons plant on Pearl Street in the future – driven by scale of size.

Future commercial size storage sites also have to battle the strong NIMB-concept (not in my backyard) and off-shore storage sites would probably be more acceptable and safe.

8.1 Available Technologies

Both pre- and post-combustion technologies are available at a significant drop in performance. No mature technology exists and the development path is at its beginning. In the near-term perspective, most manufactures probably want to stay within their current portfolios using amine technology and flue-gas recirculation. The latter is required for having sufficient partial pressure of CO_2 in the flue gas to the capture plant and reduction of the flow.

8.2 Capture readiness

The exact definition of capture readiness is hard to condense into a short description. It has already been established earlier in this section that one has the need to reduce the flow and increase the partial pressure of CO_2 in the flue gases. The first step is most likely an amine-based plant with flue gas re-circulation. The maximum recirculation (FGR) is on the order of 40 percent weight for keeping approximately 16 percent (volume) oxygen. The 16 percent limit[26] is to maintain high combustion efficiency in a DLE-system. The turbomachinery part of a gas turbine should not pose problems due to changed working media. The change in working media, for a 40 percent FGR, is not significant in terms of speed of sound and viscosity, etc. In a wider perspective, capture readiness should also mean available land space for the capture plant. The plumbing associated with getting some 400…800 kg/s flue gas to the separation plant is significant.

The probably biggest challenge in terms of turbomachinery is the steam turbine last stage design. Some 50 percent of the flow has to be extracted from the cross-over pipe for the re-boiler duty. The varying flow will change the last-stage loading and may turn the stage into turn-up mode. Loosely stated, the pressure at a certain point within the turbine is proportional to the mass flow passing downstream. Hence, one can easily show that the pressure ratio over a normal stage is constant since the pressure before and after is proportional to the flow. The condenser behaves in a different manner (driven by the LMTD and HTC-value) and the last stage takes a large hit in loading. The "turn-up mode" is direct result of light loading and an associated imbalance in the radial force field. The net result is that the stream-lines are packed towards the outer annulus, leaving a strong recirculation zone near the hub region. This recirculation zone generates large amounts of heat since the rotor feeds energy into the "trapped" steam. This phenomenon is well-

[26] This limit has been published by General Electric. There is no information about extrapolation into other technologies. The underlying combustion kinetics is indeed complicated and outside the scope of this report.

known to anyone ever attempting to start a steam turbine – the exhaust temperature drops when loading the turbine. The cure is to choose a turbine exhaust size that gives "non-optimum" exhaust loss – slightly to the right of the minimum in the loss bucket. It is also quite common to have condensate spray nozzles at the exhaust section to cool the high temperature steam. Combined cycles typically have one LP-cylinder, either single- or double flow for suitable exit losses. This leaves little freedom than removing the last stage when the unit is transferred into capture mode. Hence, the exhaust will be very small when operated in "normal" mode with associated losses. The losses may be very high, especially during cold days when the last rotor is approaching the limit loading[27]. A true "capture ready" plant is therefore not feasible and the plant has to be modified when changed for capture operation. The situation is quite different for a normal Rankine plant where one typically have two- or three LP-cylinders. One could be installed at the normal NDE (non-driving end) of the alternator and be removed from operation when the plant is operated in capture mode. True flexibility could be achieved from using a SSS-coupling but this is limited by the available sizes (currently below 300 MW@3000 rpm).

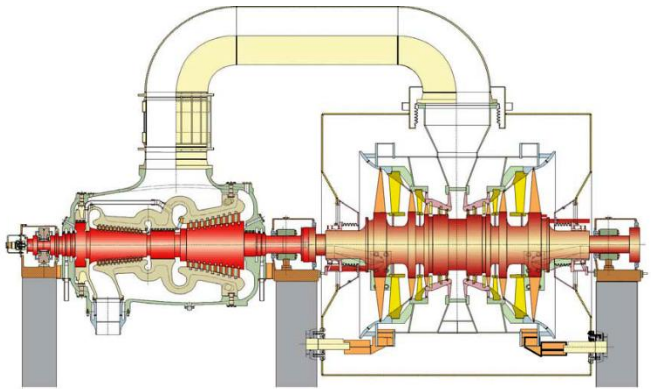


Figure 22. Typical combined cycle steam turbine (courtesy to Siemens)

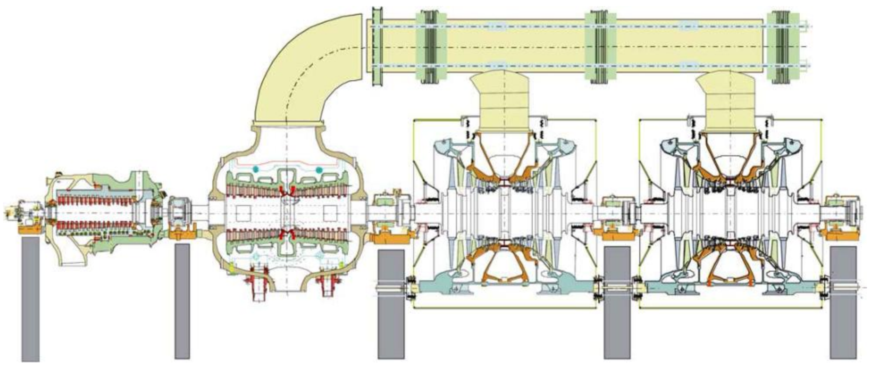


Figure 23. Typical coal fired power plant steam turbine.

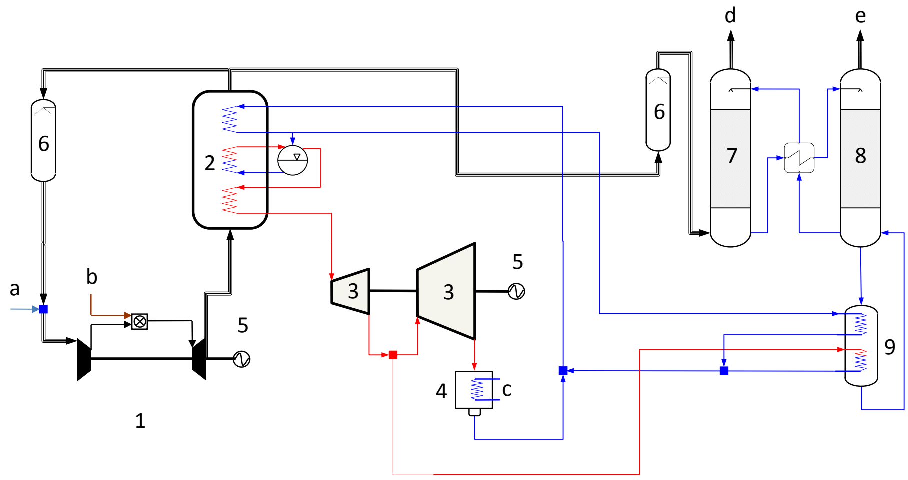
[27] Limit loading is the point where the meridional Mach number approaches unity and no further gain in lift is obtained by lowering the back pressure.

8.3 Impact on performance

The impact on performance is significant since large quantities of the steam have to be extracted and the condensate has to be returned to the cycle. A typical absorbent is MEA or chilled ammonia (CAP) that is stripped at 120°C (depending on the lean amine loading) and approximately 160°C, respectably. The penalty in exergy loss is obvious when steam is extracted from the cross-over pipe for creating a low-grade steam phase mixture at the mentioned temperature levels.

The drop in performance is typically on the order of 8 percentage points for a GE 9531FB.03 unit with a three-pressure level HRSG.

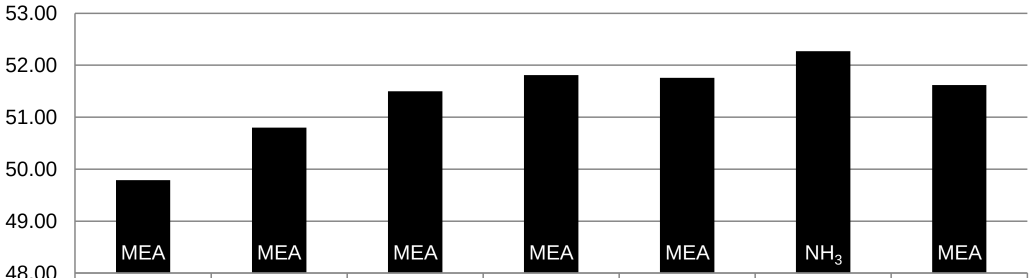
The department of Energy Sciences has developed a patent pending concept that offers superior performance for post-combustion CCS. The concept is based on utilization of pressurized water from the economizer. A knock-on effect is that the benefit from having multiple pressure levels is minimized. The reason for having multiple pressure levels in a combined cycle could be explained through the void area in a T,s-diagram. This is conceptually not a straightforward concept to grasp – but could be seen as a possibility to fit a Carnot cycle between the lines. Hence, a possibility to extract additional work in the combined cycle. The concept developed at Lund University carefully minimizes the void area by controlling the slope(s) of the water lines in a T,Q-diagram. A detailed discussion is outside the scope of this report and the reader is referred to the doctoral thesis by Jonshagen[28]. The concept also lowers the exhaust temperature from the HRSG. This results in a lower cooling duty for the flue gases before the absorption column.



1: Gas Turbine
2: Heat recovery steam generator
3: Steam turbine
4: Condenser
5: Alternator
6: Flue gas condenser/cooler
7: Absorption column
8: Stripper column
9: Re-boiler

Figure 24. Economizer-reboiler coupling for optimum performance.

[28] Jonshagen, Modern Thermal Power Plants – Aspects on Modeling and Evaluation.



	Single-pressure conventional reboiler	Single-Pressure	Single-Pressure Reheat	Dual-Pressure Reheat	Tripple-Pressure Reheat	Chilled Ammonia Single-Pressure with Reheat	Tripple-Pressure Reheat conventional reboiler
%	49.79	50.80	51.50	51.81	51.76	52.27	51.62

Figure 25. Performance impact from different CCS concepts (datum 59.6 percent excluding step-up equipment and parasitic consumption)

8.3.1 Extension to coal-fired plants

The concept should be possible to introduce on a normal coal fired boiler. There are limitations associated with extracting heat from the back draft due to H_2SO_3 condensation in the air-preheaters. The normal remedy is steam extraction for a coil on the air-side and this could be extended to include the mentioned mode of operation. Another possibility is to optimize HARP-extraction (heater above reheat point) and the de-superheater for CCS operation. The nature of a steam plants pre-heater chain also offer integration possibilities and flexibility beyond the possibilities of a combined cycle

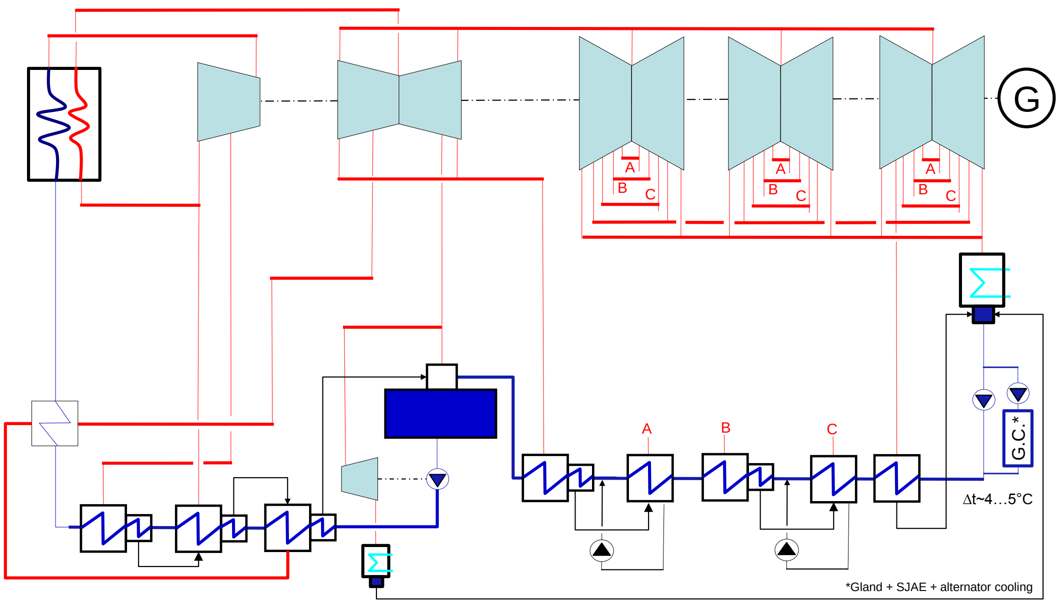


Figure 26. Advanced steam plant

Each preheater need about 3…5 percent[29] extraction flow and the exhaust loading can be increased by simply closing the first low-pressure heaters.

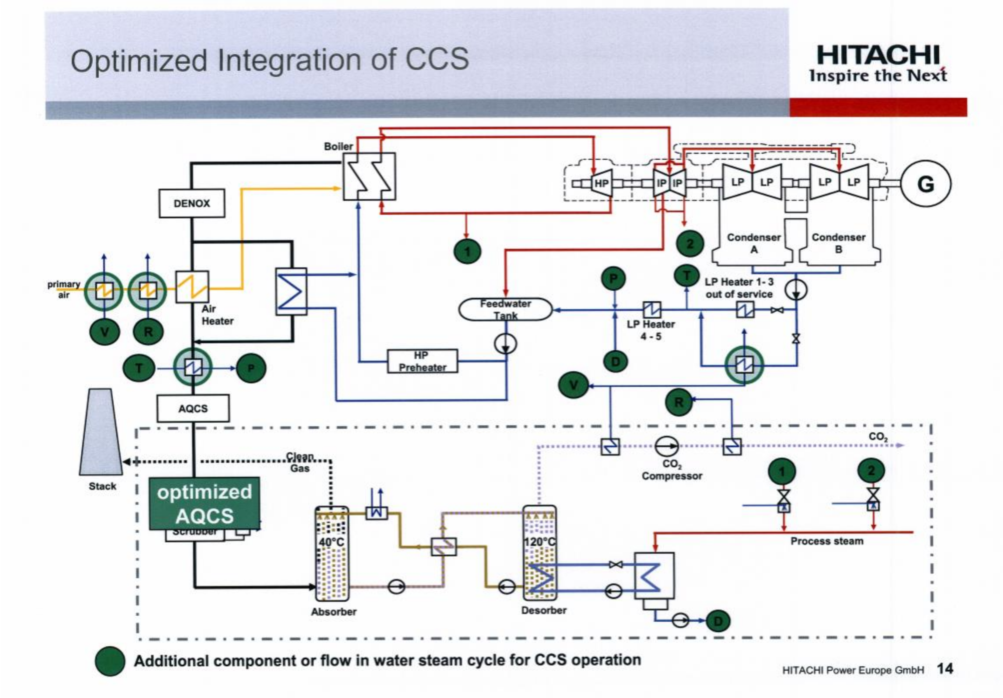


Figure 27. CCS integration in an advanced Rankine cycle (Courtesy of Hitachi)

The difference between the combined cycle and a Rankine cycle becomes obvious when comparing the integration strategies. The combined cycle is sensitive for the hot re-boiler condensate (approximately 120°C). The situation is quite different for the Rankine cycle where there are natural induction points and also possibilities of re-introducing low-grade heats into the cycle (see the figure above).

8.4 Road map for a feasible solution/technology

Beside the impact on efficiency from stripping the amine, the amine itself has a tendency to degrade. One (of many) degradation products is nitrosamine – one of the most potent carcinogens known to man. A capture plant will have some slippage of the absorbent and associated release into the atmosphere.

The Norwegian Mongstad project was put on hold in early 2010 due to the mentioned issue.

The amine-based technology is considered to be the most mature technology but there are other competing gas turbine based technologies. One promising candidate is the Semi-Closed Oxyfuel-Cycle (SCOC). The cycle uses mainly CO_2 as working media and fires natural gas together with pure oxygen in the

[29] Of the admission flow

combustor. The plant also incorporates a dual-pressure HRSG and steam cycle. The air separation unit (ASU) introduces a large hit on the net plant efficiency and the attainable levels are in the high 40 percent.

An in-depth description of the SCOC process is outside the scope of this report.

9 Fuel Flexibility

The future carbon free driver for small and medium size gas turbines will probably be bio-fuels and fossil mid-merit production. One can assume that future clean-up legislation will hit coal fired plants first and perhaps later combined cycle plants involved in base production. Mid-merit or cyclic operation does not provide a feasible platform for CCS due to inherent limitations in either downstream processing or the pressurization train.

Low-calorific fuel in gas turbines is certainly not a new application. There are quite a number of plants firing coal based Syngas. There is was even a demonstrator in Sweden in the early 90s firing gas from gasified wood chips. The plant was based on a Ruston Typhoon (now Siemens SGT100) and used a post compressor bleed for maintaining compressor stability. Another Swedish example was the Volvo 600 kW unit in Helsingborg firing landfill gas.

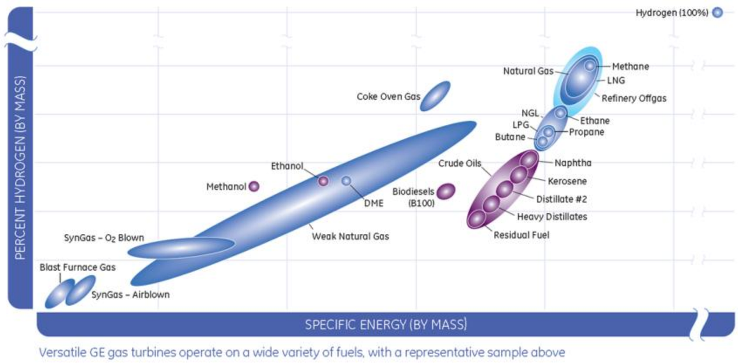


Figure 28. General Electric fuel range (Courtesy of GE)

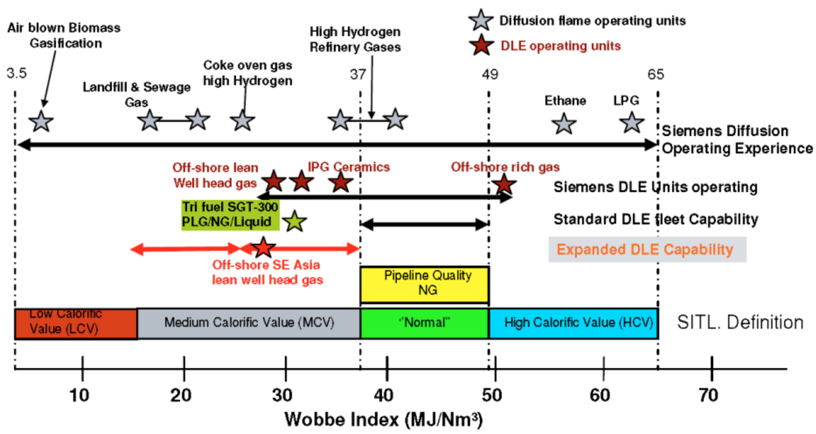


Figure 29. Siemens (small/<15MW range) fuel flexibility range (Courtesy of Siemens)

This section will not cover all aspects of low-LHV firing and the intention is to give some insights and background information. Most potential problems will hopefully be addressed by the OEM before a commercial product is available.

A severe limitation seems unavoidable in the combination of an air-blown gasifier and a twin-shaft engine. Hence, an oxygen-blown gasifier seems better suited since CO_2-ballast off-loads the compressor turbine.

It is very hard to establish any rule of thumb describing suitability for a certain fuel in a certain engine type. One conclusion, however, can be drawn from the fact that the matching issue is shared with massive steam/water injection (like Cheng™-cycles). Typical gas turbines for Cheng cycles have been Allison 501 (Rolls-Royce) and General Electric LM2500 STIG. Most engines should, from a strict turbomachinery perspective, be able to handle gases down to some 20 MJ/kg. There will be a substantial power and torque increase from the added ballast flow.

The impact on *gas turbine* O&M costs is strongly dependent on the type of fuel and its properties (corrosion and erosion). The word gas turbine is highlighted because the fuel preparation-, processing- and compression plant may turn into heavy maintenance burdens and lower the RAM-figures.

9.1 Fuel characteristics – An introduction

There is a plethora of different ways of describing fuel issues/characteristics and most issues can be boiled down to: Wobbe-index (WI), dew point, blowout, flashback, laminar and turbulent flame speed, auto ignition delay time, flammability, flame temperature, radiation.

The Wobbe-index is used for characterization of a certain fuels ability of being admitted to the combustor zone:

$$\text{Heat flux} = \text{const} \cdot A_{eff} \cdot \underbrace{\underbrace{\frac{HHV}{\sqrt{SG}}}_{WI} \cdot \sqrt{\frac{T_{std}}{T_1}}}_{WI_{Temp}} \cdot \sqrt{p_1 \cdot \Delta p}$$

Loosely stated, for a certain geometry WI_{Temp} sets the amount of energy that can pass a certain restriction.

Hence, a fuel is interchangeable if they share the same WI. The Wobbe-index does not say anything of the suitability for a certain fuel in a gas turbine – merely the theoretical energy flux. A more useful approach is to turn it around and use it as a base for designing the fuel system. It is quite common for the OEMs to specify their fuel range as a Wobbe-index and an array of limitations for various constituents like solids, water, liquefied heavy gases, oils from compression, hydrogen sulfide (H_2S), carbon dioxide, hydrogen, carbon monoxide and siloxanes. The figure below is an example from Solar turbines, showing fuel injector pressure drop vs. LHV and WI. The red line is the maximum allowable showing that their design is capable for rather low heating values.

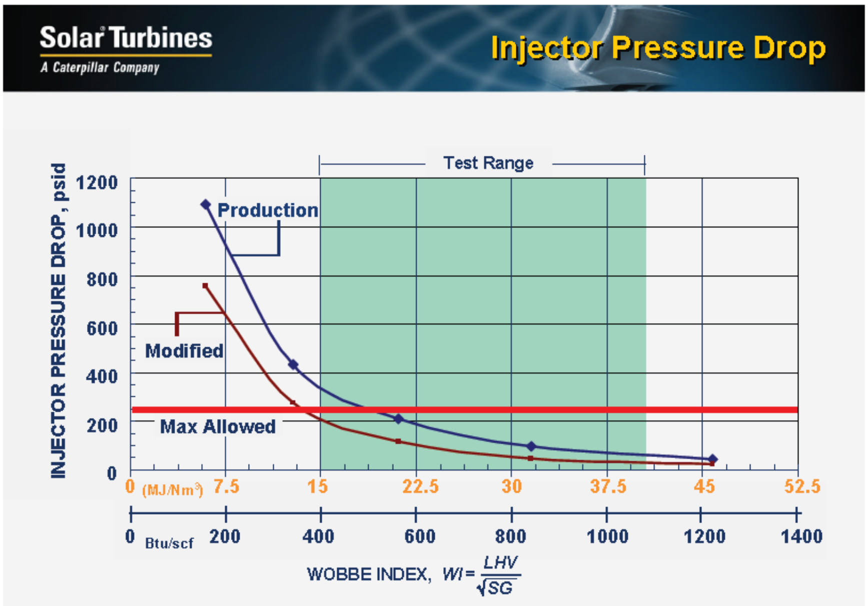


Figure 30. Fuel injector pressure drop vs. Wobbe-Index

9.2 Impact on performance

A typical 48 MJ/kg natural gas fired unit typically fires about 0.02 kg of fuel per kg air (i.e. a fuel-to-air-ratio of 0.02). The turbine expands the gas and creating work according to the sum of each stages product of flow times heat drop. When the heating value is lowered, the mass flow passing the turbine is increased and the composition is changed. The net impact is increased work but not to the full potential since one might end up with a non-optimal turbine (see previous section). The plot below was published by General Electric showing the impact on performance for different fuels and mixing ratios.

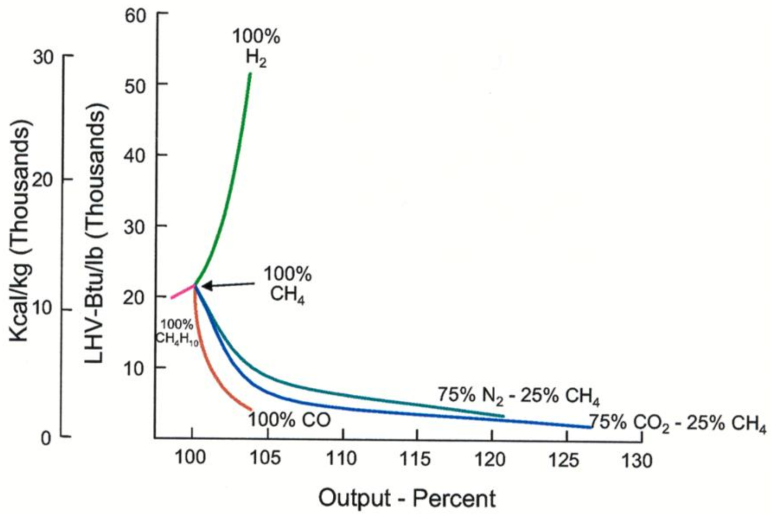


Figure 31. GE single shaft performance vs. fuel quality (Courtesy of GE)

Each line on the graph has a curve-linear relative function showing the mixing ratio. For example, half way the line from "100% CH_4" to "75% N_2-25% CH_4" represents 50 percent mixing of the mentioned qualities. The previous reasoning holds also for hydrogen – where one gets a lower turbine flow due to the high heat content but high expansion quality. The net result is an increased turbine work.

9.3 Engine matching and aero-elastic issues

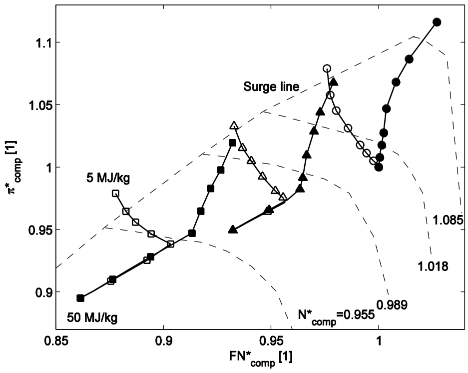
The impact from having a large amount of inert ballast through the turbine has to be analyzed with the specific performance deck[30]. A normal gas turbine has more or less the same flow at the compressor inlet as at the turbine outlet since the fuel flow balances over-board leakage. The engine pressure ratio is mainly driven by the compressor pumping capacity and the turbine capacity and the firing temperature. If a unit is fired with a low-LHV fuel then the fuel mass flow is increased to maintain the energy flux. The fuel-to-air ratio does not scale directly with the heating value since the inerts have to be heated to the firing temperature. One can easily show that a single-shaft or the gas generator of a twin-shaft will respond to a different fuel according to:

$$\underbrace{\frac{\Delta PR}{PR}}_{(A)} = \underbrace{\frac{\Delta m}{m}}_{(B)} + \frac{1}{2}\underbrace{\frac{\Delta COT}{COT}}_{(C)} - \underbrace{\frac{\Delta FN}{FN}}_{(D)} + \frac{1}{2}\underbrace{\frac{\Delta R}{R}}_{(E)} - \frac{1}{2}\underbrace{\frac{\Delta \kappa}{\kappa}}_{(F)}$$

The preceding equation gives a good hint of how an OEM will approach the matching problem. The term to the left (A) is the relative pressure ratio and will be influenced to a first order from: increased flow through the turbine due to the higher fuel flow (B) and changes in gas properties through terms (E) and (F). If one assumes that the changes in properties is one order of magnitude less than the change in mass flow – then e.g. a 10 percent change in mass flow will increase the engine pressure ratio by the same amount. This will have an impact on the rear part of the compressor since we will increase the aerodynamic loading and eventually force the compressor into surge. Exactly how this will affect all units is not possible to generalize since it is strongly dependent on the individual design. Most units should have a built-in margin (of some 10 percent) from normal operation to stalled operation (and eventual surge) that should be possible to reduce to a minimum level. There are several other issues like low-speed operation, acceleration, fouled compressor, etc will certainly add to the required margin for trouble free operation. There is no quick fix in terms of compressor technology since we have to add an ultimate and perhaps even a penultimate stage to the compressor. Instead, the probably most convenient way is to alter the engine matching by either changing the turbine size (D) or reduce the firing level (C). The latter will reduce the engine power density, efficiency and introduce combustion issues. The turbine size can be altered by either re-staggering of the first vane, trailing edge cutting, re-designing within current flow path or a complete new design. Re-staggering is simply a change to the profile setting angle applied to the first vane. The drawback is a poorly matched turbine

[30] Off-design matching program

downstream with an associated efficiency penalty. Trailing edge cutting behaves in a similar fashion and the small differences will not be discussed here. Re-designing the whole flow path is a cost-effective way to adopt the unit and maintain the same stress level for disks, attachments and blades. This option is still not optimum since the velocity level is elevated and for example, will increase blading heat transfer and turbine exit loss. Instead, the preferred option is to re-design by keeping the hub contour and accept an increased stress level. Whether this is possible or not depends on many factors and is outside the scope of this report. Post compressor bleed will also lower the mass (B) passing through the turbine. The performance penalty is severe since this bleed has to be after the compressor (i.e. at full charge). This might be justifiable if there is a need for high-pressure air in the fuel processing plant. An intermediate stage bleed would render in rear stage stall (and eventually surge) and is therefore not possible. An old misperception is to throttle the compressor inlet – this will only lower the mass flow but not provide any cure for the rear compressor stages.



A twin-shaft unit seems to be a better platform for low-calorific firing since the power turbine only has an aerodynamic coupling to the gas generator. Most reasoning above holds for the compressor turbine but the free power turbine provides effective means of having sufficient exhaust area. The probably most important feature is the fuel inert ballast. An air-blown gasifier produces a fuel with high N_2 content whereas an indirect has CO_2. An unfortunate effect of the ballast is shown in the compressor map where filled symbols are N_2 and the circles are CO_2. One can see a totally different behavior depending on the fuel quality. The lowest points are 100 percent regular natural gas and each step upwards represents different level of mixing. The three sets of curves represent different firing levels. The lowest points can be thought as the normal running line. The filled dots (N_2) at the highest and nominal firing level moves towards one a clock into where the flutter area is. Flutter has to be avoided to all cost in a compressor because it will soon have consumed the available lifing. Any serious gas turbine OEM should be aware of this phenomenon and take proper measures. This is further amplified at low ambient temperatures.

Lund University has published several papers related to the subject.

9.4 Engine handling

It is probably safe to assume that most OEMs will approach the market with re-matched engines for maintaining the surge margin. The re-matched engine has a relative large turbine section and modified burner (and subsystems). A twin-shaft engine has a running line in contrast to a single-shaft unit. This means that the engine has to pass between the surge line and hang boundary during the start-up transient. Engine hang is a transient mode where the fuelling ramp is insufficient for accelerating the gas generator. The underlying reason is, loosely stated, the imbalance in absorbed compressor work and excess turbine work preventing the gas generator from accelerating. The immediate consequence is high exhaust temperature and the start has to be aborted for preventing from engine damage. The safe passage between the surge- and hang region is ensured by carefully controlling the acceleration rate (versus referred speed). Single-shaft units use another philosophy and use massive start bleeds and are therefore not prone to hang.

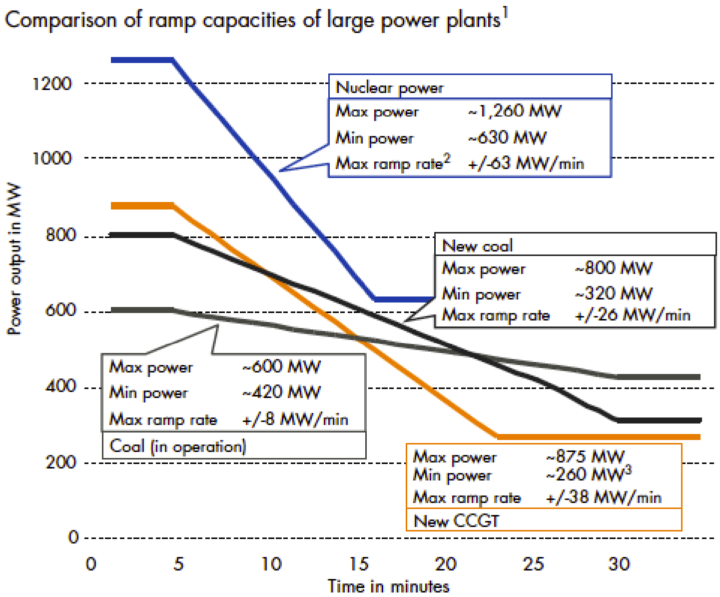
Any competent OEM should be able to set up a proper turbine governor[31] for low-calorific handling. It is not uncommon for OEMs to have in situ chromatographs for on-line fuel analysis when operated with normal natural gas feed, where minute changes are to be expected. The measured gas data are used to in the control system for adjusting ramps, energy flux, etc. There is no information available of handling changes in heating value by a factor of two. This level of change can be expected if the engine has to be started with natural gas for e.g. cold start of the fuel plant.

There are several modes in gas turbine operation where direct control is impractical. One example could be a load rejection where the engine is supposed to decelerate to idle. A flame-out is prevented from by setting the fuel to an appropriate level rather than controlling the sequence. A too high setting will due to rotor inertia surge the compressor, whilst too low will render in flame-out.

[31] The turbine governor typically includes several controllers for e.g. firing level, maximum aerodynamic speed (low ambient), maximum physical speed (high ambient), DLE system, run-up/acceleration, run-down/deceleration, self-sustaining, load rejection, load ramp, IGV/VSV scheduling… The sometimes conflicting control requirements are controlled through a minimum selector within the turbine governor.

10 Operational Flexibility

Operational flexibility will be of paramount importance once the amount of volatile production gains capacity. A typical wind farm has an approximate capacity factor on the order of 40 percent. This means that the average annual production in MWh's only reaches less than half of the installed power multiplied by 8760 hours per annum. To further illustrate this, an old rule of thumb states that wind power needs a back-up capacity of 80 percent. This opens up a huge non-spinning global market for rapid started diesel engines and gas turbines. A gas turbine may be flexible in terms of starting and ramping, ranging from 40 seconds for a small stand-by unit up to some 10 minutes for an aero-derivative or light industrial. A heavy frame and combined cycle does not provide this level of flexibility for cyclic operation. Typical heavy frame start-up times are not far from 10-15 minutes and the typical entire plant can be running at base load within 30 minutes. The starting time has been addressed by all OEMs (Siemens FlexPlant™, General Electric FlexEfficiency™ 50, Alstom GT24/26, and Mitsubishi M501GAC/F5) with higher dispatch flexibility at the hand. The FlexEfficiency™ 50 has the same order of magnitude ramp-rate as a BWR nuclear plant. The figure below is from VGB and shows a comparison between typical production types in the German theatre. Coal-based plants offer least flexibility 8…26 MW/min – on top of very long start-up times. The combined cycle below has a ramp rate of 38 MW/min and it is based on a "2+1" configuration. This means that there are two gas turbines and a single steam turbine.



1 Examples of max. and min. capacities and ramp rates can deviate.
2 For a power ramp that lies below 20% of the maximum unit power, a ramp rate of up to126 MW/min can be attained.
3 One turbine is shut down

Figure 32. Ramp-rates for different production types (VGB)

The best performing "simple cycle" engine (GE LMS100) offers 100 MW within 10 minutes at an efficiency of 46 percent. The high efficiency is a result of a very high pressure ratio and inter-cooling. Pratt & Whitney has introduced their new FT4000 Swiftpack plant. The FT4000 is rated at 120 MW with a net efficiency exceeding 41 percent. The start-up time is 10 minutes without maintenance penalty. The P&W plant is based on two gas turbines in simple cycle with no inter-cooling. The dual-engine configuration offers high part-load plant efficiency.

10.1 Strategies for providing balance power for wind and other volatile sources

The Swedish power production structure provides excellent capabilities for rapid ramping capability through hydro power plants. This offers a significant advantage over other sources like "storing" through pump stations or hydrogen production and storing. The strategy in some countries is to use nuclear power for providing balance power. The figure below is taken from VGB Power tech (Facts and Figures 2011/2012) and shows the full German theatre during a week in 2008. The maximum load is about 60 GW and the full installed wind capacity is 25 GW. The actual wind production is, however, much lower and is averaging around less than half of the installed capacity.

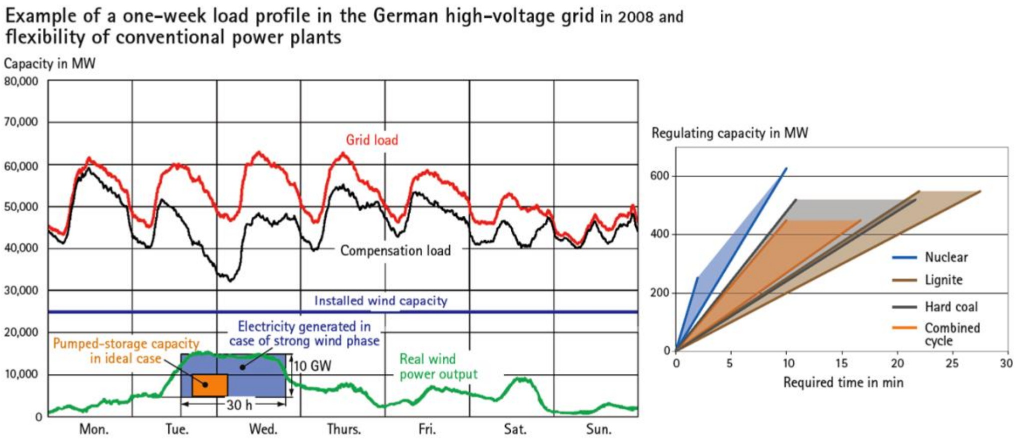


Figure 33. The German grid and production capacities (VGB).

The next figure shows an example from Spain, where high levels of renewables also are present in the system. The total production capacity is on the order of 100 MW, i.e. about three times larger than Sweden.

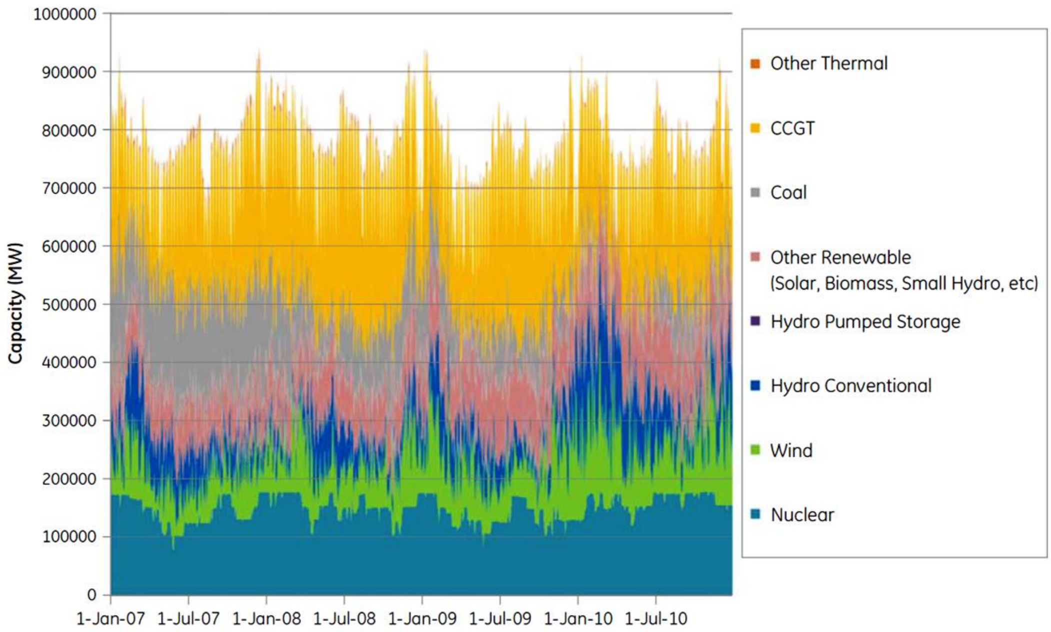


Figure 34. Spanish power production mix from 2007 to 2010 (courtesy of General Electric)

There is an example from Spain where the wind production dropped at a rate of 1.3 GW/h in December 2010. An illustrative example of wind production could be an imaged drop in wind from nine to seven meters per second. The resulting capacity would be $P_7/P_9 \approx (7/9)^3 \approx 0.47$ – a drop by 2 m/s (from 9 m/s) in wind reduced the production by a factor of half.

Figure 35. Example from Spain 2010 (Courtesy of Klimstra et al., Smart Power Generation, Wärtsila)

A typical wind power plant starts producing electricity at about 4 m/s and follows theoretically a cubic fashion up to about 12 m/s where the rated output is reached. The local wind variation, it self, typically follows a Weibull distribution. The most rapid off-loading – storm protection – occurs on a certain wind speed for protecting the gear box in the wind tower. The typical cut-off wind speed is 25 m/s.

10.2 Emerging technologies

There are advanced cycles offering flexibility in terms of load and efficiency. One prominent candidate is the wet cycle, which is offering turn-down in terms of shutting down the water cycle.

10.3 Lifing and cost of flexibility

The flexibility of a plant is strongly governed by the start- and ramp time and load capability. Load capability also includes turn-down where acceptable emission levels can be maintained. A "typical" peaker dispatch profile is Monday through Friday from 06 to 20. Hence, there are 10 hours per meridiem where there is questionable economy in keeping the engine running. One could consider running the plant at reduced output or shutting down overnight. There is no simple answer to the question whether to run at some minimum load or shutting down since it depends of a diversity of factors. Typical issues are emission turn-down and steam turbine turn-up. The turn-down in emission level is due to combustor stability issues and combustion efficiency at low firing levels. There are two ways of addressing this, namely, diffusion type pilot burner and reduced airflow[32]. Steam turbine turn-up is due to low volumetric flows in the turbine exhaust(s). It is possible to show that there will be a re-circulation flow near the hub of the last row. This re-circulation flow causes compression work to the steam and there are in many cases real needs for cooling. The cooling is supplied by spraying condensate into the turbine exhaust with dedicated nozzles. One "proof" of this re-circulation flow is traces of light blade erosion where the droplets hit the blades.

A rapid (or even emergency) start is penalized by applying severity factors when adding cycles. A typical frame rapid start is 2…5 times more severe than a normal ditto. An aero-derivative typically goes on-line within 10 minutes without any life penalty, or is limited to one low-cycle count.

10.4 Synchronous condenser operation for grid support

In synchronous mode the alternator is operated as an over-excited (leading power factor) synchronous motor. It is not possible to have a single-shaft unit operating in this mode without the possibility to disconnect the alternator with a SSS-coupling. A twin-shaft unit is better suited for this duty and some engines may have the possibility to operate in synchronous compensator mode with a spinning power turbine. The governing factor is the amount of windage heat created by the turbine. For example, both the General Electric LMS100 and the Pratt & Whitney FT8 are claimed to have synchronous compensator capability. The common factor between the engines is the non-

[32] Single-shaft units can use the variable geometry (IGV/VSVs) for direct air flow control. The situation is different for multi-spools where variable combustor geometry (by-passing air) has to be utilized.

geared 3000/3600 min^{-1} power turbine. The ventilation work that is the prime source for feeding work into the air scales with the speed cubed. A typical geared power turbine has twice the speed compared to a non-geared ditto – hence eight times (!) more work input. This operational mode requires a parallel frequency convertor for starting. The breaker is connected (synchronized) when passing synchronous speed during coasting down from a slight over-speed.

10.5 Aero-derivatives

An aero-derivative should have the possibility of being started and fully loaded within 10 minutes. This class of engine typically has efficiencies around 40 percent. Best in-class figure is the General Electric LMS100 at 46 percent followed by: Pratt & Whitney FT4000 (41 pecent) Rolls-Royce Trent (42 percent) and General Electric LM6000 (42 percent). The LMS100 is a three-shaft "conventional" engine with inter-cooling. The design is a direct response on the US market need for a flexible mid-size unit with 100 MW capabilities within 10 minutes. The engine can be used for 50 Hz operation by replacing the first power turbine guide vane. The change of the first power turbine guide vane is most likely required for maintaining the low-pressure compressor pumping capacity. This is probably driven by a reduction in power turbine capacity when operated at 82 percent speed at 50 Hz. The amount of heat available from the inter-cooler is on the order of 20…25 MW. This, in combination, with usage of the exhaust heat makes the engine attractive for integration in the pre-heater chain in a normal power plant.

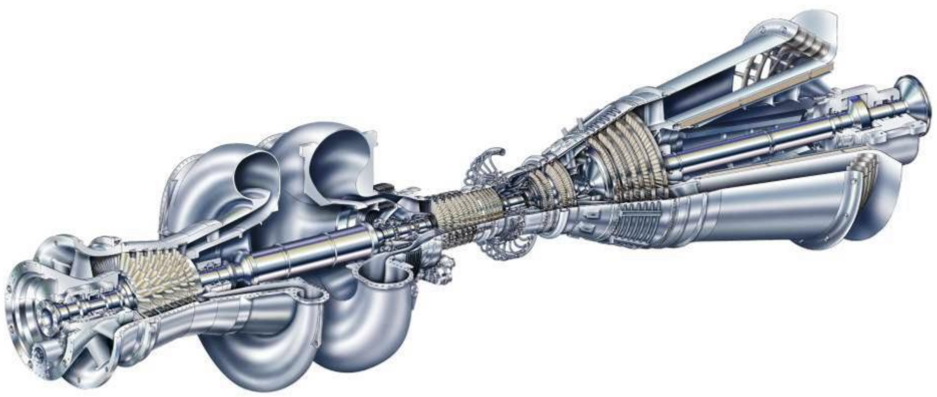


Figure 36. General Electric LMS100 (Courtesy of General Electric)

Fuel burn during start-up will increase the effective heat rate if the number of fired hours is low.

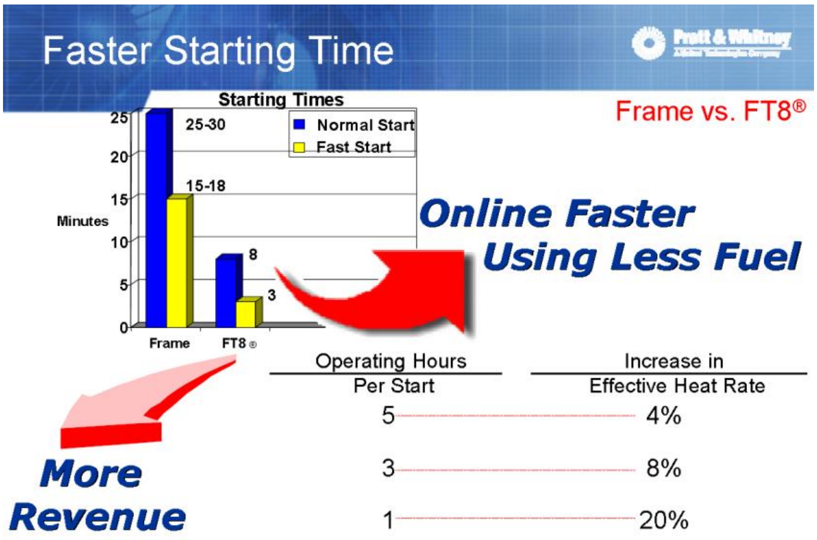


Figure 38. Start time influence on engine performance (Courtesy of Pratt & Whitney).

10.6 Heavy Frames and Combined cycles

Combined cycle offers the highest overall performance potential at the expense of flexibility.

The steam-cooled gas turbine has a longer start-up time and is therefore less flexible in terms of daily cycling. The figure below shows the start-up time for an air-cooled vs. a steam-cooled plant. There are two OEMs today offering steam-cooled units, namely General Electric and Mitsubishi. Recent designs from Siemens and Mitsubishi offer the same (or better) performance as the General Electric H-class plant. The additional complexity involved with steam cooling may also render in less reliable technology. There are three typical issues: engine start-up, getting the steam onboard rotating blades (and back to the HRSG) and ballooning[33]. In a normal air-cooled case, one still needs take into account the blade external loading with low suction side pressure. A cooled vane needs structural elements for preventing from change of the shape due to creep. A steam cooled design has very high internal pressures further amplifying this problem.

[33] Ballooning is due to creep at elevated temperatures. The imposed stress is due to the pressure difference between the steam and the lower blade profile external pressure.

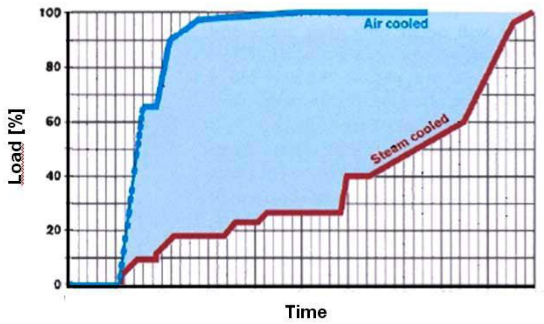


Figure 39. Plant start-up times (air- vs. steam-cooled)

The steam cycle admission data has a significant impact on the start-up time. The current trend is to design for 620°C whereas most previous plants were designed for 565°C. The latter figure is valid for utility types of turbines and the figure is even lower for smaller industrial units. The step from 565 °C to 620 °C prolongs the turbine start-up time by a factor of three (for the same stress level).

Duct burners could provide extra flexibility since it is possible to use the oxygen surplus in the flue gas directly. This normally requires a different type of HRSG design since the radiation from the burners is quite high. The cure is to have a part of the evaporator before the final superheater (or/and re-heater).

10.6.1 Siemens

60 Hz market
Siemens has launched a plant concept range called FlexPlant™, where flexibility is traded against some efficiency. The plant is capable of cyclic operation and is designed for daily start- and stop firing 4,000 to 8,000 hours per year. The plant is based on the 60 Hz F-class SGT6-5000F unit gas turbine and once-through technology heat recovery unit. The SGT6-5000F is capable of reaching 150 MW within 10 minutes and full load in 12 minutes. The rapid start capability comes at no extra maintenance, which is exceptional for a plant of this size.

The **Flex-Plant™ 10** is the smallest and has a single pressure heat recovery unit (w/o reheat). The drum has a small diameter and the pressure is chosen for optimum hoop stress[34] and lifing. The plant has a rating of 275 MW and an efficiency of 48.9 percent at an ambient temperature of approximately 30 °C. Full load is reached within one hour. The steam turbine (SST-800) has an air-cooled condenser. *The Flex-Plant™ 10 is the only combined cycle that qualifies for non-spinning reserve.*

[34] The hoop stress is a component of the stress tensor in cylindrical coordinates, which is normal to both the axis and the radius. A more practical explanation would be to consider a ring, which is pulled radially from the outside diameter with an even load (cf. pressure difference). The resulting hoop stress is developed inside the ring to maintain static equilibrium.

The higher performing **Flex-Plant™ 30** is based on the same gas turbine but has a more advanced bottoming cycle. The plant is rated at 305 MW and 57(+) percent efficiency at ISO-conditions. The heat recovery unit has three pressure levels with reheat. The HP-part is once-through Benson technology for avoiding a thick wall HP-drum[35], the steam turbine is a Finspong SST-900, enabling fast start-up capability. Cooling towers are used as a sink for the turbine condenser.

The **2*1 Flex-Plant™ 30** uses two SGT6-5000F and has a rating of 618 MW at 57.3 percent efficiency. Each gas turbine has its own HRSG and the steam turbine is replaced by a utility type unit (SST6-5000).

Flex-Plant™ 30 offers the fastest combined cycle start-up capability available today. The Siemens range is currently based on 60 Hz machines but there is no real stopper for introducing the solution on the 50Hz market.

50 Hz market
The 50 Hz market is approached with the SGT5-8000H gas turbine. The plant can be started within 30 minutes from hot condition (i.e. after 6-8 hours overnight shutdown). The maximum ramp rate is 35 MW/min.

10.6.2 Alstom

The Alstom KA26-2 ICS (integrated cycle solution) offers an efficiency of 59(+) percent and was introduced 2007. The plant can be started and loaded in 50 minutes[36] and offers high turn-down capability. The turn-down capability is down to 20 percent load is achieved by closing the compressor IGV/VSVs and shutting down the SEV-burner. This means that the plant can be operated at 160 MW maintaining 10 ppm(v) NOx. The ramp time from 20 percent load is 20 minutes.

10.6.3 Mitsubishi

Mitsubishi has been using steam cooling for its G-class and their newest J-class units. The J-class has the highest firing level in the industry and steam cooling is probably a requirement for sufficient lifing. MHI has also followed the current trend with high-performing air-cooled units and has recently released a new F-class.

[35] Not necessarily the worst component
[36] No available information on base (hot, warm, cold)

Appendix I. Introduction to gas turbine performance

The performance of a gas turbine is chiefly set by the air flow, firing level and pressure ratio. There is no true optimum combination valid for all gas turbines since the applications require different features. For a simple cycle, the optimum pressure ratio is relatively higher than for an engine aimed for combined cycle operation. The firing level is more complex, reflecting engine power density, efficiency, exhaust temperature (combined cycle efficiency) and lifing. The latter is indeed important in terms of availability and reliability where history has shown issues related to elevated firing levels. Most large state-of-the-art gas turbines have firing levels ranging from 1,300 to 1,525 °C. Smaller units (5-15 MW) typically runs at 1,100 to 1,300 °C. There is a size dependency in terms of firing (or metal temperature) and larger units are running hotter with the same cooling-, coating and material technology. The firing level itself is not the "true" driver in terms of work potential and efficiency and should be replaced with the difference between the stator outlet temperature and the compressor discharge temperature. I.e. when this figure is low, the component efficiencies get increased relative importance. The difference between the true combustor outlet temperature (COT) and the stator outlet temperature (SOT) is on the order of 100 to 140 °C. From a cycle viewpoint, the work is based on the mass flow and total enthalpy after the first stator. It is therefore instructive to introduce the concept of "non-chargeable" and "chargeable" cooling flows. Chargeable flows simply re-enters the mainstream after the rotor and not considered to do work in that stage. Non-chargeable cooling (i.e. vane cooling) are assumed to perform work with the associated drop in enthalpy (COT-SOT). The rationale for introducing steam cooled blading is to reduce the difference and increase the available air for dry low NOx combustion.

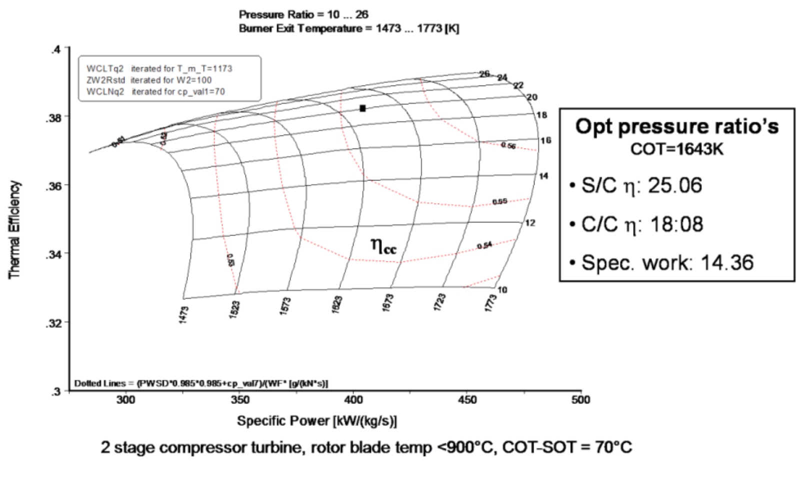


Figure 40. Example of cycle performance

The state-of-the-art gas turbine components have reached a level of maturity where one can expect e.g. compressor polytropic efficiency above 92 percent, or a turbine end-stage around 94 percent. N.B. the quoted numbers are size dependent, function of loadings and Mach numbers, etc. The first highly cooled stage suffers from its relative length and mixing of cooling. An old rule of thumb is, a drop of efficiency penalty for a four stage turbine is 0.1…0.2 percent per percent cooling. The underlying mechanisms are mixing, reduced temperature and lost work. The loss due to mixing is actually not too detrimental since good design practice is to keep the injection parallel to the main flow. The thermodynamic penalty can be reduced by using coolant from the penultimate compressor stage and swirl generators, etc. Leakage flows are much worse, mainly because the associated mixing with the mainstream is much worse – typically normal to the main flow at high velocities. The associated loss is about twice as high as for the cooling case. It is therefore very important for the designer to keep the "bookkeeping" and avoid leakage flows. Typical design features are vane segments, seals, TOBI-system[37] and packing flows[38]. One can expect to find the total cooling and secondary flow within the 18-24 percent range. There is a coupling to the firing level, but an equal driver is the cooling and sealing technology levels. In other words, one cannot draw any firm conclusion solely on assumed firing level.

Pressure drops is another principal loss source in a gas turbine where the designer strives to keep a minimum value. The combustor loss should, however, not be less than four percent if the first vane is to be film cooled. The combustor pressure drop sets the maximum pressure difference/potential for the film flow. It is quite typical to have double feeds for the first vane. Two feeds gives half speed and together with the common scaling ($\Delta p \sim velocity^2$) of pressure drops, results in one quarter pressure drop relative a single feed.

App. I.1 Frame units

A frame unit is characterized by its heavy rugged design which, by virtue of its size, is typically built in-situ. There is no direct coupling to the output class, but most frame-type units are in the higher power bracket. There are examples of engines in the 10 MW bracket, like the GE-5 and GE-10 by General Electric (fmr. Nuevo Pignone). One can follow the evolution of most frames by its characteristic letters A, B, C, …, F, G, H, … (e.g. PG9371(**F**) or SGT5-8000**H**) where each represents a development step. There is no direct definition of a development step but in most cases there is an increase in firing level and associated cooling technology. It is quite common to have features from more recent generations "back-flowing" to earlier as either upgrades or refurbishments (or both). One example is the General Electric Frame 9FB which basically is a 9FA with material developed for the 9H without steam cooling and a new compressor. The power class has no common structure between the OEM's (Original Equipment Manufacturers).

[37] Turbine On-Board Injection (TOBI) is part of the high-pressure feed to the rotor(s) and provides good control possibilities of the packing flows at the disk face(s).
[38] Packing flows are used for preventing from hot-gas ingestion and disk face cooling (design dependent)

Having established that one can expect frame-types from as low as some 5 MW up to more than 300 MW, then the role of typical applications should be explained. In the lower bracket, most frames are used for co-generation where the exhaust heat is utilized for production of large quantities of low-grade process type. The reasons for not having small combined cycles are driven by first cost and availability of such small recovery systems and steam turbines. The mid-size power range covers up to approximately 100 MW and competes with light industrials and aero-derivatives to 60 MW. The reason for the 100 MW limit is the possibility of having a geared design. A geared design introduces the possibility to significantly reduce the stage count in both the compressor and turbine. The underlying reason does not lend itself to a brief explanation but work input and output scales with blade speed whilst most loading parameters scales on speed squared. The mid-size market is typically aimed at simple or combined cycles. One important application is drivers for e.g. oil- and gas pipes, where the user is forced to use a twin-shaft for proper torque characteristics. The driver market is dominated by aero-derivatives since fuel spend is of great importance (next to reliability). The 50 and 60 Hz markets are typically approached with identical engines, maintaining speed with different gears.

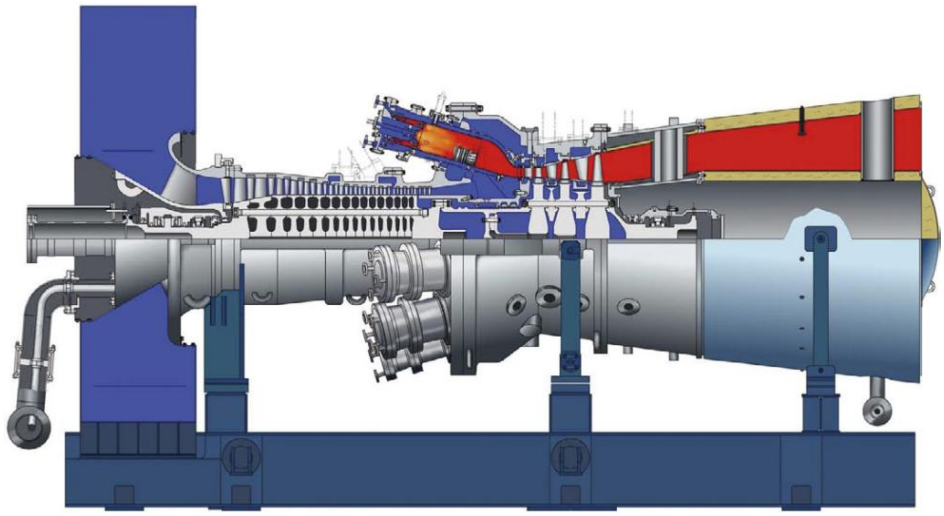


Figure 41. Hitachi H25 (Courtesy of Hitachi)

Above 100 MW there is neither a mechanical drive market nor the possibility to use a geared design. Another reason for the dominance of frames in this bracket is that there are no potential aero parent engines in this size. The manufacturers are therefore forced to approach the 50 and 60 Hz markets with different products at either 3000 rpm or 3600 rpm, respectively. Most OEMs applies simple scaling rules and develop low-risk versions at both speed levels. The basic principle is to maintain the same set of velocity triangles at the other speed level and the impact on output is $(60/50)^2=1.2^2=1.44$. I.e. a scaled 50 Hz unit should have an output that is about 1.44 times higher than the 60 Hz unit. The smaller 60 Hz unit typically has slightly lower efficiency since all features do not lend itself to simple scaling. Examples are Reynolds numbers, running clearances and trailing edge thickness. Most frames use can type of combustors and size adaption is readily available by changing the numbers of flame tubes. This also carries the same dynamics issues between the different sizes, hence providing some relief. Scaling also keeps the stress level identical in both cases and the only major difference is the power output.

The dominant application for large-size frames are high-efficiency combined cycles. A combined cycle prime mover has a different design requirement, where the net plant efficiency is of paramount importance. The gas turbine design route for a combined cycle plant is not straightforward and there are several conflicting requirements. One could say that it is beneficial to increase the gas turbine efficiency only if the drop in the steam cycle is not too big. The main driver for simple cycle efficiency in a gas turbine is pressure ratio, whilst the firing level mainly influences power density and exhaust temperature. Most, if not all standard, large frames have pressure ratios below 20 with a corresponding efficiency on the order of 38…40 percent. The resulting exhaust temperature may be above 640 °C, where three pressure level and re-heat (3P-RH) recovery systems are cost effective. The available steam production capacity from a large size gas turbine makes use of utility types of steam turbines practically. Today, several steam turbine manufacturers offer units that accept admission and re-heat temperatures up to 600 °C. The evolution in efficiency for steam turbine manufacturers like Alstom and Siemens has now reached impressive 94.1 and 96.2 percent efficiency for the HP- and IP cylinders, respectively. The presented numbers are from tests in the Boxberg lignite plant and are published by Siemens. A steam turbine for a combined cycle typically has a volumetric flow that is about one third of a coal fired plant. This gives a couple of points drop in efficiency. The maximum exhaust size for a full-speed 50 Hz low-pressure turbine is 16 m^2. This has been possible by using titanium blades in the last rotor and may result in cost effective HP-IP-LP(dual) configurations.

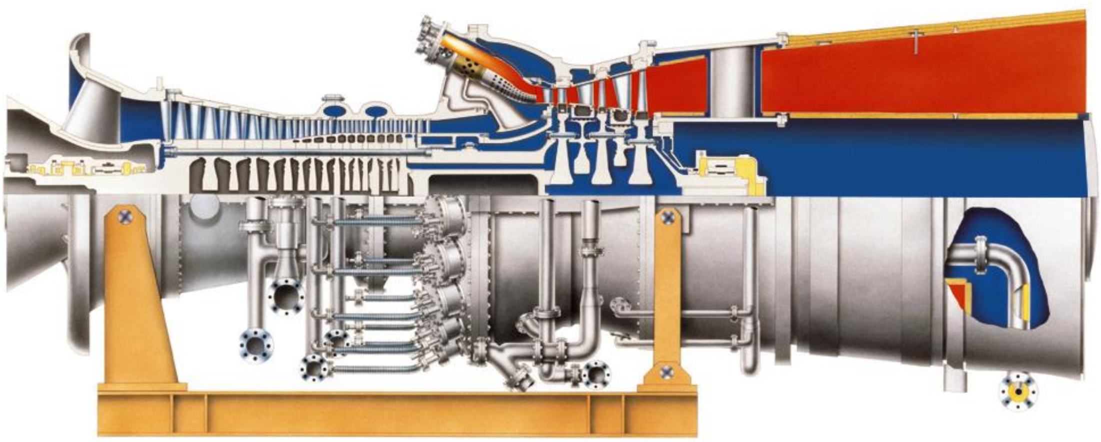


Figure 42. General Electric Fr7/9 Gas turbine

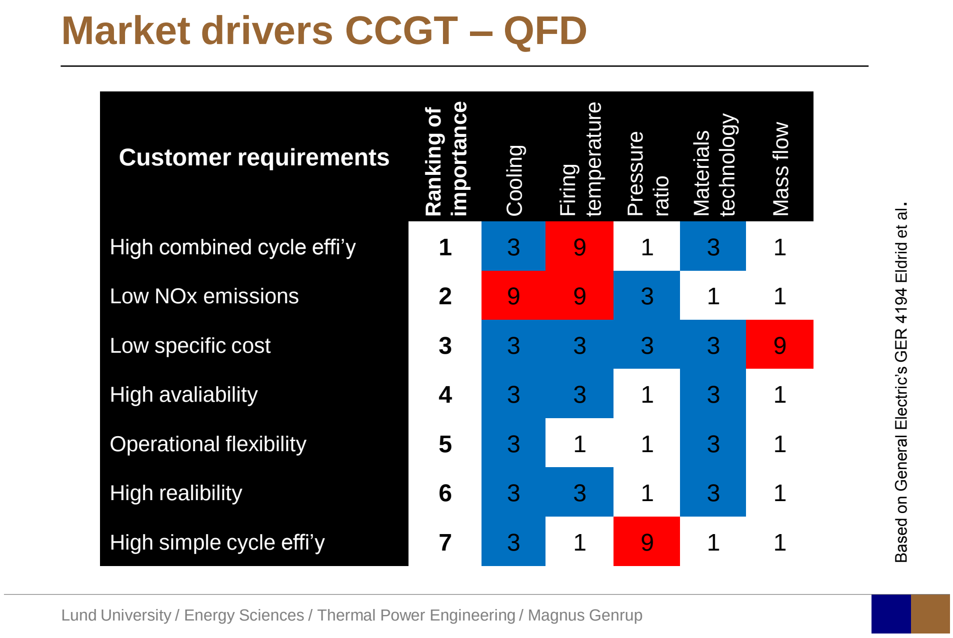
Most gas turbine OEMs offers gas turbines capable of reaching 60 percent efficiency in combined cycles for H-class. General Electric and Mitsubishi offer units that have steam-cooled combustion transition pieces (aka smileys), first nozzle vanes and rotor. Siemens offers the same efficiency level with their latest H-class without steam cooling. The rationale for having steam cooling is two-folded since steam offers better cooling potential and air can be saved for the combustion process. Less cooling results in a higher stator outlet temperature (SOT), hence higher work potential for the same metal temperature. The combustor outlet temperatures (COT) are in most cases proprietary but should be in the range of 1500…1525 °C. The reason for the OEMs not to publish the firing level is the exposure of details in the cooling and secondary air systems. The firing level itself is only one part of a rather complex combination of available cooling air temperature and available metal temperature. The latter is driven by either creep properties, oxidation properties or a combination of both. Despite fundamental different damage mechanisms, each increment of 10 degrees metal temperature reduces the available lifing by a factor of half. A typical highly stressed fourth generation single crystal blade (like CMSX-10) should be possible to operate at a metal temperature of 920…930 °C and still keep 40,000 hours in the engine. Most OEMs have a control strategy for their plant where there is no reduction in firing temperature until approximately half load, hence no gain in lifing due to part load. The control is achieved by reducing the mass flow by closing the variable geometry at the inlet to the compressor. This results in an effectively smaller engine with a hotter exhaust and the drop in cycle efficiency is counterbalanced by the steam cycle. The hotter exhaust doesn’t necessarily have a profound impact on the highly centrifugal stressed last rotor. A rotating blade’s temperature is mainly set by the relative inlet temperature, where only minute changes could be expected due to reduced engine pressure ratio. Recent work at Lund University shows that there is a way of providing relief/life extension without losing efficiency.

Most frame types gas turbines have state-of-the art compressors with high tip Mach numbers, high aerodynamic loading and moderate specific flows. The sizes are, by virtue, of the power class indeed very large and have high stage counts. Most advanced three-dimensional design features used in the aero-industry are now found in the full spectra of heavy frames.

The combustion technology has now reached a level where all manufacturers are able to offer single digit NOx and CO, where required.

The figure below is derived from a General Electric pubication showing a QFD analysis[39] from developing the PG9371FB engine. The red zones have a large impact (weight 9), the blue have medium (weight 3) and white little influence (weight 1). Each feature is weighted against the key performance indicators (1…10).

[39] Quality function deployment is part of six sigma and is a method to assess key performance indicators into design features.



Market drivers CCGT – QFD

Customer requirements	Ranking of importance	Cooling	Firing temperature	Pressure ratio	Materials technology	Mass flow
High combined cycle effi'y	1	3	9	1	3	1
Low NOx emissions	2	9	9	3	1	1
Low specific cost	3	3	3	3	3	9
High avaliability	4	3	3	1	3	1
Operational flexibility	5	3	1	1	3	1
High realibility	6	3	3	1	3	1
High simple cycle effi'y	7	3	1	9	1	1

Figure 43. Example of QFD analysis for GE Fr 9FB.

The analysis shows that a high performing combined cycle prime mover should have high firing for meeting efficiency expectations with a good cooling system for saving air for low emissions. The size is important for low specific cost and pressure ratio for simple cycle.

App. I.2 Industrial

It is sometimes hard to distinguish firmly between a small heavy frame and an industrial unit. An industrial unit is typically a geared single- or twin shaft unit between 100 kW and 40 MW. Many manufacturers offer both a single shaft and twin-shaft version for power generation and mechanical drive respectively.

A single shaft unit offers fewer complexities in terms of rotor design, where a common two-bearing beam type of rotor could be used. Both sub- and super critical designs are commonly used with varying rotor build concepts (e.g. welded or bolted compressor rotor). The need for a third bearing under the combustor (as for twin-shafts) is effectively avoided by the beam rotor. A bearing under the combustor introduces a high pressure lubrication oil level circuit and breeders for de-pressurization with associated complexity. The engine torque characteristics are unsuitable for variable speed operation since the torque increases linearly from zero at zero speed to nominal at full speed. The engine speed level is typically set by the balance between the compressor inlet geometry and the turbine exhaust size. The stress level at the last stage is normally the highest and poses a severe limitation in available size for a certain speed level. The direct centrifugal stress for a turbine blade could be shown to scale on annulus area times speed squared ($\sigma \sim AN^2$). The compressor stage count (and rotor length) is a strong function of engine speed and this is a strong incentive for choosing a high speed level. The typical limiting factor is the smaller available turbine exhaust and need for efficient exhaust kinetic energy recovery. To keep the exhaust losses at reasonable level, the designer should strive to keep the velocity at a maximum of Mach 0.6. A caveat is in place since all organizations have their own set of design rules and the quoted numbers are only given as ball park estimates.

A single-shaft unit is not an optimum solution for emergency power since the produced power drops steeply with load speed (e.g. grid code emergency operation). Another drawback is high starting power requirement. Both issues are effectively avoided with a multi-shaft unit, since the gas generator operates independently from the load turbine (to a first order) and the starting power is significantly less.

A twin-shaft unit offers a wide application range since the unit can be used for all types of plants. The engine has a potential for several fewer compressor stages and typically one additional stage in the turbine (3 or 4 in total). The compressor capacity is typically at a level of 900…1000 kg/s scaled to 3000 rpm. This figure can be thought of as a comparison between a non-geared 3000 rpm unit with a typical number on the order or 650 kg/s. I.e. a higher throughflow level maintaining the same aerodynamic characteristics at a higher speed level. The number of compressor turbine stages varies, but most pressure ratios and firing levels are covered with two stages. One could consider a single compressor turbine stage if the pressure ratio is lower than approximately 18. This is also highly dependent on the engine firing level, acceptable inter-turbine swirl level and Mach number.

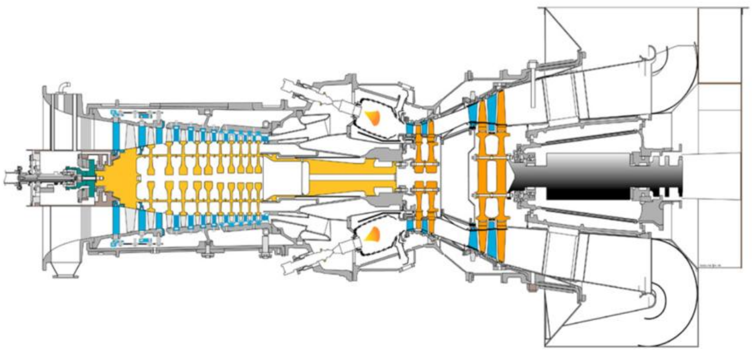


Figure 44. Example of an industrial twin-shaft gas turbine (Siemens SGT-700)

Many industrial units are used in simple cycle mode for either power generation or mechanical drive and the need for high simple cycle efficiency results in higher levels of engine pressure ratios. There are examples of industrial units (Solar Titan 250) with efficiency levels exceeding 40 percent. The aero-derivative lead in efficiency has been reduced significantly over the last 10 years – mainly driven by pressure ratio and component efficiency.

There is no practical means of controlling the airflow to a multi-shaft unit and part-load control of CO emission it typically introduced by variable combustor geometry.

App. I.3 Aero-derivatives

The era of aero-derivatives started with Pratt & Whitney and General Electric where, at that time, successful flying engines where equipped with a power turbine. All western-world aero engine companies have a product range with derivative engines. A state-of-the-art aero-derivative of today is not a simple standard jet engine with a rugged power turbine. All sub-sonic fix-wing transport engines have large by-pass ratios with a large fan. The fan has to be removed when turning a flying engine into a land based unit. The excess power is then turned into shaft power driving an alternator. It is not possible to generalize the exact process of transforming the engine since the original architecture varies. General Electric and Pratt & Whitney typically have large twin-spool, whilst Rolls-Royce builds three-spool engines in the higher trust bracket. These features are normally carried into the land-based range as either compound engines or standard twin- or three spool aero-derivatives. The latter are generally very similar (or vice versa) to industrial engines.

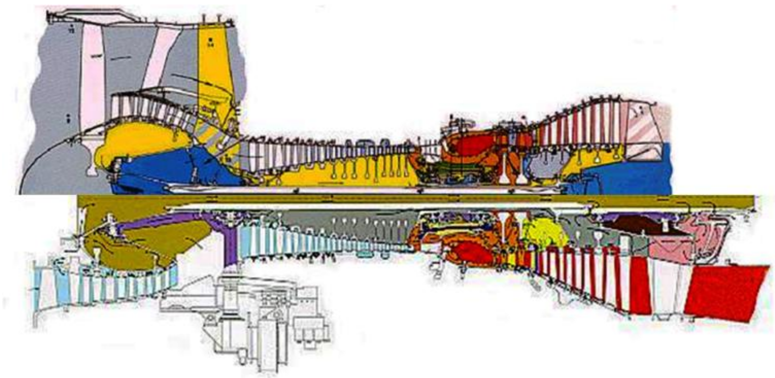


Figure 45. CF6 aero-engine and derivative LM6000

A modern flying engine typically has a pressure ratio well above 30 and offers a high efficiency potential when turned into a land-based engine. Both General Electric and Roll-Royce offers engines with efficiencies exceeding 42 percent. GE has a recent 100 MW inter-cooled peak-lopping unit with a claimed efficiency of 46 percent. The pressure ratio is on the order of 42 and the only practical ways to achieve this level is to introduce inter-cooling and multi-spool. The cooling plant has the size of a normal city bus and increases the footprint significantly. The engine is capable of being started and fully loaded within 10 min, hence offering a potential compromise between efficiency and flexibility.

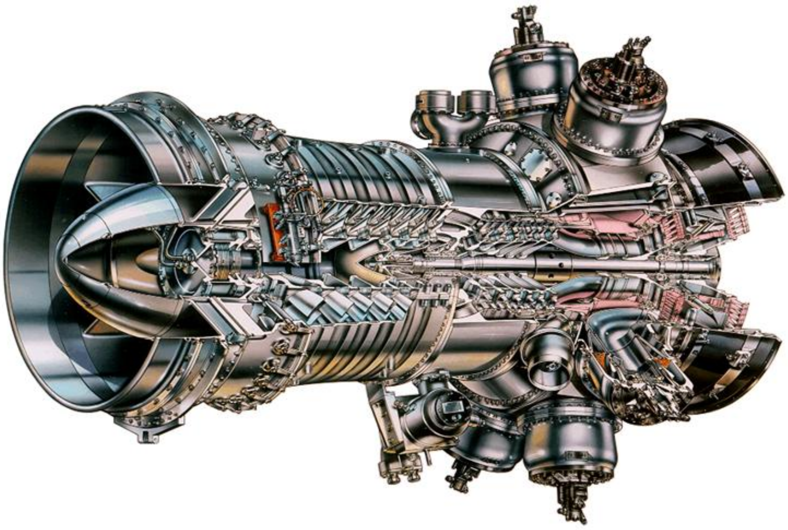


Figure 46. Rolls-Royce RB211 gas generator

Another prominent aero-derivative is the General Electric LM2500, with current sales on the order of 2000 engines. The engine has been highly successful on all land-based and marine markets. The engine is derived from the first successful high by-pass turbofan and was introduced in the 70s. The initial rating was at 25000 hp (LM2500 is an acronym of light module with a rated power of 25000 hp) and the engine has today a fourth generation rating of 33.9 MW with an efficiency of 39.6 percent. This engine has followed a typical evolution process with an increase in performance 17.9 MW@35.8% to 33.9MW@39.6%. The biggest step in performance was the introduction of the "+"-rating, which incorporated a zero stage blisk (bladed disk) for increasing the engine flow and pressure ratio. The LM2500P has the lowest hub/tip ratio design in the entire business resulting in a high throughflow capacity at lower relative tip Mach number. All four major evaluation steps have introduced improved components and increased firing.

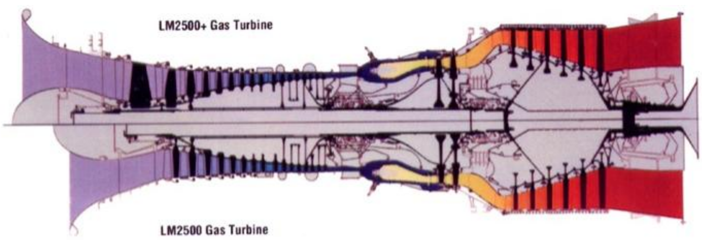


Figure 47. General Electric LM2500 vs. LM2500+ (Courtesy of General Electric)

The figure below shows the evolution history of the RB211-24 and Trent engines. Both engines offer efficiency levels above 40 percent with DLE technology.

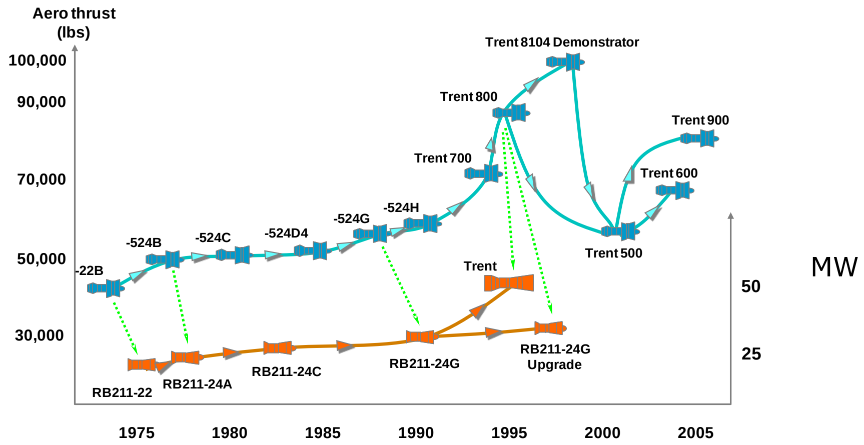


Figure 48. Rolls-Royce aero-derivatives and parent engines

App. I.4 Advanced cycles

The combined cycle is the predominant process for power generation with its high efficiency potential. There are both size aspects of combined processes and application driven issues – like naval ships. For the latter, part load performance is of paramount importance since a naval ship will spend most of its time cruising at half speed. Half speed will, due to the cubic relation between speed and power, result in $(1/2)^3=1/8$ of maximum power. Since virtually the complete gas turbine loss is in a single source in the exhaust, a recovery system will provide relief. The idea of recuperated (or regenerative) engines is certainly not new. There have been examples even from the 50s (HMS Gray Goose), but today only two engines are available above 1 MW namely Rolls-Royce WR-21 and Solar Mercury 50. The Rolls-Royce WR-21 is currently only available within their marine product range. A recuperated engine should have relative low pressure ratio for good exhaust heat utilization – hence less complicated turbomachinery. The concept has been used on a very small basis since the introduction of land based gas turbines and the reason is reliability. The heat transfer from a gas to another gas requires large surfaces (due to low HTC) and in concert with a large pressure difference, results in a leaky large structure. There has been little (for not to say no) market acceptance and the situation will probably be the same until the lifing of the system reaches 120,000 operating hours.

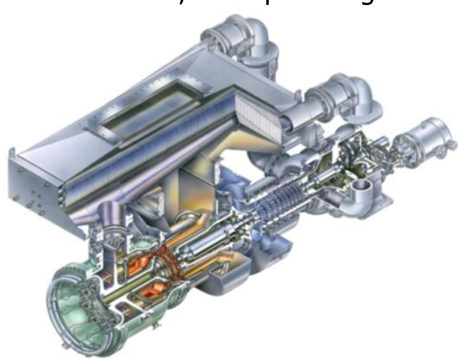


Figure 49. Solar Mercury 50 recuperated engine.

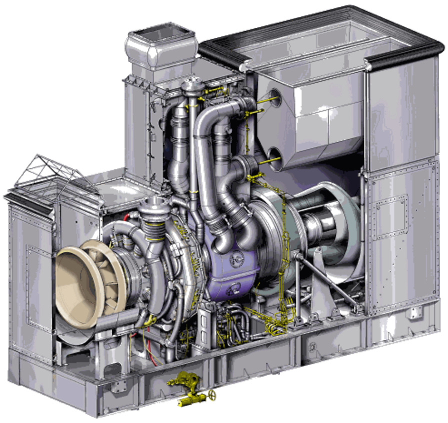


Figure 50. Rolls-Royce WR-21 ICR (Courtesy of Roll-Royce)

There have been recent developments in introducing advance cycle turbo-fan engines for flying applications. This is still in its infancy and there will certainly be many years of development before a certified mature product is available on the market. The combination of inter-cooling and recuperation[40] is very useful since the available exhaust heat that can be transferred increases. This concept is also available in turbo-fan engines by virtue of the large by-pass flow. An in-depth analysis of advanced aero-engines is outside the scope of this report – but the technology will certainly find its way into land-based engines. Again, the market has a strong reluctance for recuperated engines but massive research activities are ongoing.

High power density re-heated engines are also beneficial if the exhaust heat can be recovered. At the time of writing, only one commercial engine exists (Alstom GT26). There is one scaled ($SF=1.2^2=1.44$) 60 Hz version for the North American 60 Hz market (Alstom GT24). The pressure ratio of such engine should be higher than a normal cycle for a combined process. The GT26 has a pressure ratio of 30 and this level is considerable higher than e.g. a GE Frame 9 with approximately 19. The efficiency of the 424 MW Alstom plant is 58.3 percent, lagging 1.7 units behind[41] the best available technology. Alstom also presented an "overnight" concept at ASME in Orlando 2009, where the secondary burner is shut-down for low-load overnight operation.

One immediate conclusion is that despite the performance and cost potential, most OEMs have not introduced these features. Both the Solar Mercury 50 and Alstom GT2X have had problems in the past but are now available.

Wet gas turbine technology has been developed since late 30s without being able to get market acceptance. One (or probably the) reason for this is the market reluctance for recuperators. The cycle, in full bloom, has the potential of reaching the efficiency level of a combined cycle. The inherent advantage is, on top of efficiency, rapid start and ramping capability. The figure below

[40] Rolls-Royce WR-21 has both inter-cooling and exhaust heat recovery with variable power turbine geometry.

[41] The presented numbers are based on GTW. There are claims of 59.5 and the target is 60 percent.

has been presented at European Turbine Network (ETN) in 2006 (and web published) and is showing a comparison between different cycle options.

The figure also includes "High-OPR IC+STIG", which is similar in concept to the Swedish "TopCycle" by EuroTurbine. There is no information about the technology platform, or shaft-configurations in the figure below. The output level (or torque), however, indicates another platform than the current largest Rolls-Royce Trent 60 engine. The performance is on the order of 150 MW at an efficiency of almost 55 percent. The Topcycle concept by EuroTurbine has a single-shaft low-pressure cycle and a separate high-pressure part with its own alternator. The TopCycle is based on a 40 percent "low-flowed" compressor. This level of de-rating is probably only feasible by having a smaller flow path and certainly beyond the possibility by simply removing the first stage(s) and increasing the shaft speed.

The challenges shown in the chapter about fuel flexibility also applies to steam injection. The steam injection (or any additional flow added to the unit before the turbine but after the compressor) upsets the engine matching and pushes the rear (and penultimate) stage(s) into positive stall and ultimately surge. Various methods are briefly discussed in the chapter related to fuel flexibility – but most can be boiled down to either reduce the compressor pumping capacity, or increase the turbine swallowing capacity.

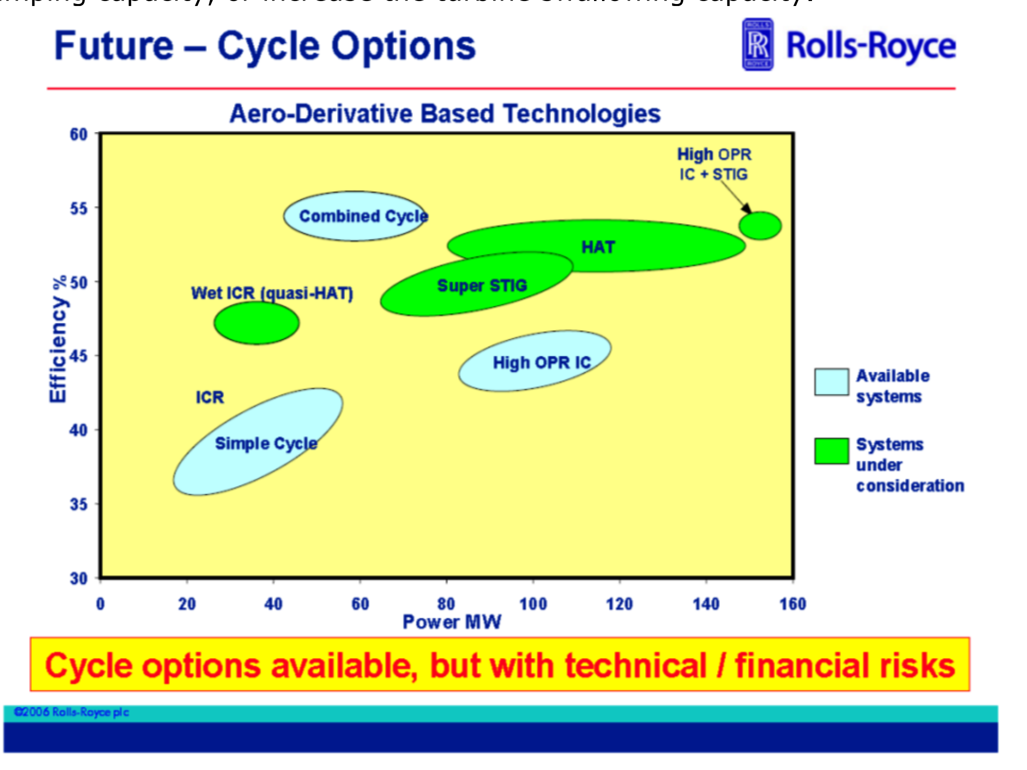


Figure 51. Rolls-Royce cycle view

App. I.5 Typical evolution paths

Most engines are introduced at "introductory" rating where the engine type is operated until the design is proven. The typical initial rating is a reduction in firing level and a reduction in flow capacity by means of the compressor variable geometry. Once introduced at its design rating, the typical growth evolution may follow the paths:

(i) Increased firing

(ii) High-flowing or zero staging

(iii) Linear scaling

(iv) Increased pressure ratio (rear stage with unchanged flow)

Increased firing is typically associated with either improved cooling or materials when sufficient experience is gained. It is not uncommon to introduce combinations, i.e. both improved cooling and material temperature capability. Some OEMs carry out rainbow tests where an engine is furnished with different configurations for proof of design before market release. This significantly reduces the risk for the end user and insurer.

High flowing is a re-design of the front end of the compressor where either the flow path height is increased or the stages are set for higher capacity. Both changes may incorporate aerodynamic re-design of the stages. Zero staging gives typically an increase in capacity of 15-20 percent and additional pressure ratio (extra work is added). In both cases, the rear stages are operated at changed incidence level. Zero staging is normally not possible on single shaft units due to the turbine exit velocity level. It is not uncommon even for very competent designs to have issues with aero-elastics. Many multi-shaft engines have an upper aerodynamic speed where the engine is topped[42]. Aerodynamic speed[43] is used synonymously with referred speed and is more relevant than physical ditto. This limit is reached on low ambient temperatures in contrast to high physical speed topping at high ambients.

Linear scaling is a common method for low-risk development where the velocity triangles are kept unchanged. Unchanged velocity triangles also result in the same disc and blade stresses. This method has been commonly used for scaling between 50 and 60 Hz.

Added rear stage gives higher pressure ratio capability (together with reduced turbine capacity).

[42] Topping is commonly referred to when the engine is operated at lower firing level to maintain e.g. maximum gas generator speed(s)

[43] Reflects compressor velocity triangles based on Mach number

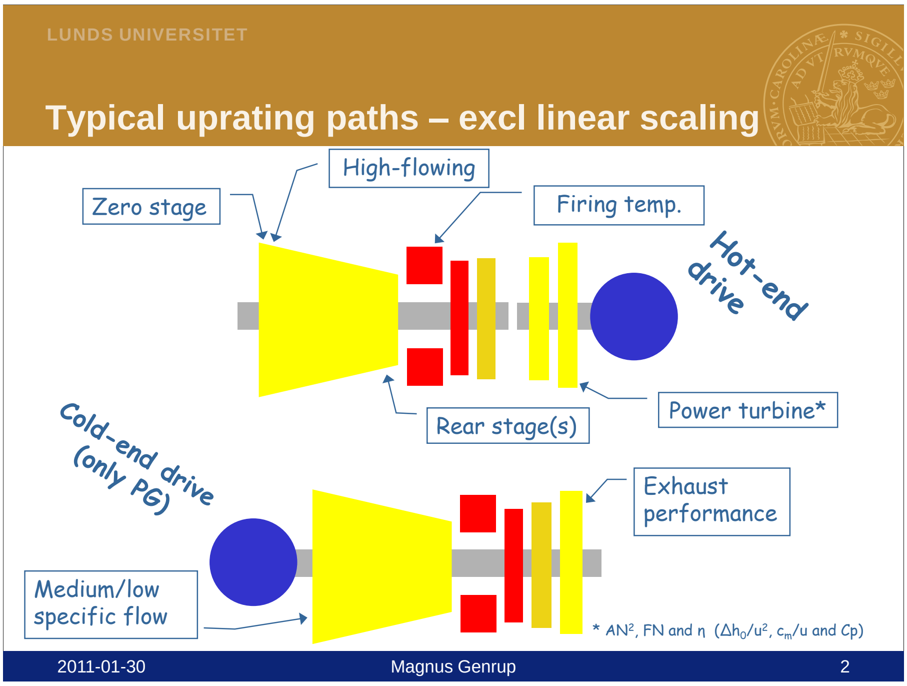


Figure 52 – Typical evolution steps

The figure below shows Solar™ turbines compressor product development. Solar has used all available linear scaling and adding (removal) of stages for a large product portfolio. It is impossible to say if these are the only changes between the models. It is quite common to fix issues and improve e.g. stage matching when redesigning.

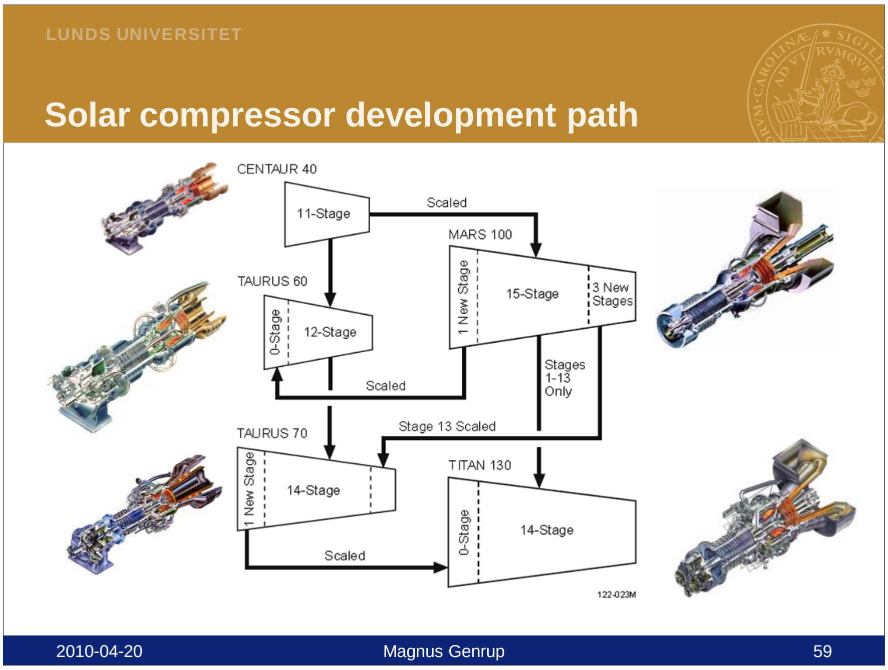
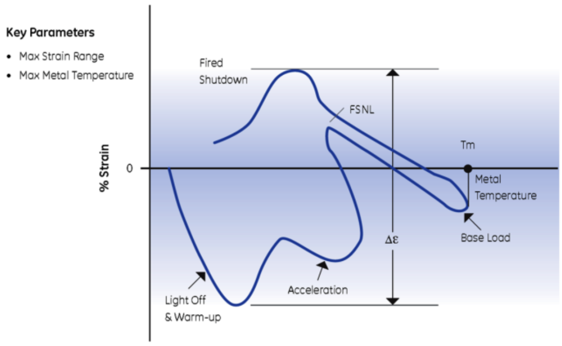


Figure 53. Solar compressor family

The growth potential of an engine typically does not come for free. The designs have to have a "built-in" margin in terms of velocities, basic cooling system, stage count, etc.

App. I.6 Hot component failure modes

The most common hot end failure modes are: creep, thermal fatigue, hot corrosion and oxidation. Creep is plastic deformation that occurs when a material is loaded over time at elevated temperatures. This effect is present even if the stress is not exceeding the yield limit. Thermal fatigue is caused by engine handling like starting and stopping and local temperature gradients (see figure). The latter may be due to combustor hot- and cold spots and typically has the highest influence on this particular failure mechanism. A hot engine trip may give 8 times the strain compared to a normal shut-down, i.e. a trip cycle can be equivalent of eight normal cycles. The previous failure modes are mechanical (or structural), whilst hot corrosion and oxidation affects mainly the hot surfaces. Hot corrosion is typically due to either air- or fuel borne vanadium, sodium and sulphur, causing a corrosive attack on the material. Oxidation is when the surface material reacts with the oxygen in the hot gas. A material with high oxidation resistance does not have the strength of a nickel-based material, hence should be used for coating. A typical coating is aluminides, which forms a thin oxide on the blade, resisting further attack on the blade. The coating is brittle and particles break away when the engine runs through different operation modes. Fortunately, a new oxide film forms as long as there is coating material left.



A turbine failure is indeed expensive and Boyce gives the following average costs for a turbine failure (in 2002 USD):

1-10 MW	USD 55,000…100,000
10-50 MW	USD 500,000**(+)**
>50 MW	USD 700,000**(+)**

The time for repairing a turbine failure is typically between 12 to 16 weeks for a heavy-frame. Smaller size units offer the possibility for an engine swap, where the plant can be up and running within 24 hours from the replacement arrives to the site.

The associated costs are indeed high and the costs for a competent monitoring system are justifiable if a single turbine failure can be avoided. There exists several technologies that will do the job but the most important thing is to have the competence of the OEM in the analysis. Any competent system should, as a minimum, look for subtle changes in the exhaust temperature pattern and changes in NO_X-levels. The easiest way is to plot the temperature pattern in a polar plot together with an expected profile. How to create the expected profile is outside the scope of this report – but artificial neural networks (ANNs) have proven to be a possible way forward here. If a "dent" in the profile rotates/moves with load, then the problem most likely is due to a real engine failure or burner problem. All engines have an alarm (and ultimately trip) if one individual value departs too much from the average.

App. I.7 ANSQ explained

The ANSQ or AN^2 is a gauge of the root stress of the blade and the definition is included for completeness. The derivation is based on the method by Jack Mattingly.

The root pull or total centrifugal force can be written as:

$$F_c = \int_{r_{hub}}^{r_{tip}} \rho \omega^2 A(r)\, r\, dr$$

The principal tensile stress at the hub is F_c/A_{hub}, hence:

$$\sigma_c = \frac{F_c}{A_{hub}} = \rho \omega^2 \int_{r_{hub}}^{r_{tip}} \frac{A(r)}{A_{hub}} r\, dr$$

The area ratio within the integrand is certainly monotone but probably not linear. It is, however, justifiable for the sake of the discussion to assume a linear dependency, as:

$$\frac{A(r)}{A_{hub}} = 1 - \left(1 - \frac{A_{tip}}{A_{hub}}\right)\left(\frac{r - r_{hub}}{r_{tip} - r_{hub}}\right)$$

The preceding equation is inserted to the principle stress equation to yield:

$$\sigma_c = \rho \omega^2 \left\{ \frac{A}{2\pi} - \left(1 - \frac{A_{tip}}{A_{hub}}\right) \int_{r_{hub}}^{r_{tip}} \left(\frac{r - r_{hub}}{r_{tip} - r_{hub}}\right) r\, dr \right\} =$$

$$= \frac{\rho \omega^2 A}{4\pi} \left\{ 2 - \frac{2}{3}\left(1 - \frac{A_{tip}}{A_{hub}}\right)\left(1 + \frac{1}{1 + r_{hub}/r_{tip}}\right) \right\}$$

The hub-tip-ratio (r_{hub}/r_{tip}) is limited to unity and the resulting upper limit is:

$$\sigma_c = \frac{\rho \omega^2 A}{4\pi}\left(1 + \frac{A_{tip}}{A_{hub}}\right) = \rho A N^2 \frac{\pi}{3600}\left(1 + \frac{A_{tip}}{A_{hub}}\right)$$

The stress ratio between an un-tapered blade (i.e. $A_{tip}=A_{hub}$) and a blade with infinite taper ($A_{tip}=0$) is two. Hence, the maximum reduction in stress by introducing taper is a factor of 0.5. The preceding equation shows that the centrifugal stress is proportional to AN^2. The typical range of maximum AN^2 is $40…60\times10^6$. The largest steam turbine exhausts (16 m^2) uses titanium blades but this is not feasible for gas turbines. One promising candidate for gas turbine last stages is γTiAl, which has a density about half of the equivalent nickel-based alloy (e.g. Inconel). Another promising feature is the high ratio of Youngs modulus (E) to the density. One can easily show that the natural frequency of a material scales with $\sqrt{(E/\rho)}$. It is easier to meet frequency issues with such materials, hence larger freedom for aerodynamic perfection. One draw-back, however, is the low ductility of the material.

App. I.8 Shaft configurations

Several gas turbine schemes have been suggested throughout the evolution of gas turbines. The probably most famous investigation is the one by Mallinson and Lewis (Royal Gas Turbine Establishment) in the late 40's. In their study, the part load performance of a large number of configurations was investigated. Their report discusses several pros and cons that were valid when the technology was in its early phase. Today, we have variable geometry in e.g. compressors and can have trouble-free off-design operation with compound schemes. An in-depth discussion of all concepts is outside the scope of the report and the reader is referred to the standard gas turbine literature for information. The figure below shows the 20 schemes from the late 40s and the current productions engines are market with red rings. The PFBC-unit is market with a hatched ring. In the original report by Mallinson and Lewis, the concept was reported to have very unfavorable operational characteristics. Again, variable geometry solved the issues with the engine matching. The same concept was used on the famous Gray Goose ship.

The current production engine shaft configurations are:

- Single-shaft (C-CC-T)
- Twin-shaft (C-CC-CT-PT)
- Three-shaft (LPC-HPC-CC-HPT-IPT-PT)
- Compound (LPC-HPC-CC-HPT-LPT)

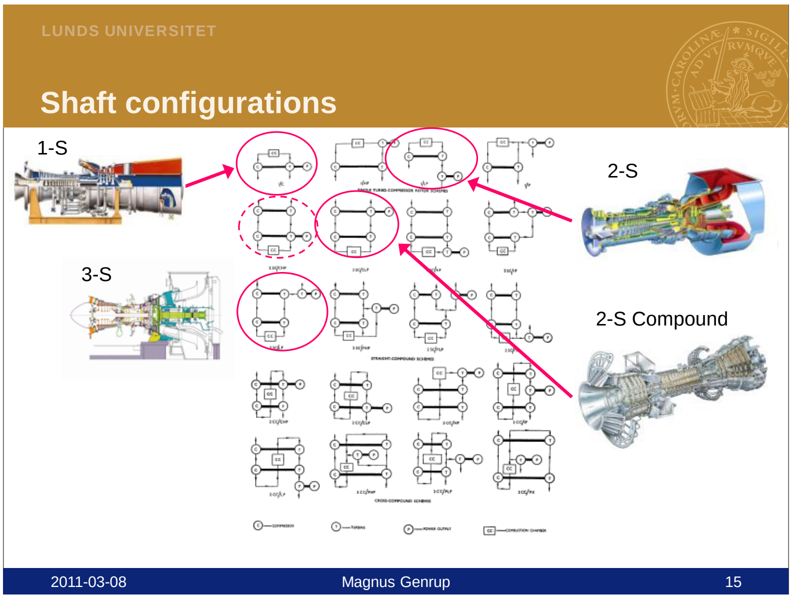


Figure 54. The Mallinson and Lewis shaft configuration study

Appendix II. Carbon Capture and Storage[44]

App. II.1 Introduction

Life on earth depends mainly on the atmosphere greenhouse effect. A differentiation between natural and man-made greenhouse effect must be made however. The natural greenhouse effect renders the earth environment inhabitant where without it the earth temperature would drop to -18°C. The man-made greenhouse effect however occurs by burning fossil fuels that increases the atmospheric concentration of CO_2. As a consequence earth global temperature including oceans also increases. Subsequently earth balance is also changed, leading to natural disasters we have become used to witness. Fossil fuel can be exploited in energy production, transportation, industry and houses. Current fossil fuel consumption shows no indication of decreased future energy consumption, more the opposite. New energy legislations together with efficient and renewable energy production as well as a more environment-conscious life style contribute to lower emissions.
The European Commission agreed on the "climate and energy package" in June 2009. The directives agreed upon in the package state that emissions should be cut by 20%. Furthermore, a 20% improvement in energy efficiency and a 20% increase in renewable energy should be achieved by 2020. The climate package comprises four main measures, including the revision and strengthening of the Emissions Trading System (ETS) as well as emission regulation in sectors not covered by the ETS, e.g. transport and housing. The measures further underline the importance of binding national targets for renewable energy and encouragement of the development and safe use of carbon capture and storage. The purpose of a carbon capture and sequestration (CCS) is to produce a concentrated CO_2 stream that can be easily transported to the end storage. CO_2 capture and storage is most applicable to large, centralized sources like power plants and large industries. At present, CO_2 is routinely separated at some large industrial plants such as natural gas processing and ammonia production facilities; however, these plants remove CO_2 to meet process demands and not for storage. There are three main technologies for CO_2 capture for industrial and power plant applications, namely: Post-combustion, Oxy-fuel combustion and Pre-combustion. Post-combustion technology separates CO_2 from the flue gases produced by combustion of a fuel in air. Oxy-fuel combustion uses oxygen instead of air for combustion, producing a flue gases that contains mainly H_2O and CO_2; thus CO_2 is easily separated by condensing the water vapour. Pre-combustion technology processes the primary fuel in a reactor to produce separate streams of CO_2 for storage and H_2, which is used as a fuel.

[44] Written by Majed Sammak, LTH

App. II.2 Post-combustion capture technology

The most common method for separating CO_2 from a gas stream in use today is the chemical absorption using alkaline solvents, figure 55. The flue gas passes through an aqueous alkaline solvent, and since CO_2 is acidic it is bound to the solvent. The absorption process takes place in the absorption column. The flue gas enters at the bottom of the absorber, while the solvent is pumped to the top of the absorber. After the reaction and CO_2 absorption, the rich-CO_2 solvent drops to the bottom of the absorber and then it is pumped to the separation unit (CO_2 stripper). In CO_2 stripper the rich-CO_2 solvent is heated up depending on the solvent type to 100-140°C. This reverses the absorption process and releases most of the CO_2 in a pure stream for compression and transport. The lean-CO_2 solvent is transported back to the absorber for reuse. The most used solvent for CO_2 capture is monoethanolamine (MEA). Aqueous Ammonia is used as well in capturing carbon dioxide from flue gases.

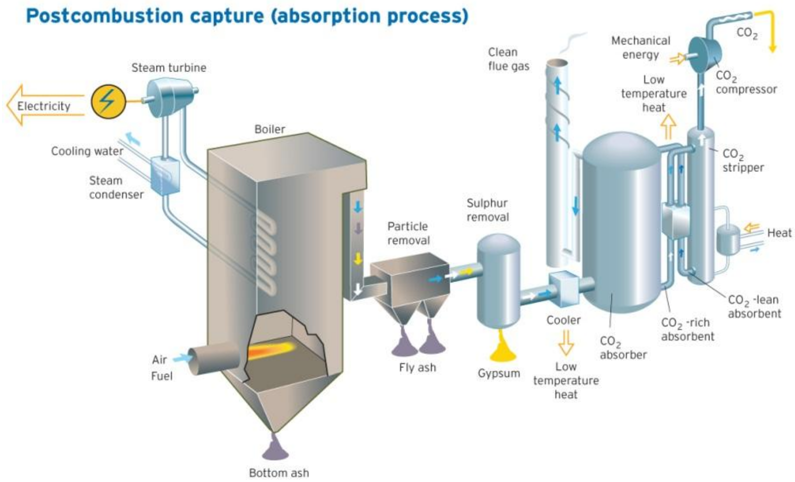


Figure 55. Post-combustion capture technology, source Vattenfall.

App. II.3 Oxy-fuel combustion capture technology

In oxy-fuel combustion technology nearly pure oxygen is used for combustion instead of air, figure 56. This result is a flue gas that is mainly CO_2 and H_2O. If fuel is burnt in pure oxygen, the flame temperature is excessively high and thus CO_2 and/or H_2O-rich flue gas is recirculated to the combustor to moderate the temperature. The steam can easily be removed by condensation, leaving a rich- CO_2 stream ready for compression and storage. Oxygen is usually produced in cryogenic air separation unit (ASU) which is the major demanding component in the process.

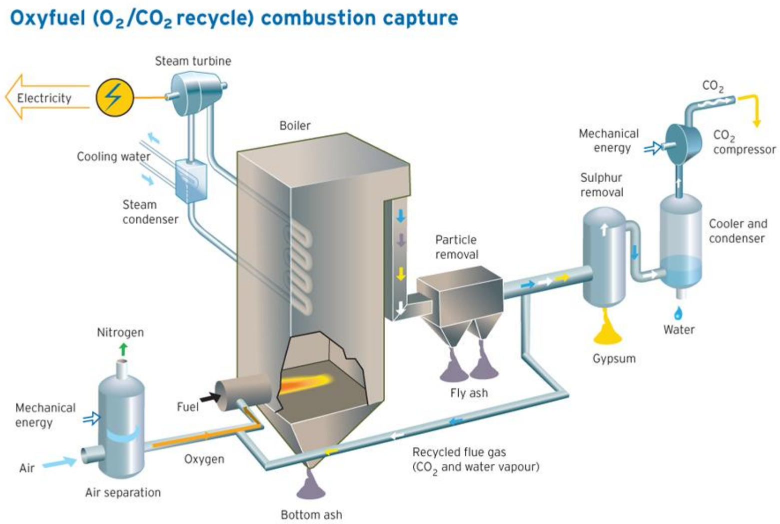


Figure 56. Oxy-fuel combustion capture technology, source Vattenfall.

App. II.4 Pre-combustion capture technology

The pre-combustion technology refers to removing of the carbon from the fuel before combustion, figure 57. Pre-combustion capture involves reacting fuel with oxygen or air and/or steam to give mainly syngas composed of carbon monoxide and hydrogen. The hydrogen can then be used directly as a fuel in the gas turbine. Burning hydrogen emits no CO_2 and the primary exhaust gas from hydrogen combustion is water.

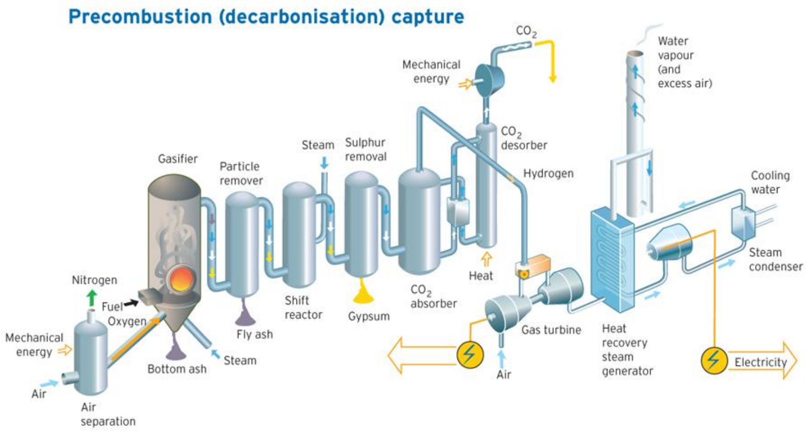


Figure 57 Pre-combustion capture technology, source Vattenfall.

App. II.5 Post Combustion Capture[45]

This chapter has been divided into two main topics, solvent development and process development.

App. II.6 Solvent Development

A variety of solvents could be used in the absorption/regeneration process and each has its advantages and disadvantages. The most important physical factors in solvent selection are high thermal stability, heat of absorption/regeneration, CO_2 absorption rate, resistance to degradation and impurities and volatility rate. Choosing the right solvent is important to reduce the energy penalty of the capture process. A solvent with a low heat of absorption requires less energy during regeneration, which can lead to significant energy savings. A solvent with a high absorption rate minimizes absorber size and pressure drop across the absorber and the associated pumping costs. Solvents are desirable also with high tolerance to SO_2 and NO_x and low cost. High resistance to degradation reduces solvent make-up costs and can reduce gas clean-up costs in the stages before the absorption system. Corrosive solvents are undesirable because they reduce process equipment lifetime. Solvent volatility is also an important issue since a volatile solvent can escape with the flue gases to the atmosphere. These solvents could have impact on both the environment and human health. The most attractive solvents for CO_2 capture are amine solvents and aqueous ammonia. There are several post-combustion capture technologies that are used commercially or under pilot demonstration based on amines solvents and ammonia aqueous. The amine solvent that has proved to be of commercial interest for CO_2 capture is monoethanolamine (MEA).
The energy required for CO_2 regeneration is usually extracted from the power plant. The CO_2 stripper reboiler heat duty is calculated as the following:

$$Q_{Reboiler}=Q_{Sensible}+ Q_{Vaporization} + Q_{Absorption}$$

The reboiler heat duty can also be calculated from the mass balance around the CO_2 stripper, figure 58.

[45] This chapter is based on the attended conference 1st Post Combustion Capture Conference PCCC1, 2011, Abu Dhabi.

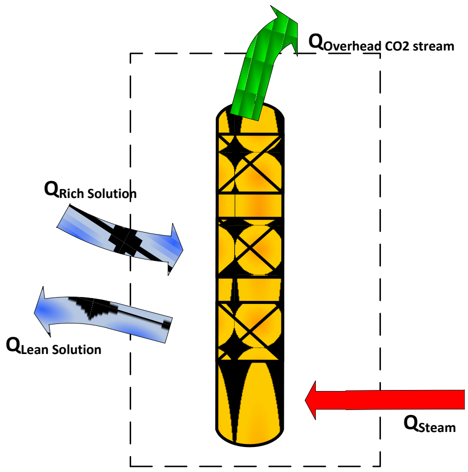


Figure 58. CO_2 regenerator mass balance

$Q_{Reboiler} = Q_{Steam} + Q_{Sensible} - Q_{Overhead\ CO2\ stream}$

The reboiler heat duty can only be reduced by lower heat of absorption or increase in concentration of the solvent. The heat required for CO_2 regeneration varies up to around 4 GJ/ton CO_2. The heat exchange between the CO_2-rich solution and the regenerated lean solution in the rich/lean heat exchanger alleviates the rich solution temperature before it's pumped to the CO_2 stripper. This reduces the reboiler heat duty and, as a consequence, reduces the total parasitic load.

App. II.7 Monoethanolamine (MEA)

Monoethanolamine (MEA) is the most common amine solvent currently produced and has been used for more than 50 years for boiler flue gas CO_2 recovery. The choice of amine concentration may be quite arbitrary and is usually made on the basis of operating experience. Typical concentrations of monoethanolamine range from 12 wt% to a maximum of 32 wt%[46].

Most of the existing thermodynamic models for MEA are based on experimental data for 30wt% MEA concentration. The chemical equilibrium in H_2O-MEA- CO_2 system can be described by the following:

Dissociation of water

Dissociation of carbon dioxide

Dissociation of bicarbomate

[46] Liang-shih fan, 2010, Chemical looping systems for fossil energy conversions", ISBN 978–0-470–87252–9

Dissociation of protonated MEA

MEA carbamate revision to bicarbonate

The chemical and phase equilibrium are described in the figure 59[47].

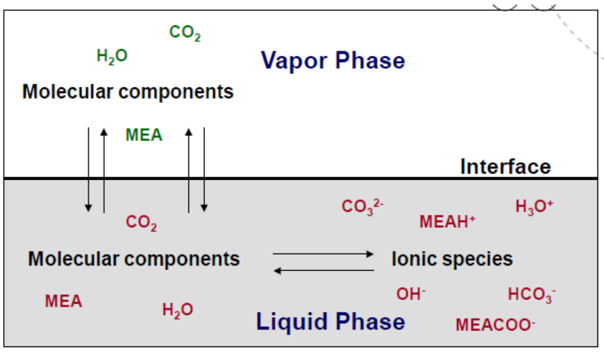


Figure 59 MEA chemical and phase equilibrium

There are many efforts to develop new MEA with lower heat of absorption in order to reduce the reboiler heat duty. Mitsubishi Heavy Industries (MHI) and Kansai Electric Power Co. (KEPCO)[47] have developed an energy efficient solvent KS-1[48]. KS-1 reacts with one mole amine and one mole CO_2 and thus has higher CO_2 absorption efficiency and requires less energy for regeneration. According to MHI, the new solvent KS-1 reduced regeneration energy with 20% compared to MEA. KEPCO research institute has also developed a new amine solvent (KoSol series). KoSol has been tested and gave 30% reduction in regeneration energy, from 3.8 to 2.7 GJ/ton CO_2[49].
MEA solvent has tendency to react with NO_X and SO_2 to form a salts that reduce the absorption capacity of the solvent because these salts are not re-generable. It is expected that 1 mol of NO_x or SO_2 in the flue gas will lead to a loss of 2 mol of MEA solvent. Thus, the presence of SO_2 and NO_x in the flue gas can increase significantly the operating cost of the MEA system and the removal prior to the MEA system is thus necessary. The gas turbine combined cycles have advantage with very low percent of SO_2 present in the gases compared to the coal fired power plant. Thus flue-gas desulfurization plant (FGD) doesn't use in gas turbine power plants.

[47] Ugochukwu, E., et al, 2011," Equilibrium in the H_2O-MEA-CO_2 system", 1st post combustion conference.
[48] Lee, J., et al, "Development of Amine Absorbents for Post-combustion Capture", KEPCO Research Institute, 1st post combustion conference.
[49] These numbers are taken as the optimal energy requirement however it depends on the CO_2 concentration.

App. II.8 Aqueous Ammonia

Aqueous ammonia is used in two types of absorption processes, at ambient temperature and at low temperature (Chilled Ammonia). Chilled ammonia is patented by Alstom and is similar to the MEA process. However, it operates at low temperature of absorption which leads to precipitation. Ammonium carbonates and bicarbonate slurries are used to capture the CO_2 in the flue gas stream at a temperature around 10°C and atmospheric pressure. The chemical equilibrium reactions are described in figure 60[50].

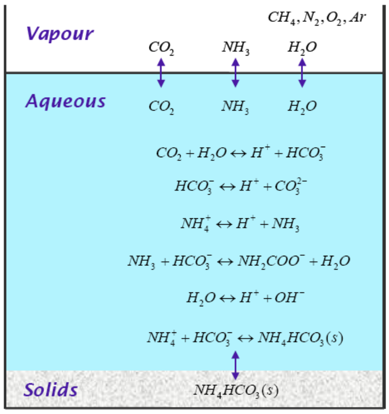


Figure 60. Aqueous Ammonia chemical equilibrium reactions[49]

There are three possible phases present in the NH_3-CO_2-H_2O system: gas, liquid and solid. Jilvero et al[50]. did a literature survey of reboiler heat duty both for equilibrium simulation models and pilot plants. Equilibrium simulations showed that the reboiler heat duty ranges from 2 to 4 GJ/ton CO_2 depending on the ammonia concentration. The pilot plant data was only recorded at low ammonia concentrations (2 to 5 wt%) and the lowest registered regeneration heat was around 4 GJ/ton CO_2[51], which agrees with the equilibrium simulation models under similar conditions. Aqueous ammonia has several advantages over MEA. High CO_2 loading capacity, low equipment corrosion risk, low absorbent degradation and a low energy requirement for regeneration. Some literature suggests that NH_3 solutions have 25-60% higher CO_2 capacity than MEA and thus lower regeneration energy. NH_3 solution has also lower solvent degradation, which means lower solvent make-up cost. The degradation of NH_3 is in the form of ammonium sulfate or ammonium nitrate; both are saleable fertilizers. However, there are problems with using ammonia as a solvent. The volatility of ammonia is a major environmental problem since it has a tendency to escape the absorber column to the atmosphere with the flue gas. Procedures like cooling down the exit flue gas or adding water or acid wash could reduce these levels but would add additional costs and complications.

[50] Lioliou,M., 2011," Modelling of liquid-vapor-solid equilibria in the NH_3-CO_2 –H_2O system", 1st Post combustion capture conference.
[51] Jilvero, H., 2011," Heat of desorption of CO_2 in aqueous ammonia", 1st Post combustion capture conference.

On the other hand other literature sources are less positive with using ammonia as solvent. They suggested that the energy requirements for CO_2 regeneration are similar as those for MEA and the rate of absorption for the NH_3 process would be much slower than MEA, which may require the absorber to be three times the height of an MEA absorber.
The Chilled Ammonia optimization process is more complex than for the MEA process. The low-temperature absorption requires cooling down the absorbent, which requires using a chilling machine.
There are other solvents that have been investigated where the focus has been on developing energy efficient solvents with less environmental impacts. Potassium carbonate solution and amino-acid salts are example of these solvents.

App. II.9 Piperazine/Potassium Carbonate Solution

Researchers investigated the possibility of using an aqueous piperazine/potassium carbonate (K_2CO_3/PZ)[52] solution as a replacement for MEA in a chemical absorption system. The research showed that a piperazine/potassium carbonate solution is better than an MEA solution in terms of heat of absorption and rate of absorption. K_2CO_3/PZ has also a lower heat of regeneration than the MEA in ranges between 25 to 49%. Furthermore K_2CO_3/PZ solution has a rate of absorption that is 1 to 5 times faster than MEA, allowing for a smaller absorption column. K_2CO_3/PZ solution has been shown to have a low volatility, reducing the environmental emissions problem. However K_2CO_3/PZ is more expensive than MEA and ammonia solvents.

App. II.10 Amino-acid Salts

Some amino-acid salts were found to absorb CO_2 faster than MEA and can potentially reduce the required energy for regeneration. Other amino-acid salts have good resistance towards oxidative degradation and low volatilities. One distinct advantage that some amino-acid salts have is that they have a particularly high CO_2 capacity, which could allow for considerably smaller equipment sizes. The amino-acid salts have a complicating factor in that they form precipitates during absorption, which requires that the equipment is able to handle slurries. The following table gives a comparison of physical properties for MEA solvent, aqueous ammonia and K_2CO_3/PZ[50].

[52] Liang-shih fan, 2010, Chemical looping systems for fossil energy conversions”, ISBN 978–0-470–87252–9

Performance of Alternative Solvents Relative to MEA						
	Heat of Regeneration	CO_2 Absorption Rate	CO_2Absorption Capacity	Resistance to Degradation/ Impurities	Corros-ion	Volatil-ity
Aqueous NH_3[53]	+ ?	-	+	+	+	-
K_2CO_3/PZ	+	+	=	?	?	=

Performance of Alternative Solvents Relative to MEA

App. II.11 Process Development

A typical coal-fired power plant generates flue gas with composition of 12 % wt CO_2 and 8%wt O_2 while gas turbine combined cycles (GTCC) with similar power capacity generate flue gas with composition 3%wt CO_2 and 15 % wt O_2. These differences in flue gas properties result in changes in solvent performance. In the case of GTCC, it is favorable with solvents with good absorption performance at lean CO_2 loadings, and low volatility.

App. II.12 Amine Process (AP)

The Amine Process (AP) is suitable for CO_2 separation from flue gases with CO_2 concentration of 10–15%. The AP capture system is capable of separating up to 100 of the CO_2 in the flue gas however 90% capturing has been found economical. The AP is based on the chemistry of the Amine-CO_2-H_2O system and the ability of the amine solution to absorb CO_2 at low temperatures between 35ºC and 40ºC. CO_2 reacts with the amine solution in the absorption column forming carbamate and this removes CO_2 from the flue gas. CO_2 is absorbed in an amine solution at a temperature around 40ºC and at atmospheric pressure. The process consists of two main blocks, the CO_2 absorber and CO_2 stripper, figure 61.

CO_2 Absorber: The flue gas enters the CO_2 absorber with a temperature around 40ºC and atmospheric pressure. As the gas moves upwards in the absorber column counter-currently to the concentrated amine solution, CO_2 is absorbed to the solution. In order to increase the richness of the amine solution, recirculation and intercooling systems are used in the absorber. In recirculation, a part of the rich amine solution from the bottom of the absorber is cooled and recycled back to the middle of the absorber. Absorber intercooling system works by taking a part of amine solution from the middle of the absorber and cool it down before returning it back to the column. The intercooling system improves the equilibrium of the amine/CO_2 absorption and increases the richness of the amine solution. Recirculation and intercooling systems improve the energy efficiency of the regeneration system. However these solutions have to be weighed against increasing the investment costs. The flue gas leaving the top of the absorber containing mainly nitrogen, oxygen, water and a low concentration of CO_2, is emitted to the atmosphere.

[53] The chilled ammonia process requires a chiller machine in order to cool down the solvent before the absorber. The chiller requirement depends on the cooling water temperature.

Stripper: The CO_2-rich solution exiting the absorber is heated up in the rich/lean heat exchangers and sent to the regenerator in multiple feed points. The CO_2-rich solution is then regenerated in the CO_2 stripper under a high temperature (100–150°C). The rich-CO_2 solution decomposes and as the result the amine solvent is regenerated and produces a concentrated CO_2 stream. The solvent regeneration requires a large amount of high temperature steam to strip the CO_2. This reduces the overall power output and, hence, the efficiency of the power plant. The CO_2 stream and water vapor exit the top of the CO_2 stripper where water is condensed in the CO_2 stripper condenser. The lean-CO_2 solution is pumped through the rich/lean heat exchangers and sent back to the CO_2 absorber for further CO_2 absorption.

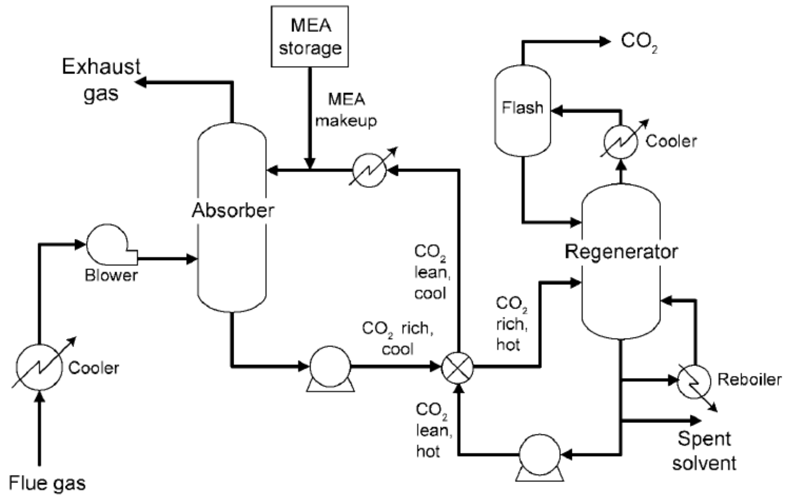


Figure 61. Amine Process (AP)[54]

App. II.13 Chilled ammonia process (CAP)

The chilled ammonia process is similar to the amine process. The CAP process consists of an absorber, which absorbs CO_2 from flue gas using ammonia solution, and a CO_2 stripper. In order to increase the absorption rate, the ammonia solution has to be cooled down to 15°C after CO_2 absorption. Part of the solution is recirculated back to the absorber and the other is sent to the rich/lean heat exchanger.

Absorber: The hot flue gases are emitted from the power plant with temperature around 70°C[55]. Flue gases are cooled down to condense the water vapor and some other gases like SO_2 and NO_x. The flue gas enters the absorber after cooling[56] with a temperature of around 10°C and reacts with the ammonia solvent forming mainly ammonia bicarbonate (NH_4HCO_3). Almost 90% of the CO_2 can be absorbed from flue gases. The chemical reaction between ammonia and carbon dioxide will generate heat that is

[54] Liang-shih fan, 2010, Chemical looping systems for fossil energy conversions", ISBN 978–0-470–87252–9

[55] Flue gases temperature could vary from coal to gas power plants. The temperature is used to be around 70-100°C.

[56] The flue gases is cooled down with cooled water

cooled down by the cooling water system before the solvent is circulated back to the absorber.
CO_2 stripper: The rich-CO_2 solution is pumped to the CO_2 stripper through the lean/rich heat exchanger, where the rich-CO_2 solution is heated up before entering the CO_2 stripper. In the CO_2 stripper, the rich-CO_2 solution is heated up to around 150°C. The CO_2 vaporizes from the liquid and moves toward the top of the regenerator. The remaining CO_2-lean solution is fed back from the bottom of the CO_2 stripper to the absorber through the heat exchanger. The CO_2 stripper operates under high pressure, around 20 bar, which is an advantages because this reduces the CO_2 compression work.
NH_3 stripper: In order to reuse the ammonia in the process, the CAP is supplied with an NH_3 stripper. The purpose of the NH_3 stripper is to regain the ammonia from the NH_3-rich solution to the process by stripping. The energy required for ammonia stripping is extracted from the power plant; however, it is much smaller than the regeneration energy required in the CO_2 stripper in the MEA process. Because of the volatility of the ammonia, the chilled ammonia process is supplied with an ammonia washer before the flue gas is emitted to the atmosphere. The purpose of the ammonia washer is to scrub the flue gas coming from the NH_3 absorber from the remained ammonia by adding sulphuric acid (H_2SO_4). The remaining NH_3 reacts with the sulphuric acid and form ammonium sulphate $(NH_4)_2SO_4$ and the cleaned flue gas is sent to the atmosphere.

App. II.14 Process Integration

The CO_2 capture process requires heat energy that is used in the regeneration process. In addition, the CO_2 capture process requires some electrical energy to power fans to support the gas flow though the absorption column and to power solvent pumps. Heat is usually taken from the power plant in the form of extracted steam from the steam turbine. The extracted steam doesn't contribute in producing electricity and thus it is counted as a parasitic loss and reduces the net power generated from the power plant. The overall electricity output penalty of carbon capture (EOP) is calculated in GJ/ton CO_2 and is defined as the sum of the contribution of steam extraction, CO_2 compression and auxiliary power. The cost of electricity (COE) produced from the power plant will also increases due to the added capital and operating expenses incurred by the CO_2 capture and storage. Thus, designing the carbon capture plant in an efficient way is important to minimize the energy penalty and COE. The energy penalty of CO_2 capture reduces the overall plant power output by 21% – 42% and the efficiency by 10-15%[57,58]. There are many methods to reduce the energy penalty; however, the most important methods are:

- Improving the design and operation of the CO_2 capture process.
- Improving the integration of the CO_2 capture process with the power plant.

[57] The energy penalty depends on the CO_2 concentration in the gas. For coal fired plants the efficiency loss is higher because higher percentage of CO_2 has to be captured.
[58] Harkin, T., 2011," Power/cost trade-offs associated with solvent based PCC", 1st Post combustion capture conference.

- Using the Heat Exchanger Network[59]
- Optimizing the power plant and CO_2 capture operating variables.

Most current designs include steam extraction from the steam turbine, which is the most important process integration. Optimizing the steam integration between the power plant and the CO_2 capture plant reduces the energy penalty. Such integration has been studied by Jonshagen et al[60].
Optimization of the CO_2 capture process includes sensitivity analysis for different key parameters in the process such as regeneration temperature and pressure and column packing types. Many other modifications have been proposed to reduce the energy consumption or equipment costs. For example power recovery turbines are sometimes proposed to capture some of the energy available when the pressure is reduced on the rich solution. Other modifications are aimed at reducing the absorber column cost by using several lean solution feed points.
The Scottish Center for Carbon Storage[61] has studied regeneration pressure and temperature effect on thermal energy of generation. At high CO_2 capture levels, the solvent capacity needs to be increased to capture the additional amount of CO_2 by increasing the solvent flow rate. This requires increasing the solvent circulating pump effect and thus the EOP increases. A higher amount of stripping is also necessary to reduce the solvent lean loading. Consequently, the optimum overall EOP occurs at lower solvent lean loading. It should be noted this has the effect of reducing the partial pressure of CO_2 in the stripper, and increase the EOP associated with CO_2 compression.
Plaza et al.[62] from university of Texas at Austin have studied process integration of the concentrated piperazine solvent using an intercooled (40°C) absorber and a high temperature (150°C) two-stage flash stripper configuration, figure 62. Concentrated piperazine (PZ) has almost double the capacity of MEA, while displaying similar volatility and higher resistance to oxidative and thermal degradation. Plaza et al. suggested that intercooling improves the absorber CO_2 removal by 10%.

[59] Heat Exchanger Network is method for flow stream optimizing in case of several heat exchangers
[60] See chapter 8.3
[61] Lucquiaudl, M., 2011,“ Future-proofing coal plants with post-combustion capture against technology development”, 1st Post combustion capture conference.
[62] Plaza, J., 2011,” Modeling CO_2 capture using concentrated PZ”, 1st Post combustion capture conference.

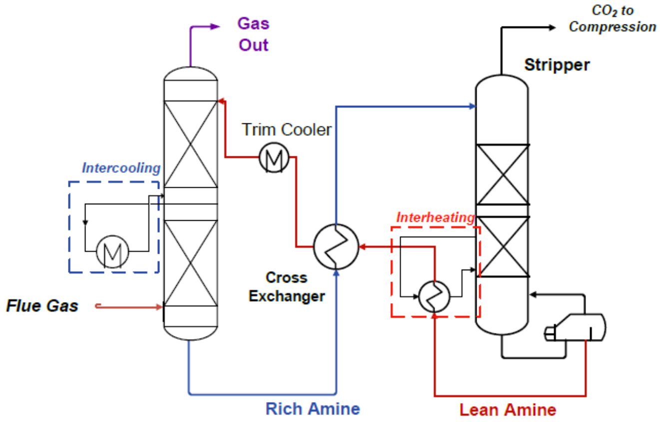


Figure 62. Piperazine process[61]

EDF Energy estimated the efficiency loss because of CO_2 capture process into 11-15% and the cost of CO_2 separation between 40-60 €/tonCO_2[63]. EDF aimed to reduce the efficiency loss through process optimization into 5% and the cost of separation into 20-30 €/tonCO_2[64].

The market electricity prices have consequences on carbon capture plant operating conditions and design. Converting the overall electricity output penalty of capture (EOP) into a loss of potential revenue and balancing this against power plants revenue is suggested sometimes to operate the post-combustion capture unit beyond its design capture level or at lower levels or even not at all. The operation of a power plant is influenced by grid balancing, rapid changes in the electricity market and variations in CO2 and fuel market prices. Thus, flexible power plant operation is necessary. This could be achieved by bypassing the CO_2 capture plant and varying carbon capture rate. The University of Edinburgh studied[65] the sensibility of the electrical and carbon price market. The electricity market daily price fluctuation is presented in figure 63 while the daily fluctuation in carbon price is presented figure 64.

[63] Coal fired power plant, Le Moullec, Y., Kanniche, M, 2011,"Optimization of MEA based post combustion CO2 capture process: Flowsheeting and energetic integration", 1st Post Combustion Capture Conference.
[64] Moullec, Y., 2011," Optimization of MEA based post-combustion CO_2 capture process", 1st Post combustion capture conference.
[65] Ereey,O., Lucquiaud, M., Chalmers, H.,Gibbins, J., 2011, "Effect of varying electricity and carbon prices on operation and capture levels in flexible post-combustion CCS power plants", 1st Post Combustion Capture Conference.

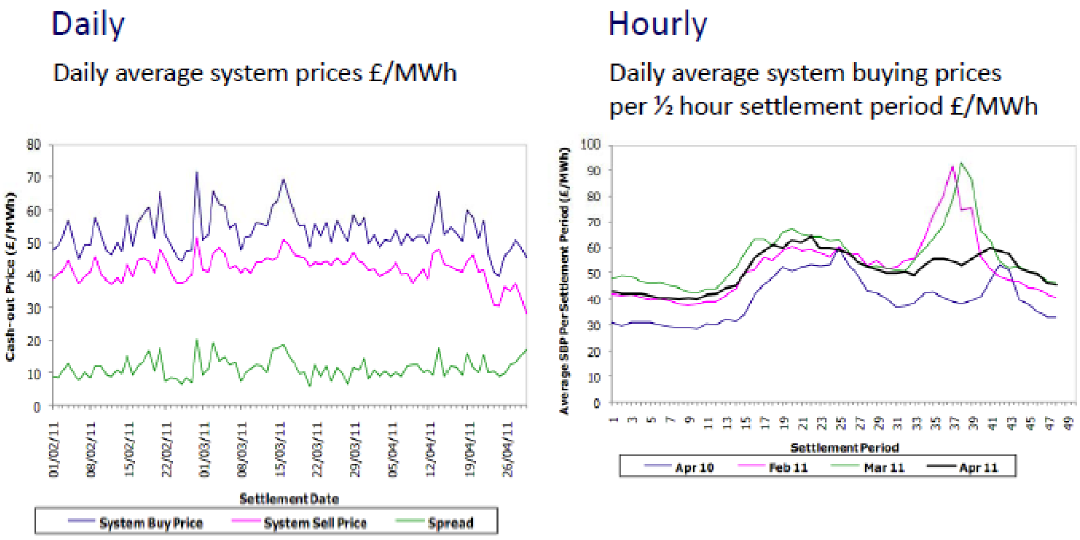


Figure 63. The electricity market price[66]

Figure 64. Daily fluctations in carbon price[67]

In order to miximaize the power plant revence two methods has been suggested:

- Steam cycles with flexible steam turbine operations: Capacity to reduce steam extraction thereby reducing the total carbon captured and increasing the energy output for a maintained period.
- Bypass: Assumption of rapid "Shutting down" of capture wing liberating extra MW capacity.

[66] This study of Great Britian market, Elexon trading operations report, 2012, ICE Futures Europe.
[67] https://www.theice.com/publicdocs/futures/ICE_Monthly_Utility_Report.pdf.

App. II.15 Oxy–fuel gas turbine cycles

This chapter presents four of most promising oxy-fuel gas turbines cycles. These innovative cycles work by replacing air with oxygen in the combustion. The product gases contains thus mainly of carbon dioxide and water. The water separates from the gases by condensation. The remaining carbon dioxide then is compressed prior to transportation and storage. Oxy gas turbine cycles could be called zero emissions cycles because SO_2 and NO_x do not present in the combustion products. However these cycles requires large amount of energy in order to separate oxygen from air with high purity around 95%. The presented oxy-gas turbines cycles are Water cycle, Matiant cycle, AZEP cycle, Graz cycle and SCOC-CC cycle.

App. II.16 Water Cycle[68]:

Water Cycle or CES (Clean Energy Systems Inc.) is a cycle concept based on Rankin cycle. The working fluid in this cycle is mostly water (around 90 % wt). The fuel (Natural gas) is compressed and preheated before the high-pressure combustion occurs in the high pressure (HP) combustor chamber at 80 bar. The oxygen is supplied to the HP combustion chamber from the ASU. In order to control the high combustion temperature, hot water is injected to the combustion chamber to controls the combustor exit temperature. The exhaust gases (mainly water and small fraction CO_2) are expanded in the high pressure turbine to exit temperature around 650°C. The gases are utilized again in the low pressure (LP) combustion chamber where more oxygen and fuel is added at 8 bar. The LP combustor exhaust gases temperature is around 1300°C and expands afterwards in the low pressure turbine to 450°C. The exhaust gases from the LP turbine are cooled down then by fuel preheating and water heating (recuperator). The exhaust gases are emitted then to the condenser where the water is separated by condensation. The gaseous CO_2 is compressed afterward to 1 bar. The liquid water resulted from the condenser is recycled back to the HP combustor, after compression and heating in the recuperator. The HP turbine inlet temperature is 900°C, which represents a very advanced steam turbine technology based on an uncooled, high-pressure turbine. The LP turbine in which the inlet temperature is around 1300°C represents on the other hand a gas turbine technology based on a cooled turbine. The flow chart of the water cycle and some key data are presented in figure 65. Bolland et al[67] presented the results of 80MW water cycle. The net power output was 56MW while the net efficiency was 39%.

[68] Bolland, O., Kvamsdal, H., and Boden, J., 2005, "A Comparison of the Efficiencies of the Oxy-Fuel Power Cycles Water-Cycle, GRAZ-Cycle and Matiant-Cycle," Carbon Dioxide Capture for Storage in Deep Geologic Formations-Results from the CO2 Capture Project, Capture and Separation of Carbon Dioxide from Combustion Sources, 1, pp. 499-511.

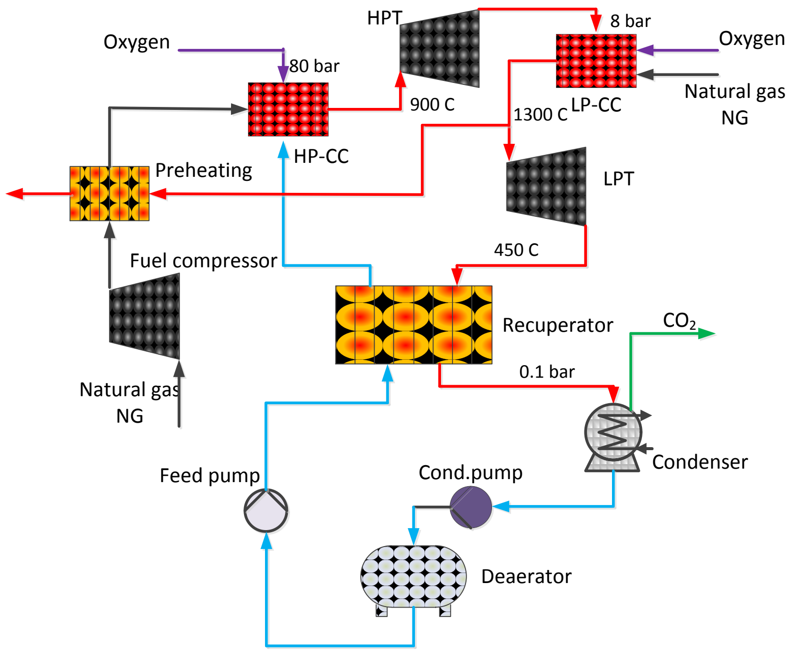


Figure 65. Water cycle.

App. II.17 Matiant Cycle[69]:

This cycle is named after the constructor of the cycle (*Mat*hieu and *Iant*ovski). The cycle is Brayton like cycle. The working fluid is mostly CO_2 and water. The oxy combustion takes place in the high pressure combustion chamber at which compressed fuel (naturel gas) and a mixture of CO_2 and oxygen is injected. The combustion outlet temperature is 1300°C and the gases expanded afterword in the high pressure turbine. The gases are reheated up in the second combustion chamber where other fraction of fuel and oxygen is added. The resulted gases expand afterwards in the low pressure turbine. The gases are sent then to the recuperator where its exchange heat with the working fluid. At the outlet of the recuperator the gases are cooled down in a cooler where the water is condensed. The remaining gas (maily CO_2) is compressed in the compression train before transportation. The working fluid in a liquid phase is then pumped to a higher pressure and heated up in the recuperator. It expands then through the high pressure turbine before the reheating in the recuperator. The working fluid is sent next to the high pressure combustion chamber. The flow chart of the Matiant cycle and some key data are presented in figure 66. Matiant cycle accordeing to Bolland et al[67] has high energy consumption and thus other cycles have been developed from the Matiant cycle. These cycles are steam injected-Matiant cycle (SI-Matiant cycle), Combied-Matiant cycle (CC- Matiant cycle) and Fuel Cell-Matiant cycle (FC- Matiant cycle).

[69] Mathieu, P., 1998, " Presentation of an innovative zero-emission cycle for mitigating the global climate change", Applied thermodynamics, pp.21-30.

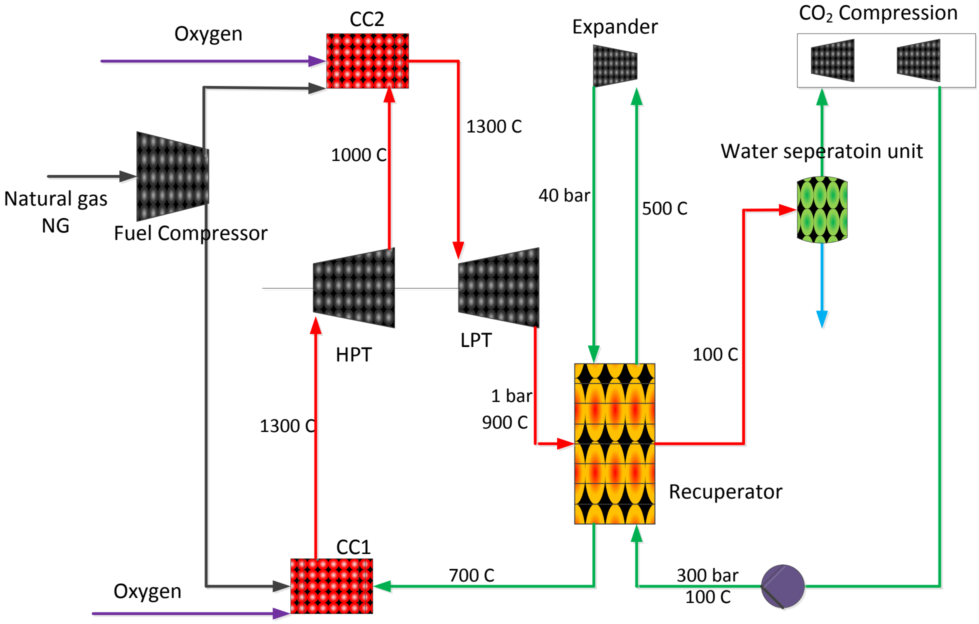


Figure 65. Matiant cycle.

App. II.18 Graz Cycle:

The graz cycle is based on the Matiant cycle. The combustion chamber is fed with fuel (natural gas), oxygen from the ASU and the working fluid (CO_2 and mostly steam 70 % wt). The combustion takes place in a pressure around 45 bar and the combustion outlet temperature is around 1400°C. The gases are expanded then in the high pressure (HP) turbine. The gases from the exit of the turbine are emitted to the Heat recovery steam generator (HRSG). In the HRSG the high pressure steam is produced and expanded in the high pressure (HP) steam turbine to the combustion chamber pressure. The gases emitted from the HRSG are split at which part is recirculated back to the compressor and the remaining is sent to a lower pressure steam cycle. The gases (mainly CO_2) are compressed in the compressor train to a higher pressure. In the lower cycle more steam are produced and expanded further in the low pressure (LP) turbine to the condenser pressure. The water collected from the condenser in the liquid phase is preheated in the CO_2 compression intercoolers before it's pressurized for steam generation. The flow chart of the Graz cycle and some key data are presented in figure 66.

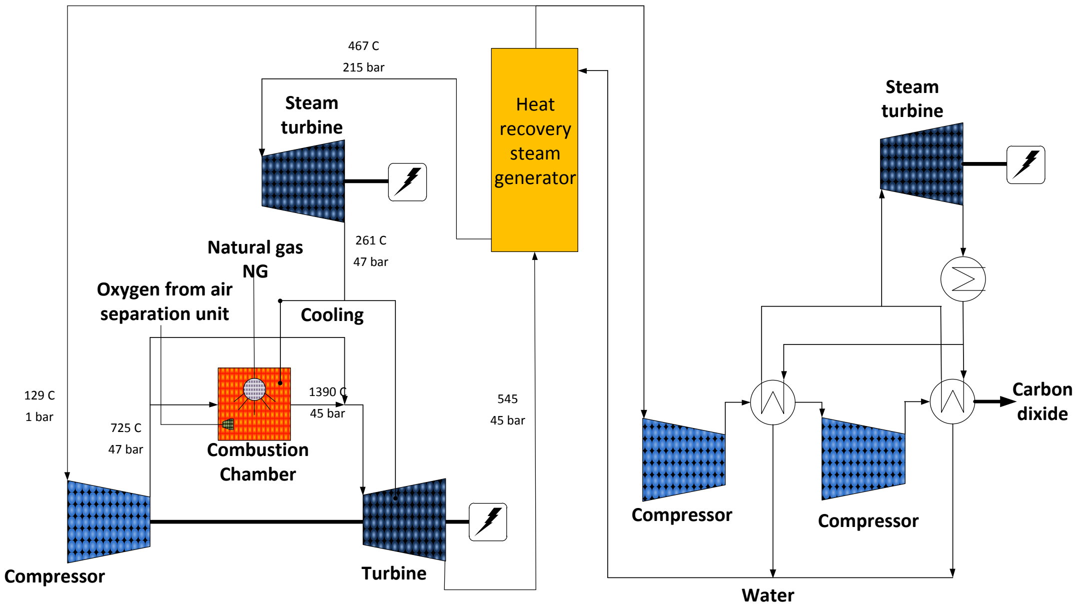


Figure 66. Graz cycle.

App. II.19 SCOC-CC cycle:

The SCOC-CC (Semi-closed oxy-fuel combustion combined-cycle)[70] can be decomposed into five main units, the gas turbine (Top cycle), the heat recovery steam generator (HRSG) and steam turbine (Bottom cycle), the flue-gas condenser, the ASU, and the CO_2 compression train. A schematic overview of the SCOC-CC is presented in figure 67. The fuel (natural gas) is combusted with oxygen under conditions close to stoichiometric. Oxygen of 95% purity is produced in the ASU. The combustion products are thus mainly carbon dioxide and water. The water is subsequently removed by cooling in the flue-gas condenser, where the flue gas leaving the flue-gas condenser consists mainly of CO_2 at a temperature around 20°C and a pressure close to atmospheric. The rich flue gas containing CO_2 is then divided into two streams. Approximately 93% is recirculated back to the gas turbine, while the remaining rich flue gas is compressed and liquefied to enable transport and storage in the dense phase. The gas turbine exhaust gas temperature is 620C of about 620°C which is optimal for the steam cycle. The SCOC-CC has been studied in detail in several papers by Sammak et al. Sammak et al[71] has presented design for single and twin shaft oxy-fuel gas turbine. The rotational speed of the designed single-shaft gas turbine was set to 5200 rpm. The designed turbine had four stages, while the compressor had 18 stages. The rotational speed of the twin-shaft gas generator was set to 7200 rpm, while

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[70] Sammak, M., Jonshagen, K., Thern, M., Genrup, M., Thorbergsson, E., Grönstedt, T., and Dahlquist, A., 2011, "Conceptual Design of A Mid-Sized Semi-Closed Oxy-Fuel Combustion Combined Cycle," ASME Turbo Expo 2011.

[71] Sammak, M., Genrup, M., Thorbergsson, E., Grönstedt, T.,"Single and Twin-Shaft Oxy-Fuel Gas Turbine Design in a Mid-Size Semi- Closed Oxy-Fuel Combustion Combined Cycle" . ASME Turbo Expo 2012.

that of the power turbine was set to 4500 rpm. Twin-shaft turbine designed with five turbine stages to maintain the exit Mach number around 0.5. The twin-shaft turbine required a lower exit Mach number to maintain reasonable diffuser performance. The compressor turbine was designed with two stages while the power turbine had three stages. The compressor was designed with 14 stages. The choice of a twin-shaft gas turbine can be motivated by the smaller compressor size and the advantage of greater flexibility in operation, mainly in off-design mode. However, the advantages of a twin-shaft design must be weighed against the inherent simplicity and low cost of the simple single-shaft design. The following table gives a compression between SCOC-CC and graz cycle for power plant with net effect around 100 MW.[72]

	Unit	SCOC-CC	Graz Cycle
Compressor mass flow	kg/s	190	62
Compressor pressure ratio	-	37	44.7
Combustor outlet temp.	°C	1400	1400
Gas turbine power	MW	86	105
Gas turbine power, Compressor	MW	67	62
Gas turbine power, Turbine	MW	155	169
Total heat input	MW	230	199
Steam turbine power	MW	48	20
Pumps power	MW	0.5	0.8
Gross power output	MW	134	125
Gross efficiency	%	58	63
O_2 generation + compression	MW	23	20
CO_2 compression	MW	5	4.3
Net power output	MW	106	96
Net efficiency	%	46	49

[72] Thorbergsson, E., Grönstedt, T.,Sammak, M., Genrup, M., "A Comparative Analysis of Two Competing Mid-Size Oxy-Fuel Combustion Cycles" . ASME Turbo Expo 2012.

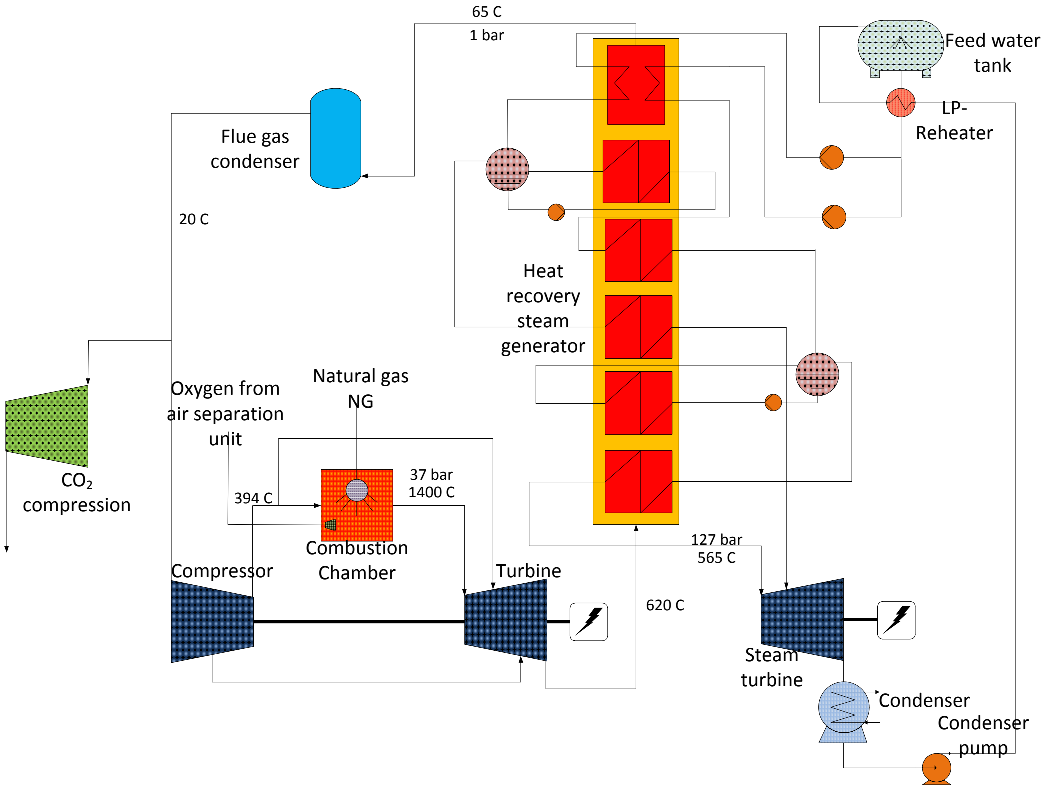

Figure 67. SCOC-CC cycle.

App. II.20 AZEP cycle:

AZEP (Advanced Zero Emission Power cycle) works with replacing the gas turbine combustion chamber with MCM (Mixed Conducting Memberan) reactor which produces pure oxygen from compressed air[73]. The MCM rector is integrated thus to a conventional gas turbine combined cycle. The traditional type of AZEP cycle which is called 100% AZEP has only MCM-reactor but MCM can be supported with an afterburner. Air is compressed in the gas turbine compressor and is heated then in the MCM-reactor. Due to MCM material and reactor design limitations the MCM-reactor outlet temperature has been limited to 1200°C. A part of the oxygen almost 50% is transferred through the membrane modules where oxygen is transported from the air to the sweep gas side. The oxygen containing sweep gas is then reacted with natural gas injected in the catalytic combustion chamber to generate heat. The heat contained in the bleed gas is recovered in a separate CO_2/H_2O HRSG, providing extra steam for the steam cycle and preheating of the natural gas fuel. After the HRSG the water is condensed and the remaining CO_2 is compressed from the MCM-reactor pressure at around 20 bar to delivery

[73] Möller,B., Torisson, T., Assadi, M., Sundkvist, S., Sjöden, M., Klang, Å., Åsen, K., Wilhelmsen, K., 2006, " AZEP gas turbine combined cycle power plants-Thermo-economic analysis", International journal of thermodynamic, pp.21-28.

pressure at 100 bar and liquefied. The flow chart of the AZEP cycle is presented in figure 68.

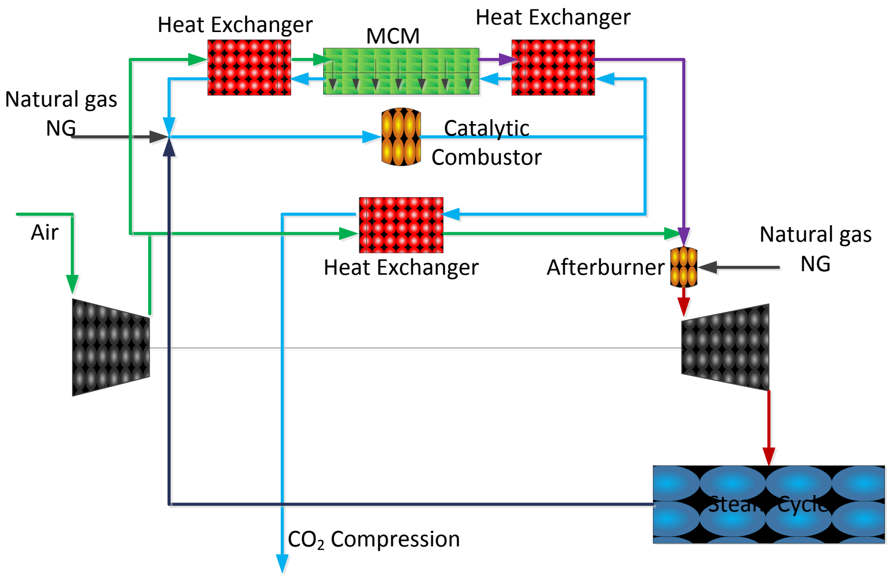

Figure 68. AZEP cycle.

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