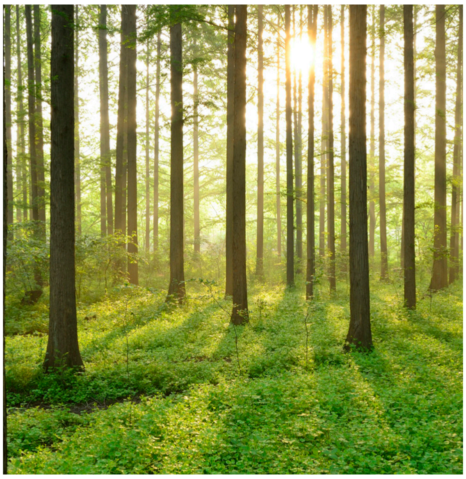
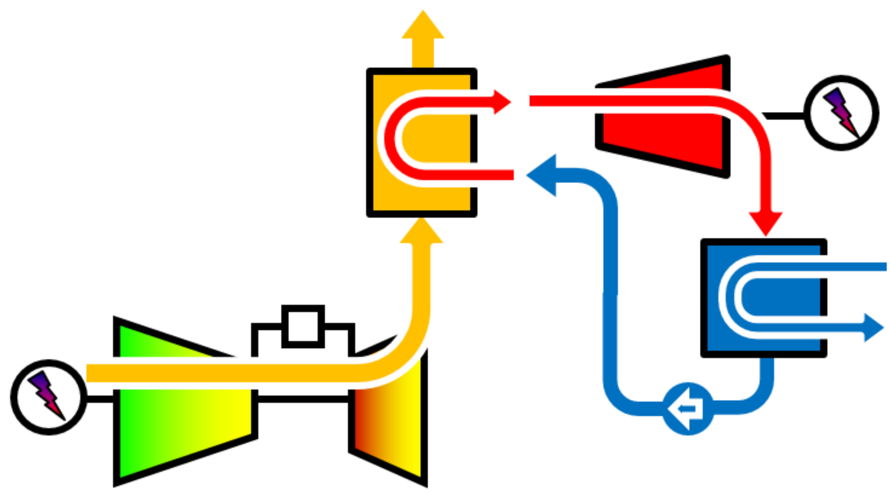
GAS TURBINE DEVELOPMENTS 2012-2015

REPORT 2016:272



UPPFÖLJNING AV
GASTURBINTEKNIKEN



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GAS TURBINE DEVELOPMENTS 2012-2015

Ny gasturbinteknik 2012-2015

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Förord

Projektet Uppföljning av gasturbintekniken har bedrivits ett antal år och resulterat i årliga rapporter. Det här är slutrapporten från den senaste treårs-etappen 2012 – 2015. Föregående etapp slutrapporterades i Elforskrapport 12:27.

Projektets mål har varit säkerställa beställarkompetens avseende moderna gasturbinkombianläggningar hos projektets deltagande parterna genom

- höja kompetensen hos kraftföretagen för att få bättre ekonomi och miljöprestanda samt ökad tillgänglighet hos gasturbinbaserade anläggningar.
- följa utvecklingen speciellt vad gäller, kunskaper om investeringskostnader, drifttillgänglighet, underhållskostnader och bränsleflexibilitet för moderna gaskombianläggningar.
- följa utvecklingen av nya framtida gasturbinkoncept.

Den senaste etappen har finansierats av E.ON Värmekraft Sverige AB, Göteborg Energi AB och Öresundskraft AB. Projektet har genomförts av Magnus Genrup och Marcus Thern, Lunds Universitet och Energiforsk har varit sammanhållande.

Projektet har följts av en styrgrupp med följande medlemmar: Fredrik Olsson och Matilda Lindroth, E.ON Värmekraft, Martin Rokka och Thomas Johnson, Göteborg Energi, Fredrik Joelsson, Öresundskraft och Bertil Wahlund, Energiforsk. Energiforsk tackar styrgruppen för värdefulla insatser i projektet.

Mars 2016

Bertil Wahlund
Energiforsk AB

Sammanfattning

Under projektets senare år har både verkningsgraden och flexibiliteten för kombianläggningar blivit avsevärt bättre. Idag erbjuder alla större tillverkare verkningsrader uppemot 61 procent. En luftkyld kombi kunde tidigare nå maximalt runt 58-59 procents verkningsgrad medan ångkylda låg runt 60 procent. Vid EONs kraftverk Ulrich Hartmann i Irsching, utanför Ingolstadt har TÜV certifierat verkningsgraden på Siemens SGT5-8000H anläggningen till 60.75 procent.

Förr ansåg 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. General Electrics[1], Mitsubishis och Siemens gaskombianläggningar klarar idag en varmstart på mindre än 30 minuter, upp till full last. Det är betydligt snabbare än vad ångkylda maskiner klarar. Luftkylda 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 gasturbiner för kombiprocesser och nått 61-62 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 (2012[2]) 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. Man bör 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 blir 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.

[1] Inklusive Alstoms flotta

[2] Inga data efter 2012

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 € (13.8×109 €). 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/H2 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.

Summary

The last six 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 Kraftwerke Ulrich Hartmann (formerly Irsching 4) site outside Ingolstadt. The old paradigm that high performance meant advanced steam-cooled gas turbines and slow started bottoming cycles has definitely proven false. General Electric[3] Mitsubishi and Siemens 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 61-62 percent efficiency. Steam cooling will most likely still 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 (2012) 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.

[3] Includes the Alstom portfolio

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 on the order of 47,000 units and the total value of the gas turbine aftermarket was 2009 13.8 B€ (13.8×109 €). 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/H2. 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.

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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 gas generators, 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 E.ON Öresundsverket in Malmö. The biggest single engine fleet is operated by the Swedish Navy with 20 four megawatt Vericor TF50 units onboard the Kockums Visby-class corvettes.

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

1.2 PROJECT EXECUTION

The project runs 2012-2015 at Lund University, Department of Energy Sciences. Project manager and responsible for the technical content is Professor Magnus Genrup and Associate Professor Marcus Thern.

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 77 different plants.

1.4 COMMON ABBREVIATIONS AND NOTATIONS

AN2 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[4]
CSA	Contractual Service Agreement
DCF	Discounted Cash Flow
DLE	Dry Low Emission
DS	Directional Solidified
EGT	Exhaust Gas Temperature
EIS	Engine/Entry in Service
FGR	Flue Gas Recirculation
FN	Turbine Flow Number or capacity (cf. a standard convergent nozzle)
FOB	Free/Freight On-Board
HARP	Heater Above Reheat Point or header/bundle set-up
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

[4] Typically used synonymously with "firing"

IPT	Intermediate Pressure Turbine
IRR	Internal Rate of Return
MEA	Monoethanolamine (C2H7NO)
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 for industrial units)
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 authors 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. The used exchange rate between € and USD is 0.8987, which is the average during 2015.

3 General trends

The general role of the combined cycle has changed 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 has created 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. The installed wind capacity in Europe is 133,968 MW and Germany alone has 39,165 MW (24,807 units). The worldwide installed capacity is 371,191 MW – hence, approximately 1/3 of the total installed wind capacity in Europe. The level of installed PV-solar is today 38.7 GW in Germany alone. The all-time-high production was approximately 22 GW in 2012.

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 Swedish situation, today, is quite different where only three cities have gas turbine based district heating production during the winter. Back-up capacity for renewables is effectively covered by the very large hydro capacity in northern Sweden rather than gas turbines.

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[5] 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

In 2011, the European Commission (EC) published that the exploitable resource of shale gas to be approximately 16 trillion cubic meters (i.e. 16 tera- or 10^{12} cubic meters). The total world exploitable resource is approximately 200 trillion cubic meters which gives a number of approximately eight percent in Europe. Based on scenarios of the IEA and

[5] Condition-based maintenance

EC, the estimate for 2035 is that the amount of shale gas will be about 77 billion cubic meters. This turns into approximately 11 percent of the total EU natural gas production.

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 improved 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 start of the recent regression in the world economy.

No OEM's besides General Electric, Rolls-Royce[6] and Pratt & Whitney[7] (PW Power Systems) 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[8] 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 within 1,400…1,500 °C. Higher frame firing levels will force the steam turbines to 600 °C (or even higher) 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 outside Japan and South Korea due to the necessity of steam cooling.
- Operational flexibility requirements with little or no RAMD-S[9] impact. Most OEMs are capable of 30 min hot-start and steep (35-60 MW/minute) ramp-rates.
- High-temperature engines will rely upon thermal barriers (TBCs). This feature is probably still 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.

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[6] Now part of Siemens
[7] Now part of Mitsubishi
[8] The word firing is used synonymously for combustor outlet temperature (COT) throughout the text.
[9] Reliability, Availability, Maintainability, Durability and Safety

- 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 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 associated efficiency potential with the fourth stage because the overall 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 General Electric. There are several of MHI G-class units that have been re-built into air-cooled units for addressing flexibility issues. The old paradigm that steam cooling was a requirement for 60 percent efficiency has definitely proven false since once introduced. The latest H-class by GE does not employ steam cooling and is fully air-cooled and offers 61(+) percent efficiency.

- Specific flow has continued to increase and approaches aero-engine technology. The latest General Electric 50 Hz engine has a flow of 980 kg/s@3000 rpm. The 50 Hz version of the MHI J-class will most likely be approximately 860 kg/s. The absolute maximum of today is around 1000 kg/s@3000 rpm but this has only previously been achieved with multi-spools. The 980-figure is, indeed, a leapfrog from the conventional level. There are a couple of conflicting requirement and one cannot look at the compressor alone because the turbine stress level is an important issue. This manifests itself as a trade-off between stresses in the turbine and the exit velocity that is considered as a loss. A clear trend has also 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 tip-timing (on an individual blade level) vibration measurements and associated protection system – in situ.

Today, all major OEMs are capable of offering 61 percent and higher. Siemens has a TÜV-certified efficiency of 60.75 percent – measured/demonstrated in-situ at the Irsching 4 site[10]. 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

[10] The Irsching 4 plant has been renamed to Kraftwerke Ulrich Hartmann in 2011.

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 start-up 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. The cure is to start the plant with lower admission temperature by utilizing over-sized spray coolers. 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 (i.e. hot starts). 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[11]. Hence, the once-through technology should generally not be superior in terms of flexibility when it comes to daily cycling.

Some of the features in the discussion are presented in the figure below.

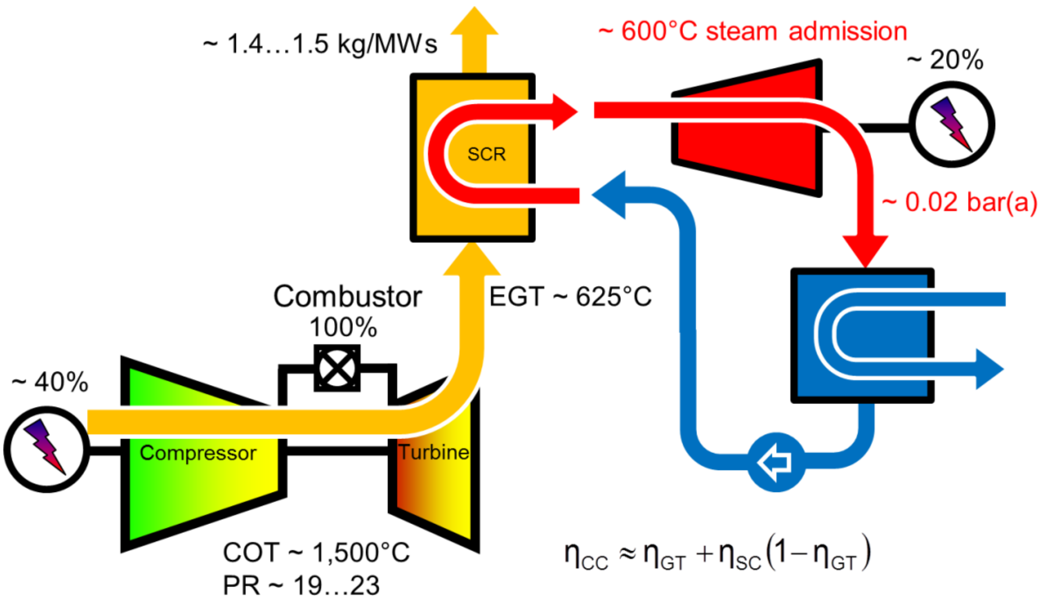


Figure 3-1. High-performing GTCC features – the path to 60+ efficiency

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. The design offers considerably lower part count and cooling consumption. No information has been published related to the production engine platform or market introduction.

3.1.1 Mitsubishi

Mitsubishi offers an array of classes of gas turbines, namely F, G and J. The 60Hz M501J engine deliver 327 MW in simple cycle or 470 MW efficiency in combined cycle. The 50 Hz version has an output of 470 MW and 680 MW in simple- and combined cycle, respectively. The efficiency is 61.5 and 61.7 for the 60 and 50 Hz versions, respectively – reflecting the scale of size on efficiency for the larger 50 Hz unit. At the

[11] Where the tubes attach to the headers or manifolds.

time of the introduction, the M701J engine had record breaking performance at almost 500 MW. The MHI F-class has always been air-cooled whilst the G- and J-class has steam cooled combustion liners. The inherent flexibility issues have ”forced” MHI to develop and introduce air-cooled G-units (GAC). Mitsubishi revealed that they are working on an air-cooled J-class engine at the PowerGen Int’l conference already in -12. The 60 Hz air-cooled unit (M510JAC) performance is available, showing only a minute drop in output of 17 MW.

MHI has a long tradition in validating their 60 Hz units at the T-point facility in Japan. The T-point station is a utility power plant where long-term testing can be conducted as part of commercial operation. This is especially important from an insurer’s point of view, because the engine can pass the 8,000 hours’ insurers mark before market introduction.

Mitsubishi has also followed the flexibility trend by introducing a high-performing all air-cooled version of their F-class unit. The new engine is called F5 and is rated at 359 MW. The combined cycle power is above 525 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).

3.1.2 Alstom

General Electric has acquired the Alstom Power segment and the development situation is unclear to the present authors. In order to prevent from a GE market dominance in Europe, the Alstom gas turbine portfolio has been sold to Ansaldo in Italy. The deal includes the GT26 unit and service for a part of the fleet. The newest addition to the Alstom portfolio is the H-class unit called GT36. The engine is not commercially available yet and neither the performance figures not the launch date have been published. The deal will include the Alstom subsidiary PSM in the US. PSM is a third-party provider (to e.g. GE-units) with their own combustor technology[12].

Alstom offered the F-class GT26 unit with a guaranteed performance of 61 percent since -11. This was possible by introducing aero-engine technology from Rolls-Royce into the GT26 (GT24) 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 unit can be operated at low load with only the first set of burners in operation and fully closed IGV/VSVs. Alstom has also introduced shutting down a combination of burners for CO mitigation during partial load.

3.1.3 General Electric

General Electric has a product range called ”FlexEfficiency™ 50” Combined Cycle Power Plant. The performance follows the trend of the other OEMs and the 50 Hz version is rated at 460 MW and 60.2 percent efficiency. The GE-plant has an impressive ramp-rate of more than 24 MW/min (or 48 MW/min in 2+1 configuration). The

[12] FlameSheet™

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 38 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 whilst maintaining the stage numbering or nomenclature[13].

The most reason unit is the ”new” HA-class[14] where GE has re-introduced the H-unit with air-cooling. The HA-unit is available in two 50 Hz sizes at 397 and 510 MW for the HA.01 and HA.02, respectively. The combined cycle is 1,181 and 1,515 MW for each of the plants in 2+1 configuration. The stated efficiency is 61.6 percent for the combined cycle. The hot start-up time is less than 30 minutes for both plants. The ramp rates are 60, 60 and 120 MW/minute for the gas turbine, 1+1 and 2+1 configurations, respectably. General Electric manufactured the first HA-unit in 2014 and had (during summer of -14) 13 60- and 50 Hz units on order. There has also been an intent to lower the life cycle cost (LCC) by five percent.

General Electric has also re-introduced the small 50 MW F-calls unit called 6F.01. The unit offers 51 MW at am efficiency of 38 percent in simple cycle operation. The combined cycle performance is 75 and 150 MW for 1+1 and 2+1 configuration, respectively. The efficiencies are 55.8 and 55.9 for each plant configuration. The driver for the high combined cycle efficiency is the combination of relative high gas turbine efficiency (38 percent) and the flue gas temperature of 597 °C, which results in an overall high cycle efficiency. This unit offers, more or less, the same performance level as for the Siemens SGT-800 based plants.

The 16 MW Nova LT16 was introduced in 2015 and is aimed at the oil- and gas market. The twin-shaft unit is intended as both mechanical drive and power generation. The efficiency is 37 and 36 percent for the driver and the power gen unit, respectively. General Electric has until the summer of -15 sold two units to a customer in Russia and TransCanada.

3.1.4 Pratt & Whitney (PW Power Systems)

The Pratt & Whitney energy segment (PW Power Systems) was acquired by Mitsubishi Power Systems in 2013. 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. The unit follows the PW practice with ungeared power turbines and the unit can be operated in synchronous condensation mode (phase compensation) with or without a SSS-clutch. The FT4000 test were completed during the summer -14. The engine is commercially available with two disclosed sales.

[13] Quite complicated to change all manufacturing drawings
[14] Nickname “Harriet” after HA

3.1.5 Hitachi

Hitachi has introduced the H80 in the 100 MW power bracket, in 2010. The engine was uprated in 2013 and offers higher performance. The upgrade is mainly based on both compressor modification and increased combustor firing level. The original design intent was a ”drop-In” replacement for older GE frame units, hence a requirement to match the precursors exhaust data and footprint.

3.1.6 Siemens Energy

The Siemens SGT5-8000H has upgraded and offers today a rating of 400 MW at an efficiency of 40.0 percent. The corresponding combined cycle numbers are 600 or 1,200 MW for 1+1 and 2+1 configurations, respectively. The stated efficiency is for both cases >60 percent. As earlier mentioned, the in-situ performance at a German E.ON-plant shows a certified efficiency of 60.75 percent.

3.1.7 Siemens Industrial and Rolls-Royce

The land-based Rolls-Royce aero-derivate segment has been acquired by Siemens in 2014. In the 40 MW bracket, two new engines were introduced in 2010-2011 by Rolls-Royce (RB211-H63) and Siemens (SGT-750). Both engines have dry ratings around 37 MW and electrical efficiencies around 40 percent. The Rolls-Royce H63 engine was never realized due to undisclosed reasons. The new Siemens engine 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 has uprated the SGT-800 unit and offers today an output of 53 MW at 39 percent efficiency. The combined cycle output is in 2+1 configuration 149.9 MW at an efficiency of 56.2 percent. The previous rating was 50.5 MW@ 38.2% and 143.6 MW @ 55.4% for the simple- and combined cycle, respectively.

3.1.8 Kawasaki

Kawasaki has introduced a new 30 MW GPB300D unit with an efficiency of 40.1 percent. The new high-performing engine is not following the common single-shaft structure by KHI and the engine is probably aimed at the mechanical drive market. The new combustor technology offers very low emission levels and rig testing shows as low as four ppm NOx. The burner technology is also used on the smaller GTM7 and L20A engines. The main driver for the high efficiency is the relative high cycle pressure ratio of 24.5.

3.1.9 Comments

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 is 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.

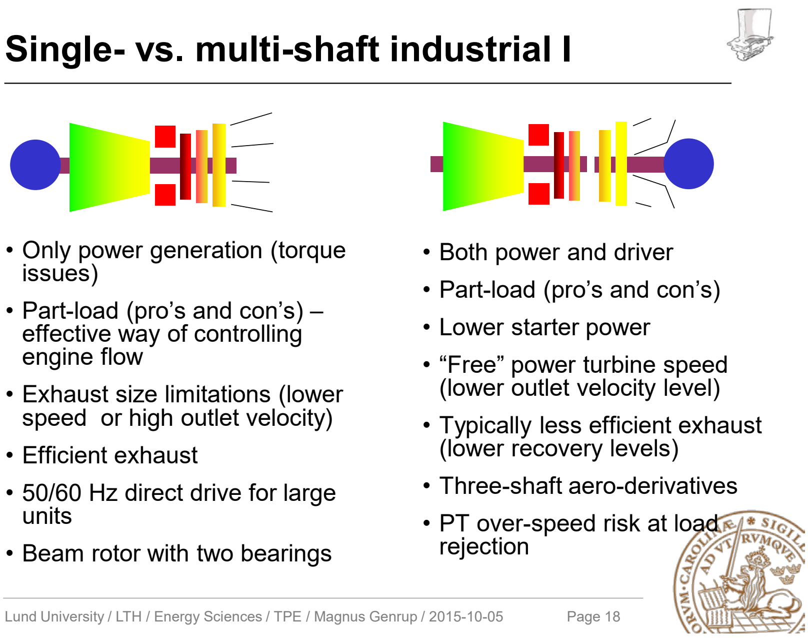


Figure 4-1. Single vs. multi-shaft industrial unit

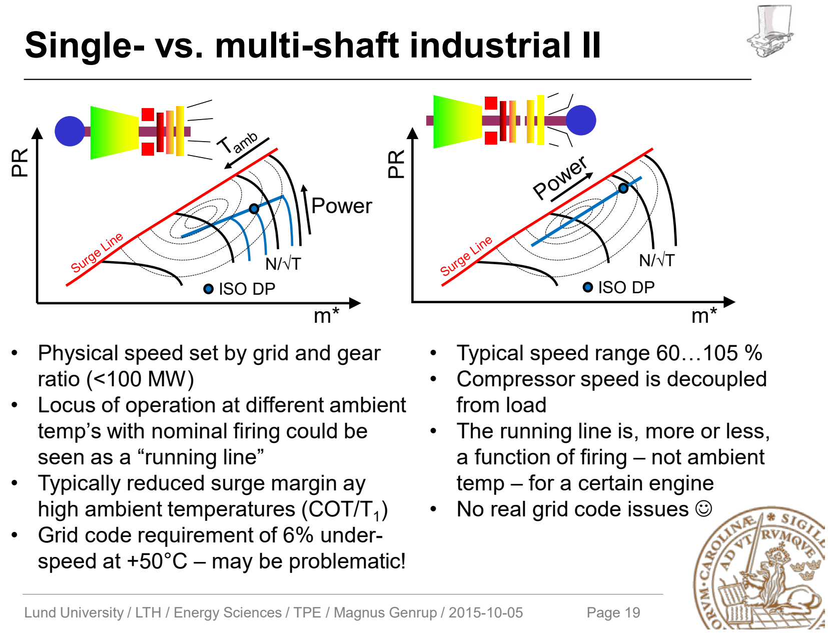


Figure 4-1. Single vs. multi-shaft industrial unit

4.2 NUMBER OF PRESSURE LEVELS IN A COMBINED CYCLE

The number of pressure levels are mainly driven by size since it adds cost to the plant. Three pressure levels (with reheat) are typically only used for the largest plants where the highest level of performance is required. This means that most mid-sized units will have a two-pressure steam cycle, whilst the larger will have three-pressures. The utility size steam turbine is the ideal candidate since it typically has a HP-, IP and LP cylinder (or a combined HP/IP). The IP steam flow is introduced before the re-heater and does not require a separate induction point in the steam turbine.

The reason for adopting additional pressure levels is shown below. The area between the red flue gas line and the blue water/steam line in figure in figure 4-2 can be shown to be the exergy destruction[15]. The most straightforward way is to discuss in terms of transferring an infinitesimal heat quantity (dQ) from T_1 to T_2. The maximum possible work potential of the dQ at state 1 is the Carnot process between T_1 and a reference state T_0. The same reasoning at T_2, offers a lower work potential – simply because the available heat is at a lower temperature. The total destruction of entropy is obtained by the integral $\int(1-T_0/T)dQ$.

[15] Should from a fully rigorous point of view be the inverse of the temperature (i.e. 1/T)

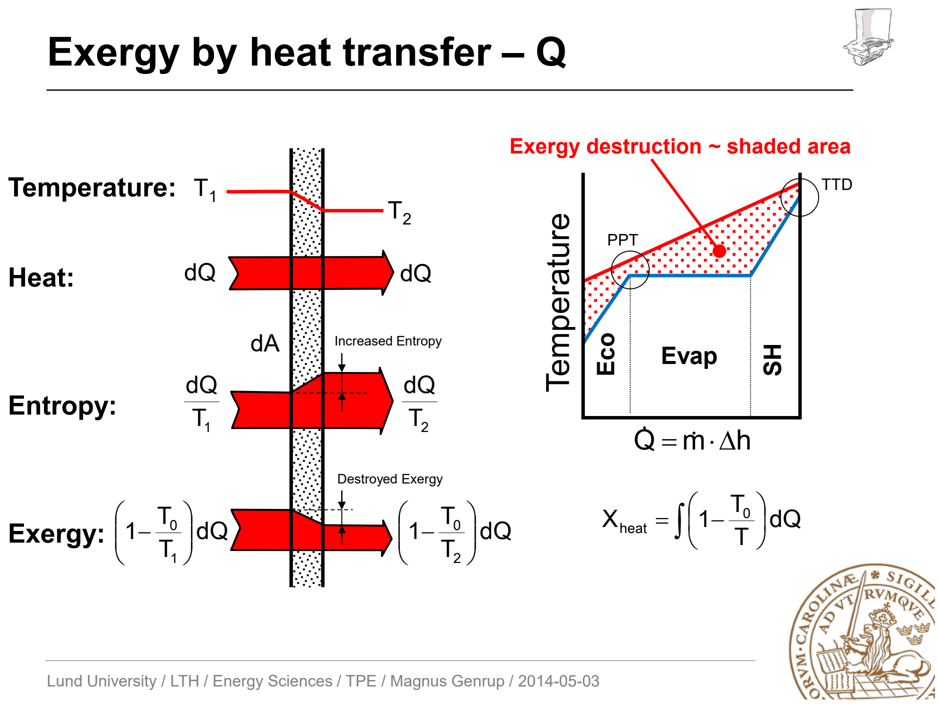


Figure 4-2. Single vs. multi-shaft industrial unit

The reasoning in figure 4-2 and above is based on a single pressure level. An additional pressure level will decrease the area between the flue gas and water / Steam side – hence lower exergy destruction. A third pressure level and reheat will act in the same manner and reduce the exergy destruction within the system.

4.3 MICRO TURBINES 20-200 KW

There are a handful of small units available from manufacturers like Ansaldo[16] and Capstone. There engines are typically recuperated and have efficiencies around 30…33 percent. Driven by size, radial components are generally used since the volumetric flows are small.

	Ansaldo 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 Ansaldo 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.

4.4 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

[16] Formerly Turbec

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[17]	Solar Titan 130	Comments
Power, kW	7,965	14,326	15,000	
Efficiency, %	34	35.4	35.2	
Pressure ratio, -	17.6:1	18.9:1	17.0:1	
Exhaust temp., °C	507	540	496	
Exhaust flow, kg/s	26.9	44.3	49.8	
First cost €/kW 2015	576	470	496	
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 configuration	1, 1+1	1+1	1, 1+1	

The selected engines show to 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[18] 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.

There are recent examples of using organic Rankine cycle (ORC) for heat recovery (figure 4-3). 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 (cyclo-pentane) seems suitable due to its properties. It is also common practice to have an intermediate thermal oil circuit (see the figure below).

[17] Newer rating presented at Power-Gen 2010.

[18] E.g. gasifier.

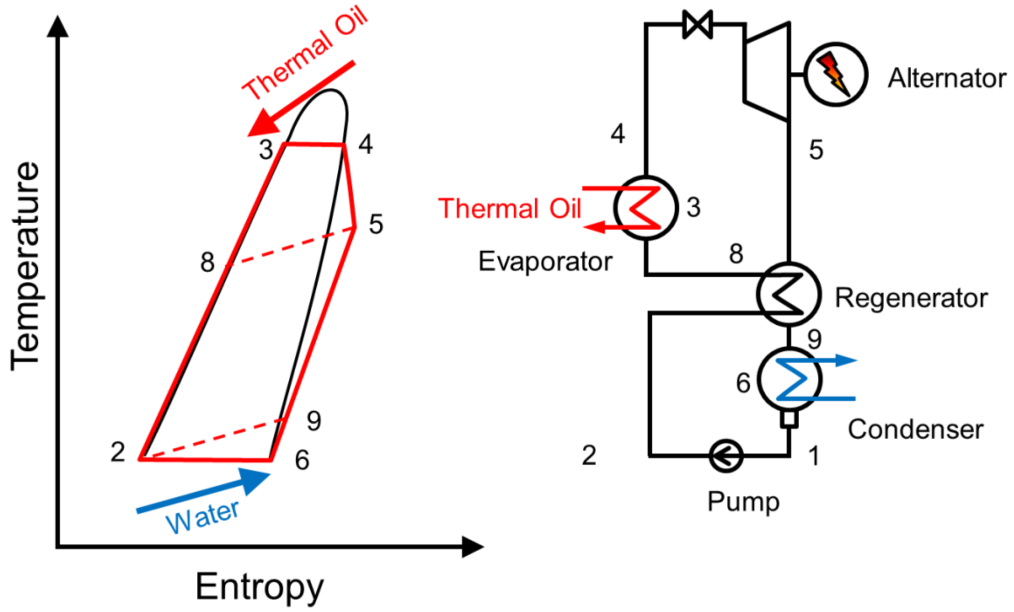


Figure 4-3. Schematic organic cycle

The word low-grade heat is used in a sense that the vapor process is operated at a lower temperature level than a normal combined cycle.

4.5 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, cogeneration and combined cycle) and mechanical drive. Gas turbines are found both upstream (e.g. off-shore compression) and downstream (e.g. pipe compression[19]) in the oil and gas industry. The driver market is typically from the lower power bracket up to some 40 MW.

	GE LM2500 RC	Siemens SGT-800[20]	Rolls-Royce Trent 60 DLE ISI	Comments
Power SC, kW	36,906	50,500	53,119	
Power CC, kW	48,897	71400	64,600	
Efficiency SC, %	37.0	37.7	42.4	
Efficiency CC, %	50.6	55.1	53.6	
Pressure ratio, -	24.7	20.8	33.3	
Exhaust temp., °C	510	553	433	
Exhaust flow, kg/s	97.3	134.2	155.2	
First cost SC, €/kW	390	320	379	
First cost CC, €/kW	1,129	980	1,043	
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/Siemens Trent 60 are available as combined cycles at 71.4 and 64.6 MW, respectively. The SGT-800 has a combined cycle efficiency of 55.1 percent, whilst the Trent has 53.6 percent. The almost five 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 729 €/kW, whilst the SGT-800 plant is slightly lower at 726 €/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.

[19] A typical pipeline has a compression station each 15…20 km.
[20] This is not the most recent version of the SGT-800

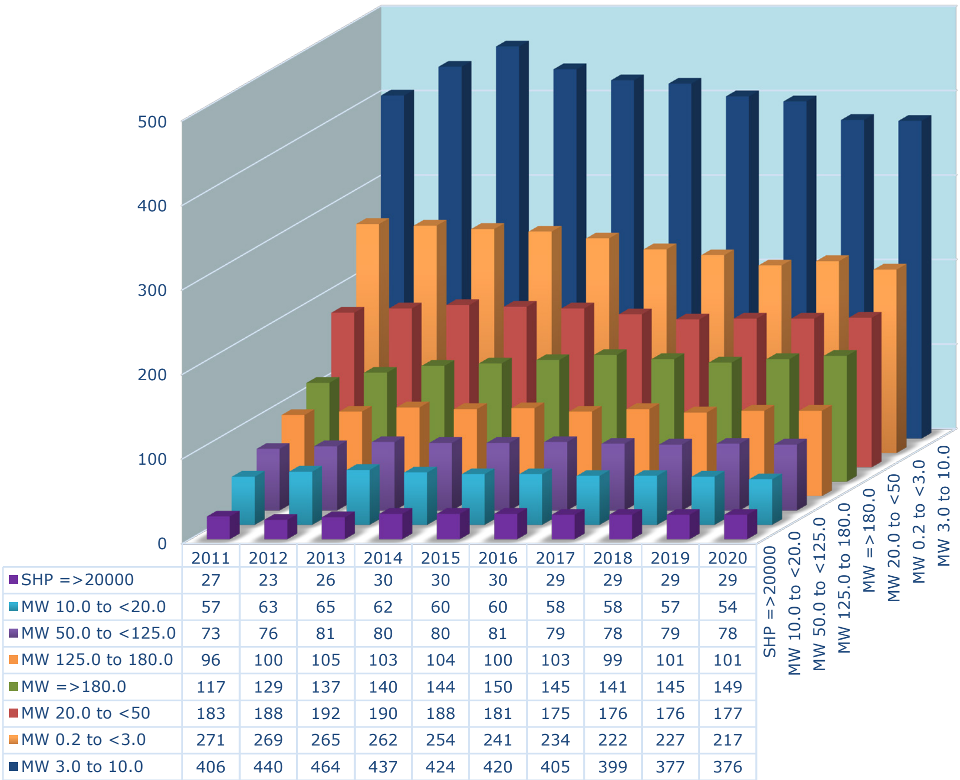
4.6 LARGE UNITS

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 9F.05	MHI M701F5	Alstom GT26	Comments
Power SC, MW	299	359	345	
Power CC, MW	460	525	500	
Efficiency SC, %	38.7	40.0	41.0	
Efficiency CC, %	60.2	61.0	60.0	
Pressure ratio, -	18.3	21	35	
Exhaust temp., °C	642	611	616	
Exhaust flow, kg/s	667	729	715	
First cost SC, €/kW	211	199	192	
First cost CC, €/kW	599	599	606	
O&M costs, €/MWh	----------- See chapter 5 ----------			
Fuel flex - WI	?	?	±10%	
Emissions NOx/CO ppmv	25/?	?	25/?	
Cooling	Air	Air	Air	
Shaft config.	1	1	1	

4.7 SALES TRENDS 2011-2020

The total number of gas turbine sales during 2011-2020 are estimated to 12,500 units. The number may at, a first glance, seem very high but still only represents some one quarter of the world’s total gas turbine fleet of 46,500 engines in 2009. The distribution of each power bracket per annum is shown in the figure below. The presented numbers are based on Forecast International (original numbers were published in Turbomachinery International) and were compiled before the recent recession and the high renewable penetration in continental Europe. The European situation is today quite different due to the high level of renewable production capacity that has reduced the electricity prices. The natural gas price is still high - rendering in poor production economy (especially in the absence of a non-spinning reserve compensation system). This is the driver behind the low utilization rate, and even mothballing, of Europe’s combined cycles. The sales trend today is due to the mentioned reason, quite different from when the graph below was prepared.



	2011	2012	2013	2014	2015	2016	2017	2018	2019	2020
SHP =>20000	27	23	26	30	30	30	29	29	29	29
MW 10.0 to <20.0	57	63	65	62	60	60	58	58	57	54
MW 50.0 to <125.0	73	76	81	80	80	81	79	78	79	78
MW 125.0 to 180.0	96	100	105	103	104	100	103	99	101	101
MW =>180.0	117	129	137	140	144	150	145	141	145	149
MW 20.0 to <50	183	188	192	190	188	181	175	176	176	177
MW 0.2 to <3.0	271	269	265	262	254	241	234	222	227	217
MW 3.0 to 10.0	406	440	464	437	424	420	405	399	377	376

Figure 4-4. Gas Turbine Sales Prognosis 2011-2020 (based on Forecast International)

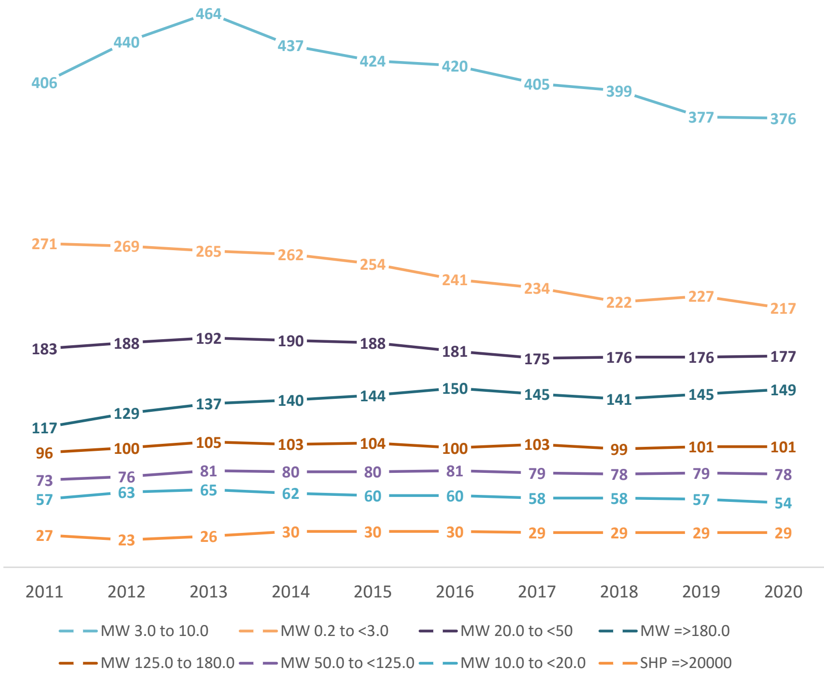


Figure 4-5. Gas Turbine Sales Prognosis 2011-2020 (based on Forecast International)

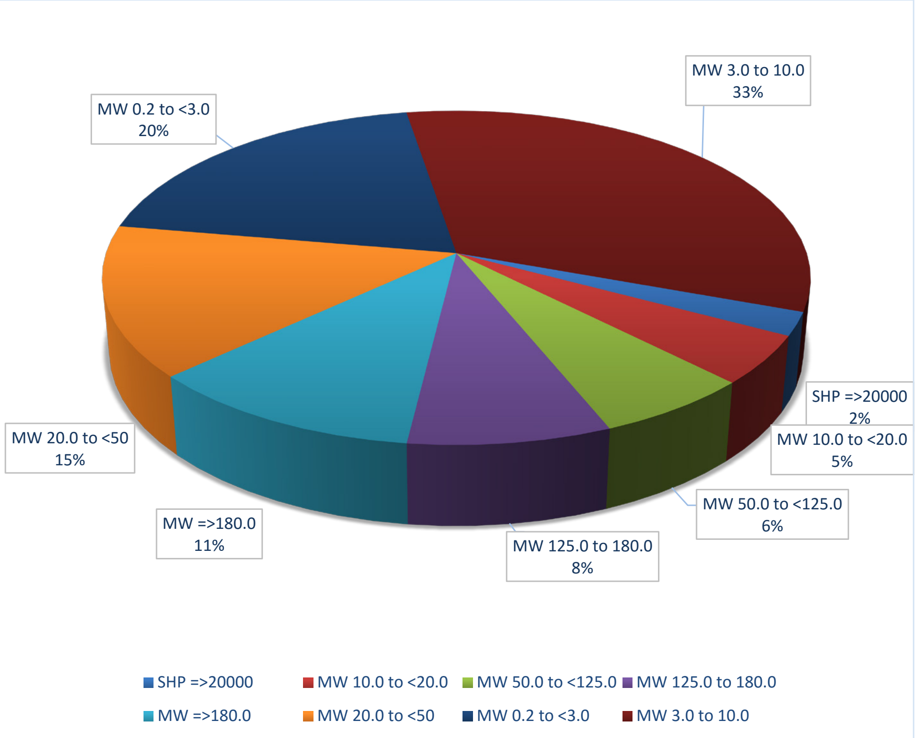


Figure 4-6. Gas Turbine Sales Prognosis 2011-2020 (based on Forecast International)

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 2000-2012

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 trend 2008-2009 shows a plateau that is probably driven by the recent recession in the world economy.

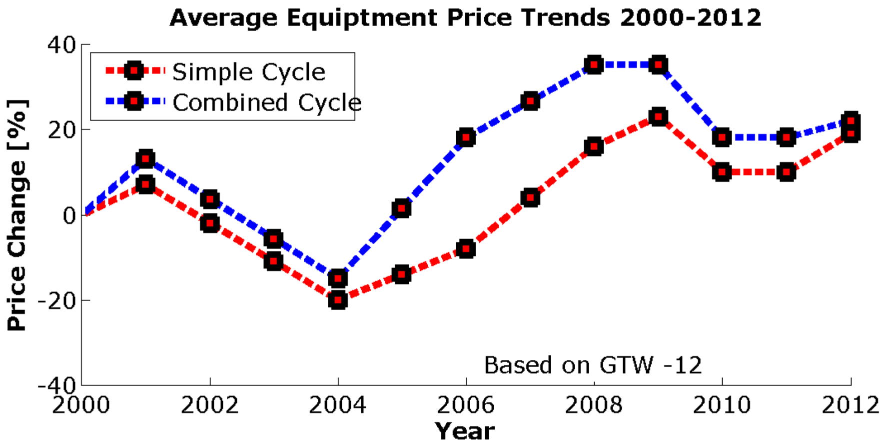


Figure 5-1. Gas Turbine World equipment prices 2000-2012[21]

5.2 2014 PRICE LEVEL

The presented figures are valid for 50 Hz combined plants and are based on Gas Turbine World 2014 GTW Handbook. The presented turnkey budget numbers in GTW are equipment only and FOB[22] factory in 2014. 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 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.

[21] No new data available (latest from 2012)
[22] Free On Board or Freight On Board as per INCO-terms.

Costs of compressor wash system and unit are excluded. The exact details are found in Gas Turbine World 2014 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 prizes have increased, on average, by 22 percent since 2012. The increase in prize is over the entire size range and no clear explanation is given in e.g. the Gas Turbine World handbook.

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. 2014 €):

- Electricity price: 30 and 60.0 €/MWh
- Fuel cost: 23.00 €/MWh >150
- 29.50 €/MWh <150
- O&M Cost: f(power) – See eq
- 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.

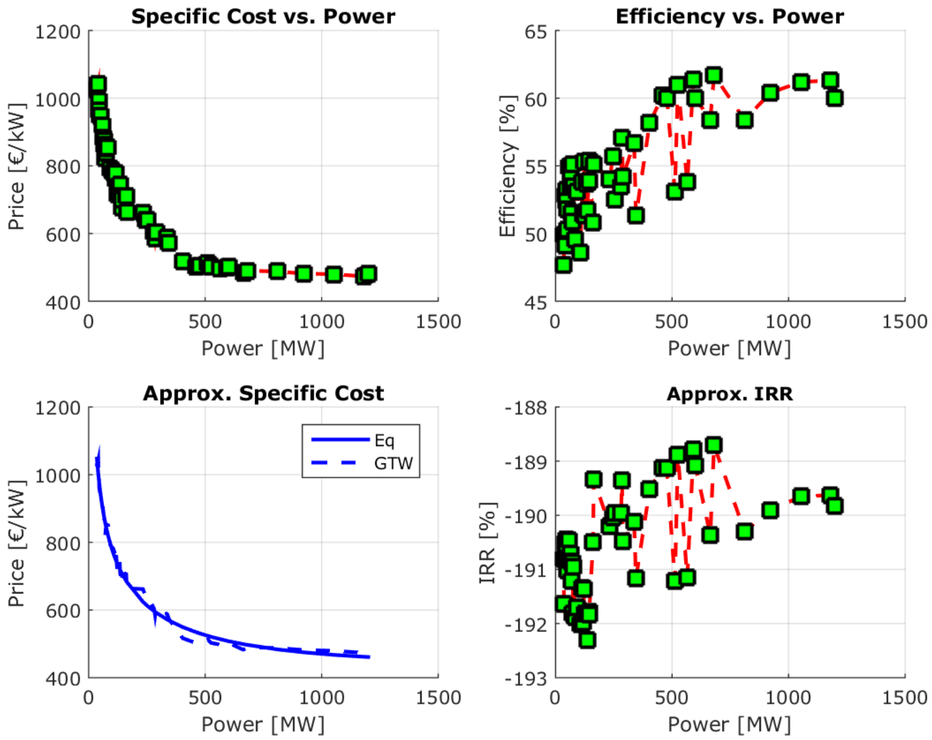


Figure 5.2. Power plant economics @ 30 €/MWh.

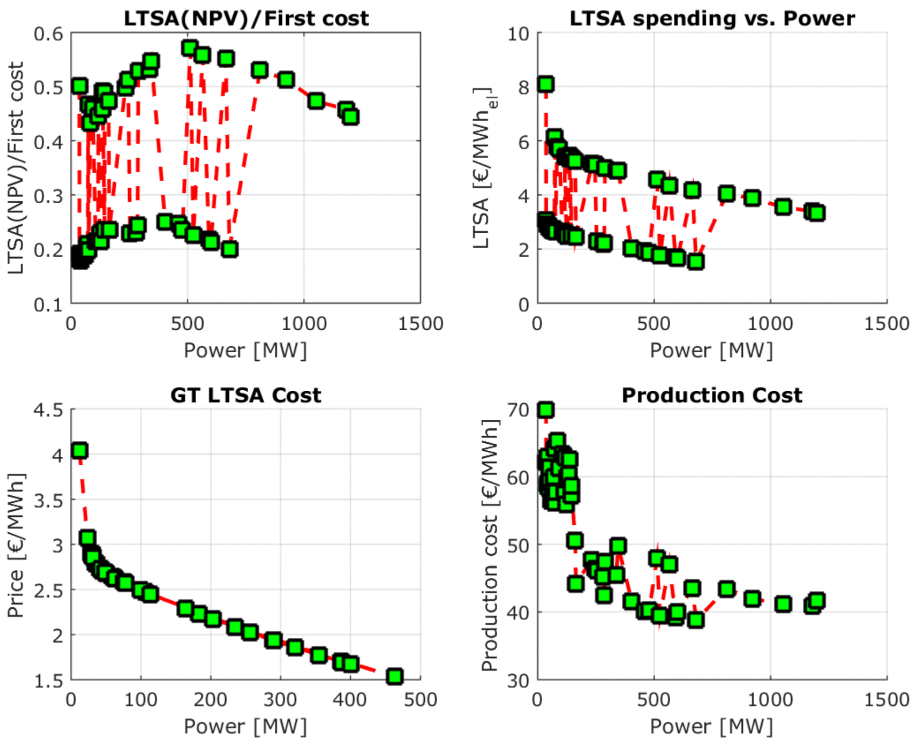


Figure 5.3. Power production costs @ 30 €/MWh

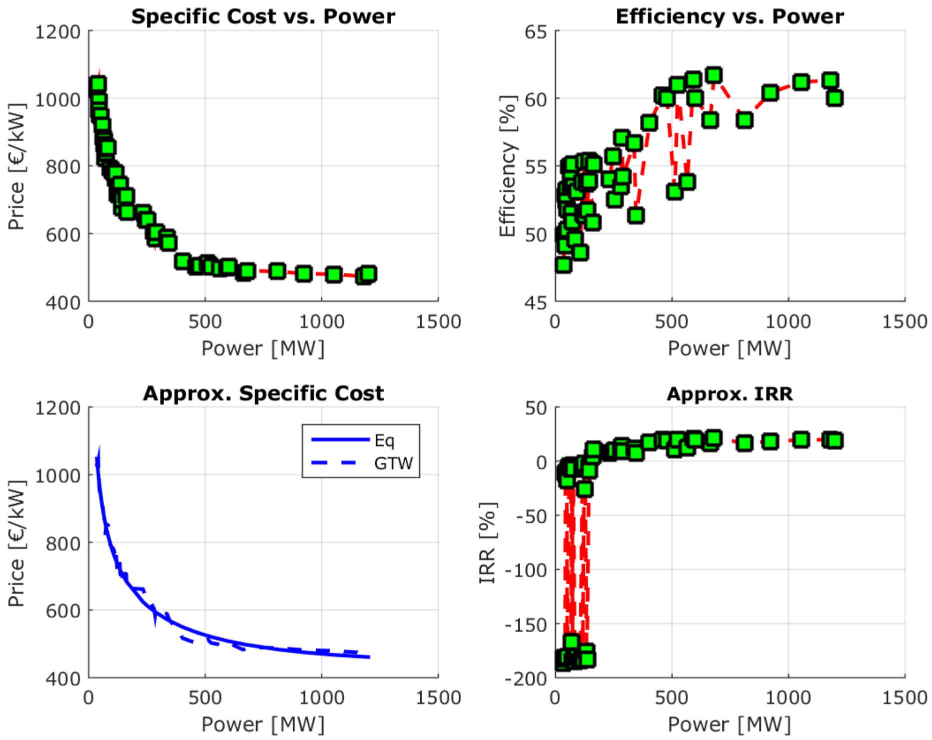


Figure 5.4. Power plant economics @ 60 €/MWh.

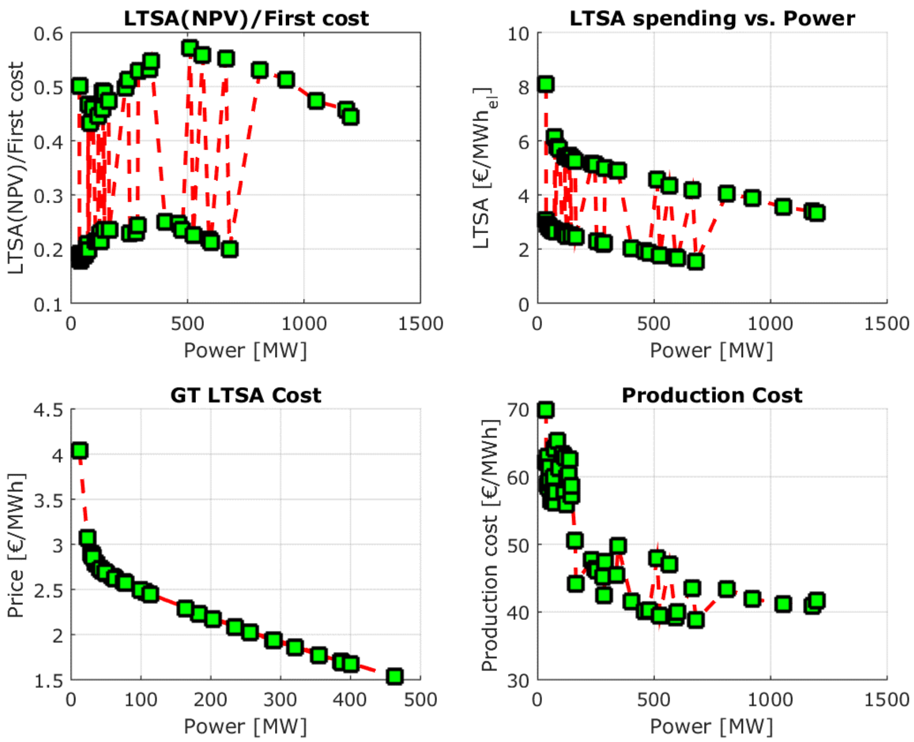


Figure 5.5. Power production costs @ 60 €/MWh

The analysis shows that most of the smaller plants has poor or even negative internal rate of return, even for the 60 €/MWh. The smaller units are heavily penalized by the very high natural gas prize below 150 MW.

The plant specific cost (in 2014 €/kW@0.7539 €/$) can for analytical purposes be approximated with:

$$\text{Spec. cost} = 3020.169 \cdot P_{el}^{-2.99664746} + P_{el}^{0.65}$$

The preceding equation is plotted in figure 5.3 (bottom left), showing a R2-value close to unity (i.e. good fit).

The other cost data are not amendable for regression analysis since the spread is significant due to a mixture of engine generations[23].

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 rule of thumb is that the maintenance cost is about 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 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):

$$\text{GT LTSA} \approx 5.313 \cdot e^{-0.1219 \cdot P_{el}} + 2.852 \cdot e^{-0.001341 \cdot P_{el}}$$

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

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.

5.2.1 Lower running hours than planned

The high penetration of renewables has turned the combined cycle units from being mid-merit plants to peakers. It is hard to discuss this in general terms since it varies significantly between the operators. One safe conclusion, however, is that even high-performing combined cycles are mothballed in Europe. The reason is a complex combination of high levels of renewables, low power consumption and the relative low cost of coal. A high-performing unit has a very large cost when compared to lower technology levels – this results that the features that will turn the plant into a fuel-efficient asset will turn it into a financial burden if operated with a low number of fired hours. One can show that a unit that has been procured for 6000 hours/annum, may end up with a four folded increase in production cost when compared on a MWh-basis. This is because of the ”fixed” capital- and staff cost have to be covered with fewer operational hours.

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 $1.96 \cdot \sigma \sim$ ”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, unfair 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[24], 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.

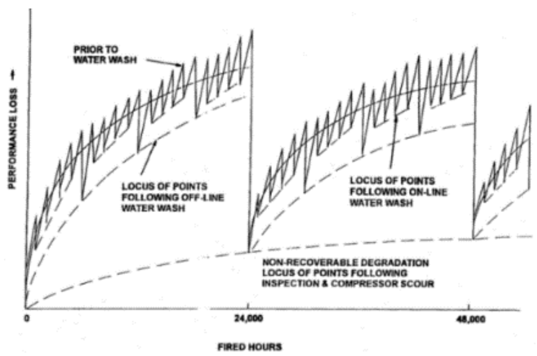


Figure 5-4. Typical engine degradation pattern (large leaps are maintenance events – see next section for details)

[24] 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 an efficient 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. Kindly note the analysis assumes the same exhaust flow as for the compressor inlet. This means that the fuel flow is the same as the total overboard bleed.

6.1 GENERAL ELECTRIC HA-CLASS

The 50 Hz General Electric HA-unit has the highest power output in the business with its 510 MW in simple cycle. The engine follows the current trend with air-cooled H-class. The driver for abandoning steam cooling is primarily flexibility issues (and cost).

The 50 Hz HA-unit is available in two sizes namely 397 and 510 MW for the HA.01 and HA.02, respectively. The combined cycle power output is 1181 and 1515 MW for each of the plants in 2+1 configuration. The efficiency is 61.6 percent for the combined cycle.

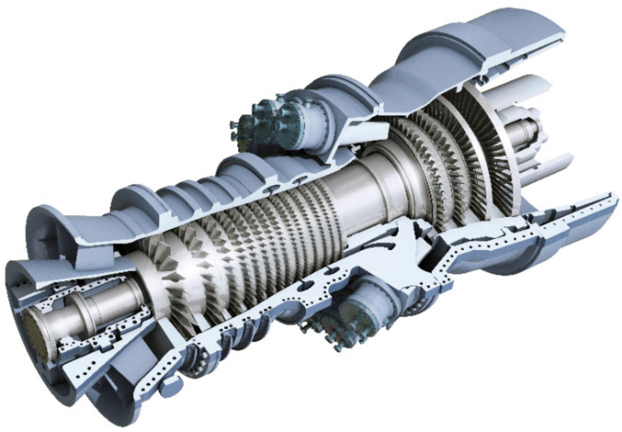


Figure 6-1. General engine layout (courtesy of General Electric)

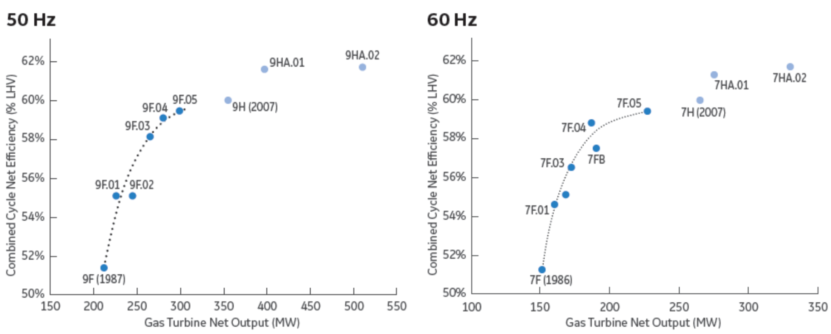


Figure 6-2. General Electric F- and H-class combined cycle efficiency (courtesy of General Electric)

General Electric manufactured and tested the first HA-unit in 2014 and had (during summer of -14) a total of 13, 60- and 50 Hz units on order. The first commercial production unit is scheduled to have completed full load validation during the first quarter of -15.

The hot start-up time is less than 30 minutes for both plants. The ramp rates are 60, 60 and 120 MW/minute for the gas turbine, 1+1 and 2+1 configurations, respectably. The

HA-class has no hydraulics, 50 percent fewer junction boxes and 70 percent less terminals. There has also been an intent to lower the life cycle cost (LCC) by five percent.

The difference in power between 9HA.01 and 9HA.02 is mainly due to higher air flow and increased firing. The air flow for the 9HA.01 is approximately 826 kg/s and the larger 9HA.02 has an increase of about 20 percent to almost 1,000 kg/s. The combination of both increased mass flow and firing calls for a re-designed turbine section. The reason for this is mainly compressor surge margin and turbine exhaust loss. The pressure ratios are 21.1:1 and 23.5:1 for the .01 and .02 versions, hence confirming the above reasoning. The situation for the 60 Hz product range is different and GE seems to limit the increase in firing in order to maintain the same exhaust temperature. The reason is not disclosed but the sheer power classes of the steam turbines are 260 and 525 MW for the 1+1 and 2+1, respectively. This size of steam turbines should be available in the 600…620 °C temperature range for the admission and reheat and this would explain the exhaust temperature level.

The air flow of almost 1000 kg/s@3000 rpm is, so far, unmatched by other OEMs and indicates an advanced aero-type of compressor.

Table 6-1. Performance data 50 Hz HA-class

	9HA.01	9HA.02
Net power output, MW	397	510
Net efficiency, %	41.5	41.8
Pressure ratio, -	21.8:1	23.5:1
Exhaust flow[25], kg/s	826	996
Exhaust temperature, °C	621	652
Combined cycle 1+1		
Net power output, MW	592	755
Net efficiency, -	61.6	61.8
Combined cycle 2+1		
Net power output, MW	1,181	1,515
Net efficiency, -	61.6	62.1

In addition to the previous discussion, the specific power for the 9HA.01 is approximately 480 kWs/kg of air going through the engine. The larger 9HA.02 has a higher figure of 512 kWs/kg which in a certain (or firm) indicator of higher power density – hence higher firing level. It should, however, be noted that the cycle pressure ratio has a profound impact on the produced net power per unit of air that is going through the system. In this case, the larger unit has a higher pressure ratio suggesting that the firing level also in increased to compensate for the relative increment in compressor power. The older 9F.05 has an assumed combustor outlet temperature (COT) on the order of 1,500 °C at a pressure ratio 18.3, resulting in an exhaust temperature of 642 °C. The 9HA.01 has a pressure ratio of 21.8 and an exhaust temperature of 621 °C. The numbers for the 9F.05 results in a specific output of 448

[25] Includes the flow from the ”frame-blower”

kWs/kg. It is, however, not possible to state whether this is solely due to higher firing because there has most likely been substantial component development between the F- and new H-class.

6.1.1 Compressor

The 14 stage compressor in the 50 Hz 9HA.01 is linear scaled from the smaller 60 Hz 7HA.01[26]. The principle of scaling is to maintain both the aerodynamics as well as the stresses by keeping all the speeds (blade and gas) at the same levels. The linear scaling factor is then (60/50)2=1.44 and this would impose the same ratios for the flow as for the output – as observed from the performance figures for both sizes. This is, however, not an exact science since both the Reynolds numbers as the relative tip clearances naturally changes when changing the size. This could be explained by e.g. the larger blade chords (hence characteristic length) for the 50 Hz version whilst maintaining the gas velocities. The running clearances are always as small as possible and if one assumes a certain minimum gap (in mm), then the relative impact on performance for the larger compressor will be less. A more detailed (or involved) discussion is outside the scope of the current report.

The air flow of almost 1,000 kg/s is unmatched for land based engines and is, indeed, a game changer. The design features of the front stages are not known to the authors, but other OEMs units with lower flows have tree-dimensional blading in order to reduce losses related to shocks. It is common practice to compare compressor designs at 3,000 rpm (i.e. re-calculate a certain flow at a certain speed into 3000 rpm whilst maintaining the velocity triangles). The state-of-the art for twin-shaft industrial units is to stay below 1,000 kg/s@3000 rpm. The reason for this figure is two-folded since one should avoid to have tip Mach numbers exceeding 1.3…1.4 and, at the same time, stay below 0.6…0.65 for the meridional Mach number. The former governs the losses associated with shocks in the first stage whilst the latter governs the compressor choke behavior. Single-shaft units have, so far, stayed below around 860 kg/s @3000 rpm (MHI J-class).

The pressure ratio is 21.8:1 and 23.5:1 for the 9HA.01 and 9HA.02, respectively. The average stage pressure ratio (PR1/n) is around 1.25 for both designs. This number should be seen as a conventional level for a 3000 rpm design.

The compressor follows the current practice with IGV and four variable stator vanes (VSVs). The airfoils are three-dimensional with super-finish blades for maximum performance. The former means controlled diffusion with advanced stacking features such as variants of local lean and sweep. The latter is an effective means of reducing viscous blade loss – and fouling.

The new design also incorporates a ”hybrid” radial diffuser. The compressor diffuser is used to minimize the velocity (and gain static pressure) before the combustion system. A design without this radial feature cannot recover the tangential velocity component from the compressor outlet guide vane (OGV). Hence, a radial hybrid diffuser could provide some relief because the OGV may be designed with some optimum swirl. Another reason may be as simple as the length is reduced by introducing a radial diffuser.

All compressor blades are replaceable in-situ.

[26] Based on the 7FB.05 compressor

6.1.2 Combustion system

The combustion system is evolved from the 2.6+ system with advanced axial fuel staging. The 9HA 2.6+ system has <25 ppm(v) NOx and CO at base load. The system offers turn-down down to 30 percent gas turbine load. The 25 ppm(v) NOx level is most likely a consequence of the firing temperature level where air-cooled engines suffer from an ”air-shortage” for both DLE and turbine cooling.

The combustion system follows the current ”practice” with flame tubes rather than annular systems. There are many reasons for this and one is that it provides good scalability by just changing the number of cans. The larger 50 Hz unit has 16 cans whilst the smaller 60 Hz has 12. One could argue that this would have a negative impact on the temperature spread (OTDF), but the cans, however, effectively resolves the time- and dynamics problems associated with scaling the combustions system. The combustion process kinetics (incl. mixing) requires certain residence time to take place and this results in certain length requirements.

The unit has dual fuel (i.e. gas- and liquid) capability.

The 9HA units also have advanced model based controls. The exact meaning of the word in unclear but it most likely means rather advanced algorithms instead of e.g. simple staging, x and y tables and polynomials. One such thing may be the firing level per se where the ”standard” has been a rather simple function[27] of the pressure ratio and the exhaust temperature (and perhaps the cooling air temperature). This approach is adequate for most applications but does not provide a very exact control under all circumstances. One could also consider a case where the control system minimizes the exhaust temperature spread by individually controlling the burners in each tube. This is quite intricate because there is a load dependent tangential transport mechanism through the turbine stages. Another issue that could be introduced is compressor flutter avoidance measures where one limits the engine pressure ratio during cold days. Grid code requirements may also introduce issues when the engine should be able to perform at six percent under-speed at 50 °C (!). This is very challenging for a single-shat engine because the surge margin drops with COT/T_{amb}.

The dynamics of the combustion process per se is indeed challenging and several different control modes exists for staging (and splits). One, indeed, powerful means of avoiding lean blow-out (LBO) is to use the low-frequency (or cold-) tone in the combustion system as an indicator of being near LBO.

It should be emphasized that the previous reasoning is speculative since the details are not known to the authors.

6.1.3 Turbine section

The expander is the HA-unit has four stages which follows the practice from the previous H-class. The F-class has three stages which is adequate for the lower pressure ratio (and firing). The drivers for having a four stage design (rather than three) are mainly stage loading and available exhaust size. One can show that the efficiency scales with the stage loading ($\psi = \Delta h_0/U^2$). It means that one should have low loadings and this calls for more stages for a certain turbine total heat (or pressure-) drop. The last turbine stage should always be designed for a loading coefficient (ψ) of slightly above

[27] Cf. a polytropic expansion process

unity for good exhaust system performance. This means that a fourth stage will provide an effective means of optimizing the other three for maximized performance.

Single-crystal materials are used in the first stage and TBC is used in both the first and second stage.

The turbine has state-of-the art three-dimensional blading for optimum performance.

6.2 GENERAL ELECTRIC 6F.01

The GE 6F.01[28] has been re-introduced[29] at 52 MW. The small F-class unit offers 51 MW at an efficiency of 38.4 percent in simple cycle operation. The combined cycle performance is 76 and 154 MW for 1+1 and 2+1 configuration, respectively. The efficiencies are 56.5 and 56.9 for each plant configuration. The driver for the high combined cycle efficiency is the combination of relative high gas turbine efficiency (38.4 percent) and the flue gas temperature of 603 °C, which results in an overall high combined cycle efficiency. The unit offer turn-down to 40 percent load and dual-fuel capability.

This unit offers, more or less, the same combined cycle performance level as for the Siemens SGT-800 based plant.

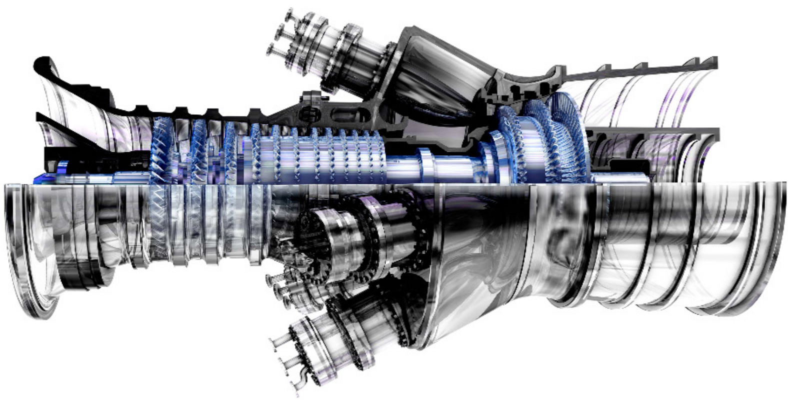


Figure 6-2. General Electric 6F.01 engine layout (courtesy of GE)

Table 6-2. Performance comparison between 6F.01 and SGT-800

Engine	GE 6F.01	Siemens SGT-800
Power output, kW	52,000	53,000
Efficiency, %	38.4	39.0
Pressure ratio, -	21.4:1	21.4:1
Stages, compr./turbine	12/3	15/3
Shaft speed, rpm	7,266	6,608
Exhaust flow, kg/s	126.1	137.2
Specific work, kWs/kg	404	386
Specific flow @3000 rpm	740	665

[28] Formerly the GE 6C

[29] Introduced at 42 MW in 2003 and upgraded to 46 MW in 2005

Average pressure ratio/stage ($PR^{1/n}$)	1.29	1.22
Exhaust temperature, °C	603	551
NOx emissions, ppm(v)	<25	<15
Combined cycle 1+1		
Net power output, MW	76	
Net efficiency, %	56.5	
Combined cycle 2+1		
Net power output, MW	154.0	149.9
Net efficiency, %	56.9	56.2

The combination of exhaust temperature, pressure ratio and assumed cooling flow level indicates a firing level close to ”normal” F-class. The approximate specific power is 404 kWs/kg in contrast to the large 9FB.05 which has 448 kWs/kg at a lower pressure ratio. A direct comparison is hard because a larger unit typically has higher component efficiencies and lower amounts of chargeable cooling flows. A safe conclusion is, however, that the main driver for high power density (specific work = work per kg/s air through the engine) is increased firing. The engine was presented at ASME IGTI in 2014 (in Düsseldorf) together with some data. The 6F.01 has, for example, 50 °C higher stator inlet temperature (SOT) than the old 46 MW 6C-unit. It is, again, hard to draw firm conclusions about the firing level because an improved cooling system will change the difference between COT and SOT[30].

The service intervals are set to (cf. the next section for an in-depth explanation):

- 16 kOH for combustion inspection (CI)
- 32 kOH for hot gas path inspection (HGP)
- 64 kOH for major inspection (MI)

6.2.1 Compressor

The compressor has 12 stages for a pressure ratio of 20.7:1. The specific flow is 740 kg/s@3000 rpm and the average stage pressure ratio is 1.29. All blades are field removable.

6.2.2 Combustion section

The combustion section of the 6F.01 is a typical canned system with GEs DLN 2.5 combustion technology. The engine offers 25 and 9 ppm(v) NOx and CO, respectively.

6.2.3 Turbine section

The turbine section has three stages otherwise no available information at the time of writing.

[30] An estimate indicates a firing temperature of

6.3 GENERAL ELECTRIC NOVA LT16

The new 16 MW Nova LT16 was introduced in 2015 and is aimed at the oil- and gas market. By virtue of being twin-shaft unit, the intention is to offers both a mechanical drive and a power generation unit. The efficiency is 37 and 36 percent for the driver and the power generation unit, respectively. The unit follows the GE (Italy) practice with a variable power turbine first guide vane. The reason for introducing such feature is mainly emission turn-down, part-speed performance and engine performance at either hot- or cold climate. The former is simply addressing the need for a lower air-flow for maintaining sufficient firing temperature at part load (in order to control the CO-level).

The power turbine design speed is 7,800 rpm and this level provides a good match with the relevant compressors within the 16 MW gas pipeline segment.

The engine is designed for 35,000 hours of operation for the gas generator and twice that number for the power turbine (i.e. four- and eight years for the gas generator and power turbine, respectively. The service concept is ”swapping” and the full engine can be replaced within 24 hours from reaching cold condition.

6.3.1 Compressor

The 12 stage compressor is a further development of the MS5002E (directly scaled from GE10) with an additional stage to accommodate the increased pressure ratio (19:1 from 17:1 in the MS5002E). The parent engine (MS5002E) has a much higher airflow of 101 kg/s at 7,455 rpm and the corresponding specific flow is 623[31] kg/s@3000 rpm. This rather low number indicates that the new Nova LT16 is not scaled directly from the parent compressor. No speed information is available at the time of writing but maintaining the same specific flow as for the GE10 results in approximately 10,300 rpm. This number could be considered as quite low for a state-of-the art twin-shaft unit and a reduction in compressor stage count could perhaps have been possible with a higher number.

6.3.2 Combustion section

The annular combustion system is derived from the smaller GE5-unit and currently offers 25 ppm(v) NOx. The target is, however, set at 15 ppm(v). The annular system has 39 double counter-rotating swirlers. The airflow to the system is controlled at part load by means of the variable power turbine. This means a possibility for high turn-down with respect to CO.

6.3.3 Turbine section

The turbine 2+2 turbine section rotates at 10,300 (assumed) and 7,800 rpm. The first stage of the power turbine is variable and therefore offer a variable swallowing capacity.

[31] 638 kg/s@3000 rpm for the GE10

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[32] 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 hundreds of percent's in contrast to typically ten for a complete new turn-key plant. The total aftermarket spending was 13.82 B€ (13.82×109) or 18.3 BUSD (18.4×109) in 2009[33]. The total world gas turbine fleet is on the order of 47,000 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.

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

[32] 1.5…1.7 (+) times

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

has 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 from 2003 in Power-Gen Worldwide (February and March) 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, however, is 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, fuses,… 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.

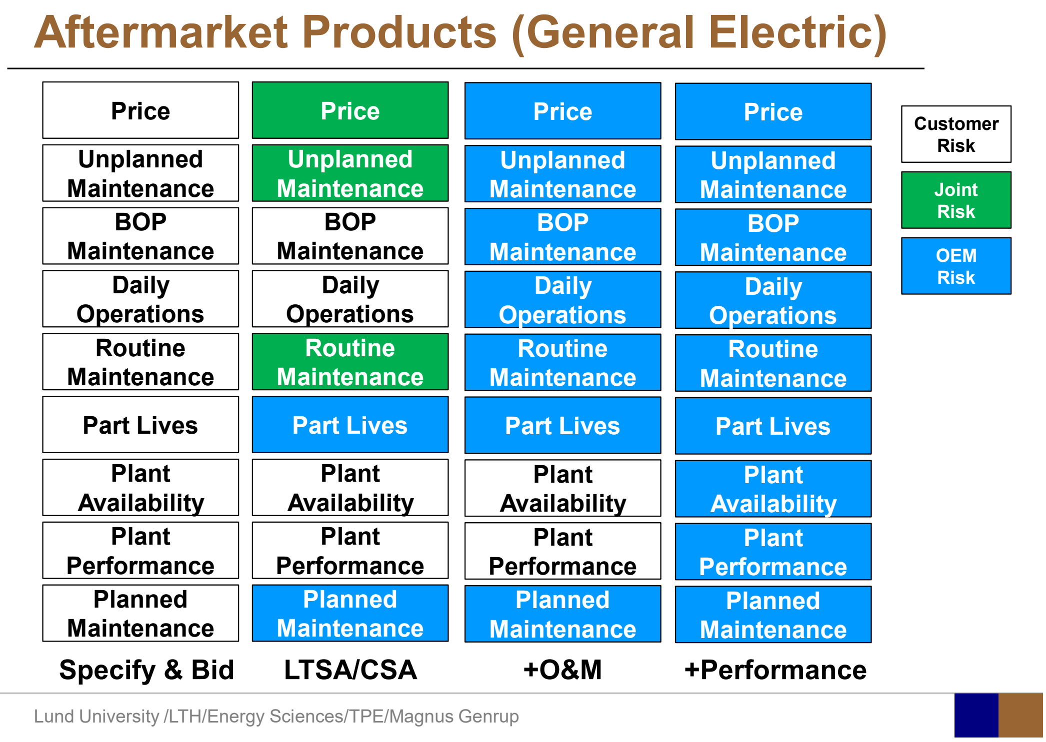


Figure 7-1. Different levels of contractual services.

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[34]. The interval for hot gas path inspection (HGP) is nominally 24,000 hours or 900 starts.

Factored hours for evaluating maintenance intervals:

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

Where:

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)

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[35]. If one assumes that the factor of ten is valid for a 56 °C increase in firing

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

[35] 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.

temperature and the nominal firing has a value of unity – then a reduction of 56 °C should give a number on the order of 1/2. 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 yields:

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

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[36]. 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.

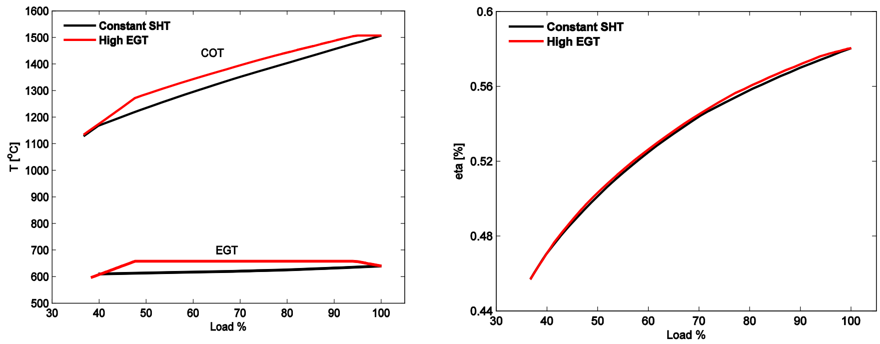


Figure 7-2. Plant performance GE 9FB.03 CCGT

The presented concept offers a possibility for having a lower firing level with only a minute performance penalty. The drop in firing level is on the order of 50 °C and this level of reduction may translate into a substantial increase in lifing. The control concept should not pose any hardware modifications and all standard engine control functions will still be active. The true limiting factor is the burner emissions and stability but any competent design should be possible to operate at a 50 °C reduction in firing level. The presented figures are valid for the specific plant, but the concept should be possible to implement on any state-of-the-art single-shaft gas turbine plant.

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,

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

the complete periodic system can be present. There are factors for crude oil and residuals of 2-4 and 3-4, respectively.

Factored events for evaluating maintenance intervals

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

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

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.

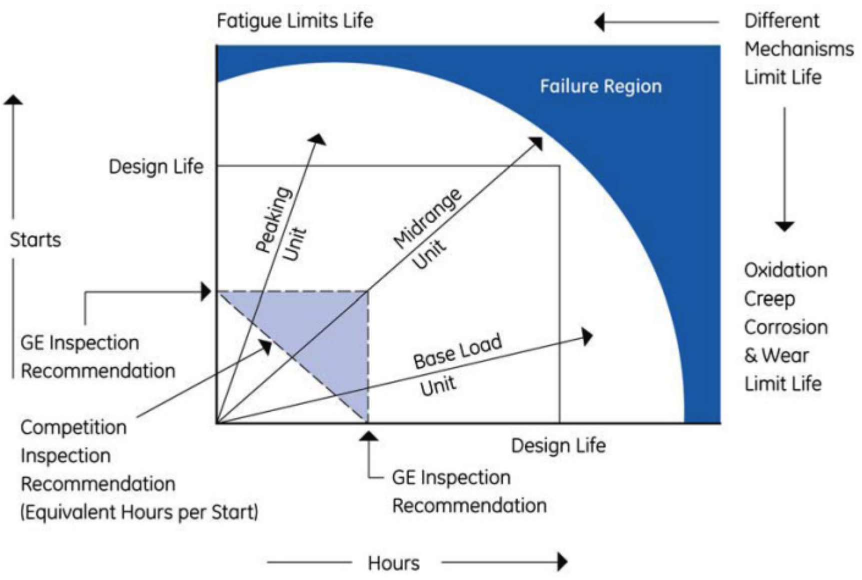


Figure 7-3. Lifing diagram for a heavy frame (Courtesy of GE).

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.

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

Where:

EOH	equivalent operating hours
OH	actual operating hours
Ffuel	factor depending on fuel
Ffiring	factor depending on firing level
nstarts	number of ”fired” starts
Fstarts	number of hours per start
Fload rate	load rate factor

As already mentioned, the method of equivalent hours should be avoided since the two life consuming mechanisms do not from rigorous point of view 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 450th 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[37], 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 ”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

[37] Corrosion, oxidation and erosion

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. The rejuvenation process will be further discussed in the next section.

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.

7.3.5 Blading repair strategies

The described different maintenance events (e.g. Hot Gas Path Inspection and Major Inspection) includes replacement or repair of hot components according to the OEM specification. The former is simply changing the old components for new OEM-components (or third party) whilst the latter is advanced restoration of the blades. The latter is often referred to as ”rejuvenation” where the blade is, from a metallurgical perspective, restored to pristinely. This means that a blade can be both repaired (e.g. weld and brazing) and restored – giving an extra life of two to three times the nominal convention. This offers a significant cost saving compared to the cost associated with new blades.

The word ”rejuvenation” is strongly associated with Liburdi (a Canadian company called) but it should, however, be mentioned that there are several third party organizations and OEM's that provides blading restoring services[38]. This report does neither recommend nor endorse any specific service provider. One conclusion,

[38] The terminology is also used in the literature and one prominent example is the Gas Turbine Engineering Handbook by Boyce.

however, is that it offers a significant saving by re-using the blades rather than scrapping. Rejuvenation includes HIP-treatment (Hot Isostatic Pressing) for e.g. closing creep voids and heat treatment for restoring the optimal morphology.

The two principal *recoverable* life consuming mechanisms are ageing and creep. The former is a metallurgical high-temperature phenomenon that is caused by diffusion-driven processes, where the alloying elements are degraded. A detailed discussion is outside the scope of the current report and only course explanation will be given for completeness. Typical nickel-based material ageing effects are: Strengthen gamma prime (γ') coarsening, carbide degeneration, grain boundary coarsening and formation of undesirable topologically close packed phases (TCP-phases). Creep is a result of high stress at elevated temperatures (cf. the Larson-Miller parameter) and ultimately changes the material properties by forming voids along the grain boundaries. Over time, the voids link up to create cracks in the base material. Creep is typically divided in to three categories, or levels, namely: primary, secondary and tertiary. The traditional way for gas turbine blades has been either directionally solidified (DS) or single crystal blades for maximum strength in the direction of the centrifugal pull. The cure to the creep damage is HIP-treatment where the voids are closed during process.

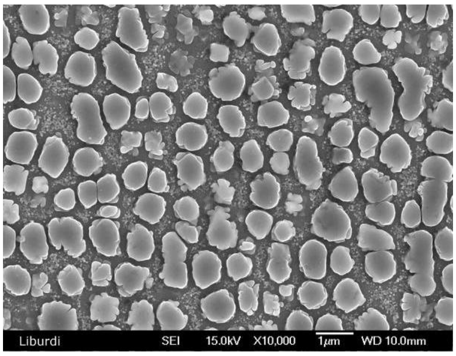
The first stage vane- and rotor in a state-of-the-art highly fired units are most likely to be single-crystal structures. Typical single-crystal materials today are CMSX4, Rene M5, PWA1483, etc. and it is not clear to what extent such materials may be rejuvenated. The problem lies in potential growth of secondary crystals during heat treatment for rejuvenation.

A typical single-crystal material has no grain-strengthen elements and is therefore very sensitive for growth of crystals. This is not to say that it is impractical, but it has to be evaluated on a specific material basis and it is therefore not possible to give firm information for certain materials. Standard gas turbine materials such as IN738, IN792, IN939, MAR M002, MAR M247, Hastelloy X (a standard combustor material), GTD 111/111DS appears to be ideally suited for restoration[39]. It should be noted, however, that the processes are different for each material.

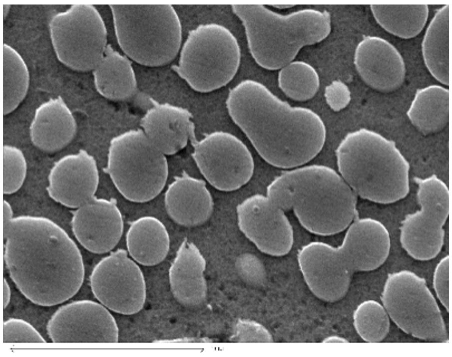
One has to distinguish between rejuvenation and re-coating, where the temperature levels are approximately 1,100…1,200 °C and 800 °C, respectively. The prime purpose of the former (rejuvenation) process is to restore the γ and γ' phase in the material. The latter is to provide e.g. oxidation protection by adding aluminum through a diffusion process. This means that it is possible to both remedy in-service issues and restore the lifing of the blades.

The two examples in figures 7-4 and 7-5, GTD111DS and IN738 represents two typical state-of-the-art turbine blading materials. Both examples have been rejuvenated with Liburdi's proprietary process and the results shows that the gamma prime phase has been restored.

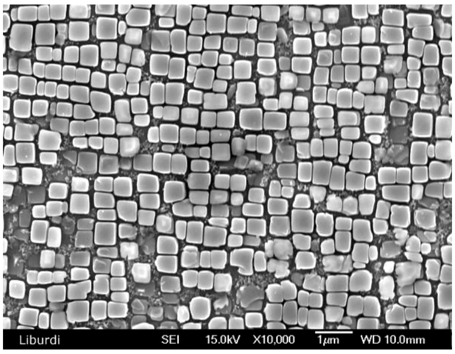
[39] Liburdi et al., "Practical Experience with the Development of Superalloy Rejuvenation", ASME GT2009-59444.



Blade root structure (service-exposed but at low temp) GTD111DS

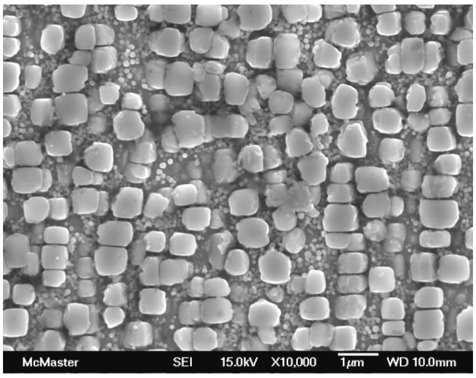


Service-exposed trailing edge at mid-height, GTD111DS

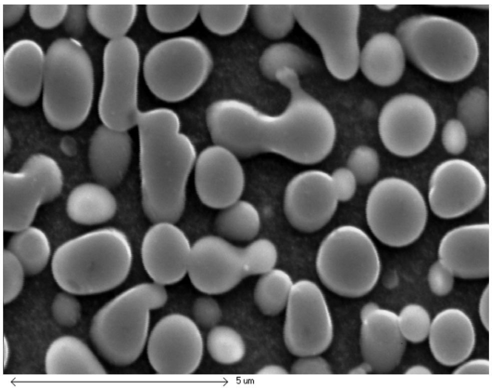


HIP and rejuvenated, GTD111DS

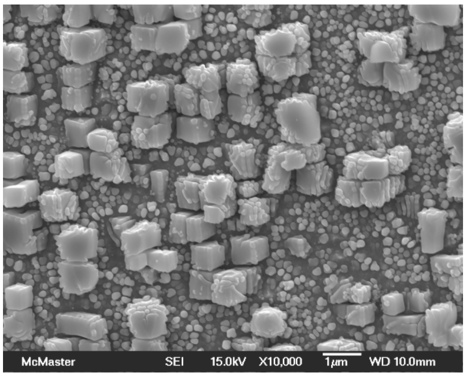
Figure 7-4. GTD111DS Rejuvenation (Courtesy of Liburdi Engineering)



Blade root structure (service-exposed but at low temp) IN738



Service-exposed trailing edge at mid-height, IN738



HIP and rejuvenated, IN738

Figure 7-5. IN738 (Inconel) Rejuvenation (Courtesy of Liburdi)

The preceding figures showed that the gamma-prime phase was significantly improved, or even restored, during the heat treatment. It is, indeed, hard to state whether the full lifing (at relevant engine conditions) has been restored. Liburdi has presented examples[40] of GTD11DS and IN-730 and the results indicated good strength recovery.

The rejuvenation process offers a significant cost saving for the operator – especially since a blade may return to service for two or even three times. As previously mentioned, the state-of-the-art single crystal blade (e.g. CMSX4) may not be repaired. The issue is that any heat treatment such as rejuvenation (and indeed welding), may destroy the blade because of grain growth. A detailed discussion related to advanced single crystal blades are outside the scope of the present discussion. It should not, however, be impossible to develop the advanced gas turbine materials to the point where repair is possible.

[40] Liburdi et al., "Practical Experience with the Development of Superalloy Rejuvenation", ASME GT2009-59444.

The figure below shows an Inconel X-750 comparison of a: new (pristine), service exposed, two variants of heat treatment, HIP and heat treatment in combination. The X-750 material is today obsolete in terms of gas turbine usage but the example is instructive. More modern materials are discussed in the literature, such as the mentioned paper by Liburdi.

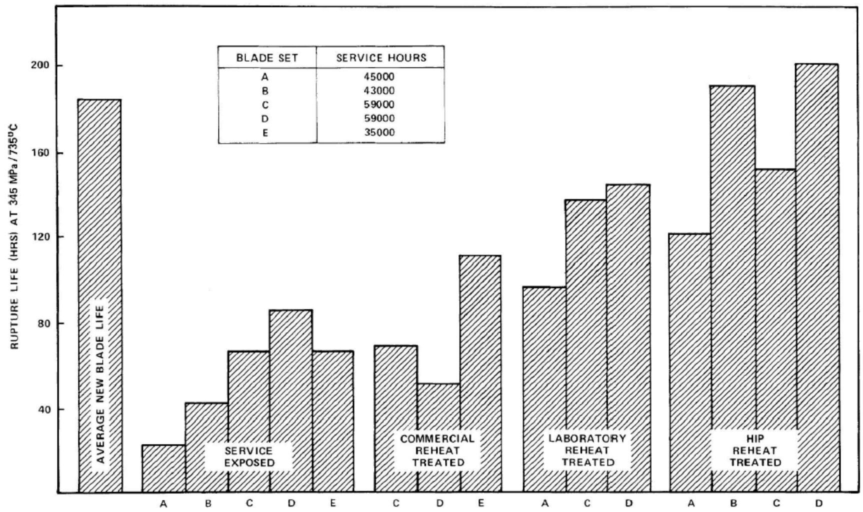


Figure 7-6. Rejuvenation of Inconel X-750 (Courtesy of Westinghouse)

The test showed that the combination of HIP and heat treatment was able to restore the strength of the X-750 material.

7.3.6 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 choose 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 choose 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.

7.3.7 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[41]. 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).

7.3.8 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

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

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 Evaluation, 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 7-6. 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 mass flow of 640 kg/s translates into 522 m3/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 tons.

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.

7.3.9 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[42] 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.

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

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 occur 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!

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.

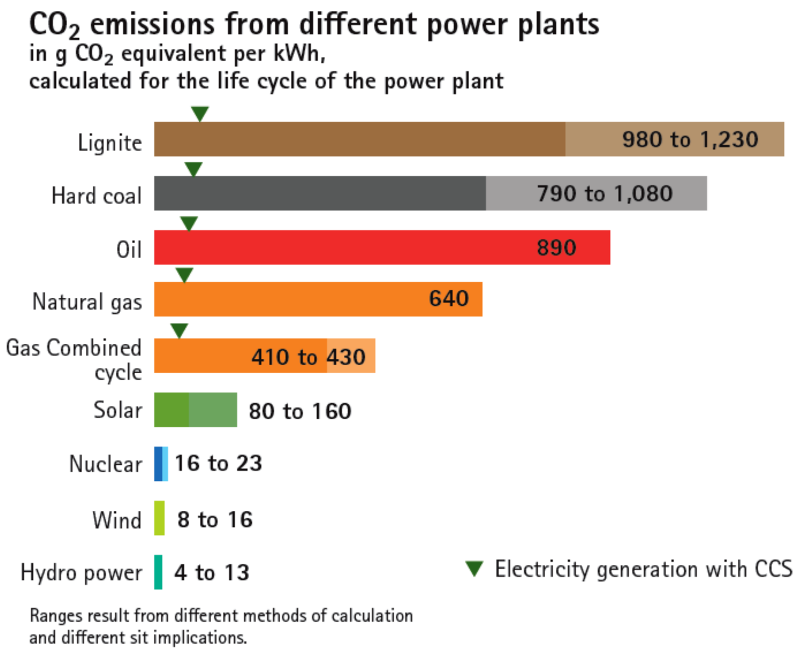


Figure 8-1. Carbon emission from different technologies (VGB[43]).

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.

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

[43] VGB, Facts and Figures 2014/2015 – Electricity Generation

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[44] 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-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

[44] 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.

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[45]. 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).

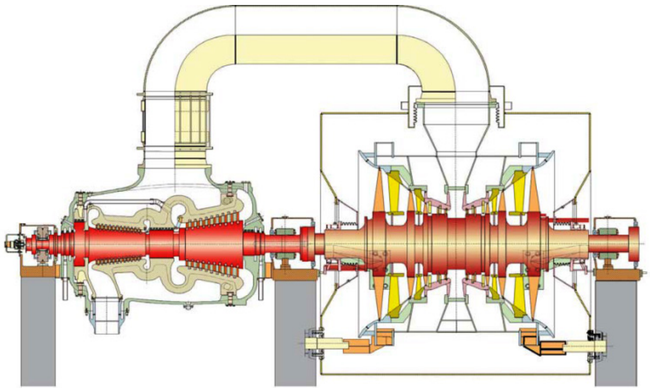


Figure 8-2. Typical combined cycle steam turbine (courtesy to Siemens)

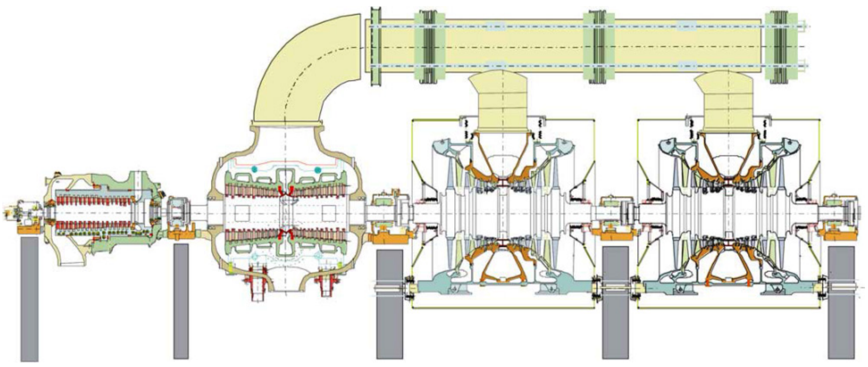


Figure 8-3. Typical coal fired power plant steam turbine.

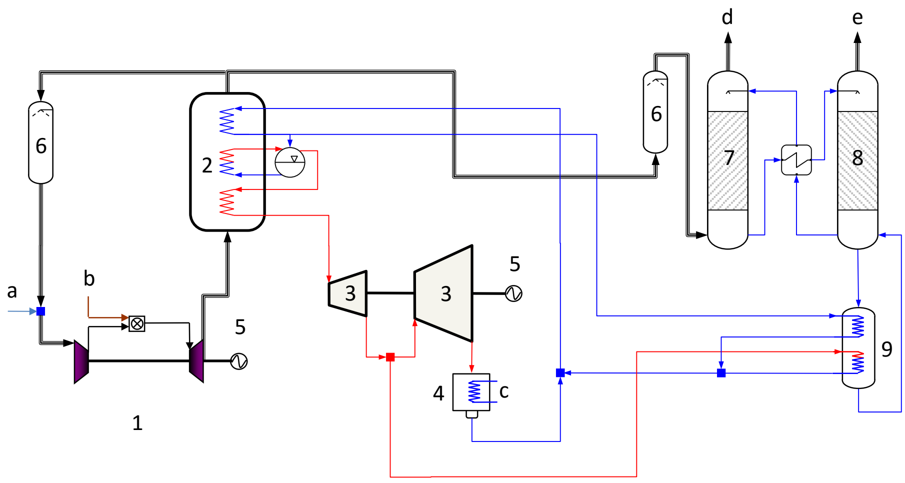
8.3 IMPACT ON PERFORMANCE

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.

[45] 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.

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[46]. 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 8-4. Economizer-reboiler coupling for optimum performance.

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

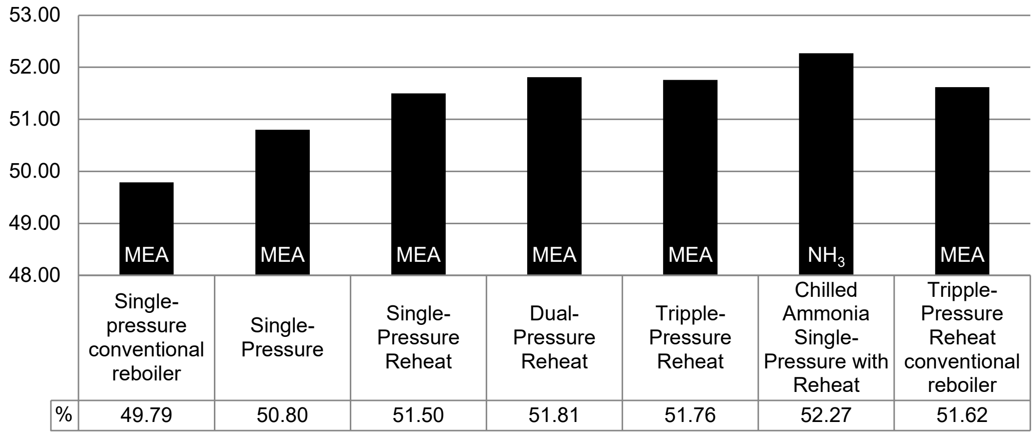


Figure 8-5. Performance impact from different CCS concepts (datum 59.6 percent excluding step-up equipment and parasitic consumption)

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

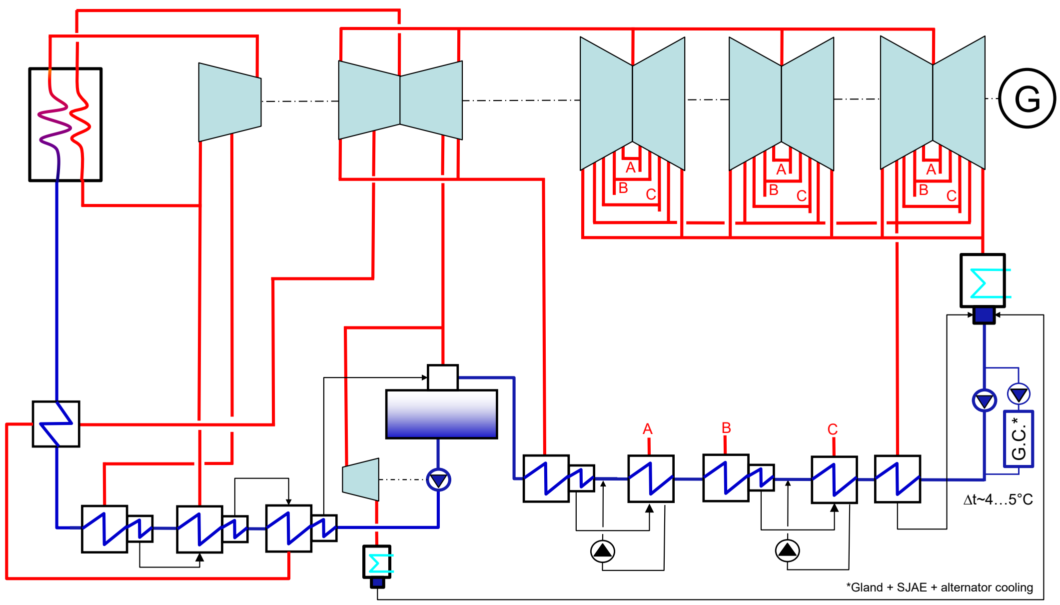


Figure 8-6. Advanced steam plant

Each preheater need about 3…5 percent[47] extraction flow and the exhaust loading can be increased by simply closing the first low-pressure heaters.

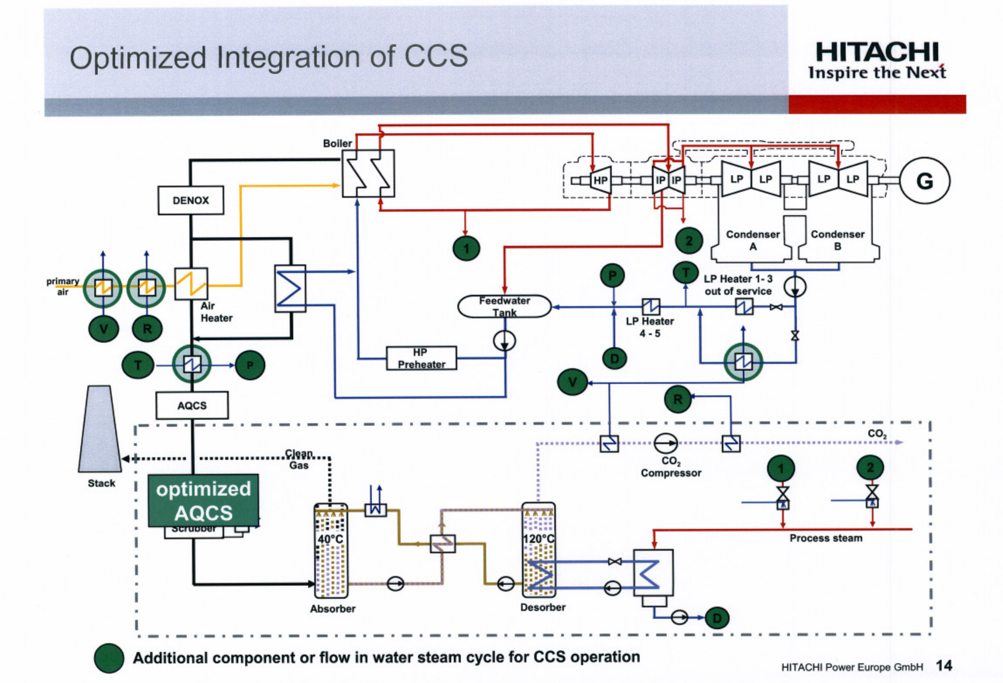


Figure 8-7. CCS integration in an advanced Rankine cycle (Courtesy of Hitachi)

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 CO2 as working media and fires natural gas together with pure oxygen in the 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.

[47] Of the admission flow

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.

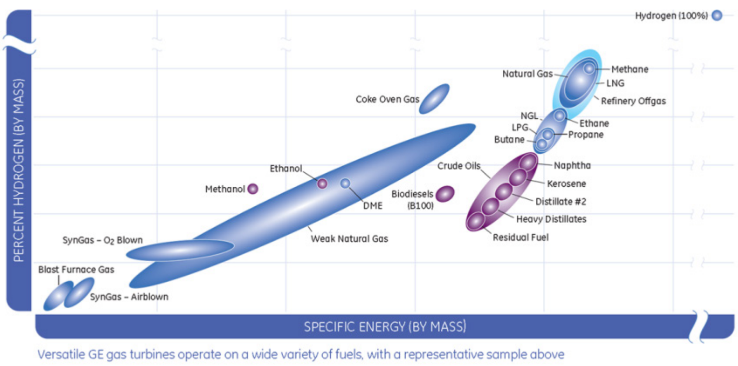


Figure 9-1. General Electric fuel range (Courtesy of GE)

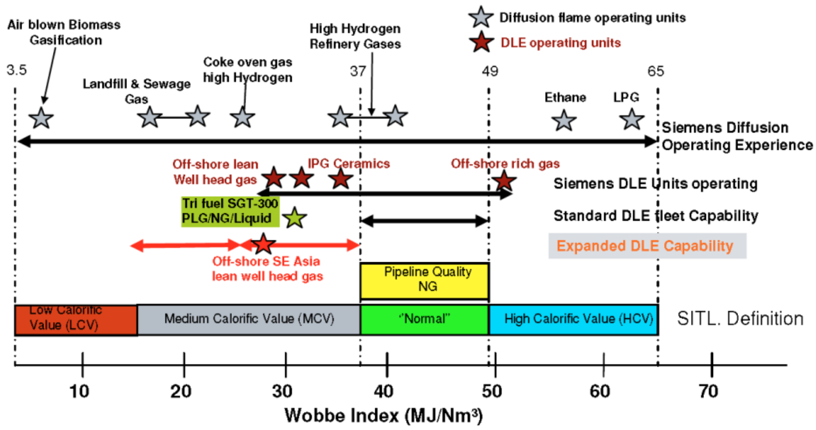


Figure 9-2. Siemens (small/<15MW range) fuel flexibility range (Courtesy of Siemens)

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 CO2-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:

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

Loosely stated, for a certain geometry WITemp sets the amount of energy that can pass through a certain restriction.

Hence, a fuel is capacity-wise 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 (H2S), 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.

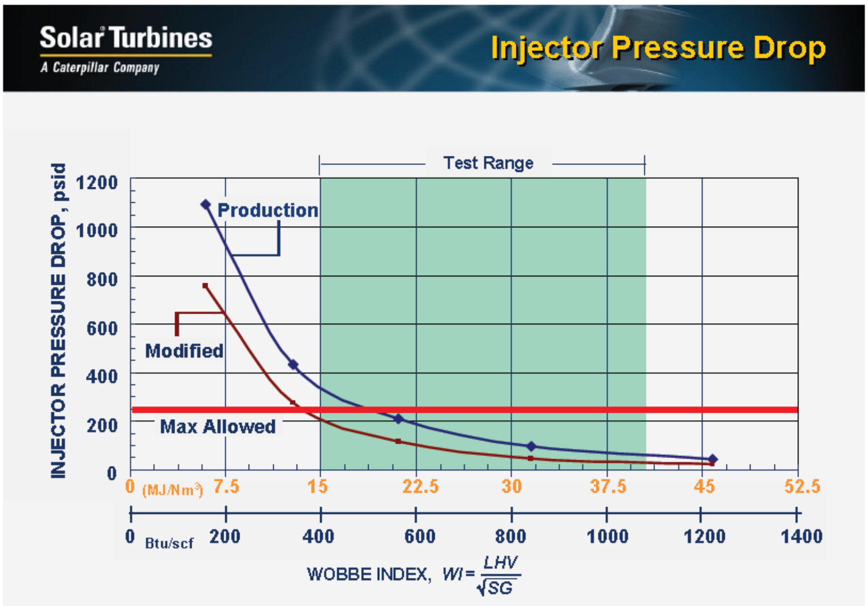


Figure 9-3. Fuel injector pressure drop vs. Wobbe-Index

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.

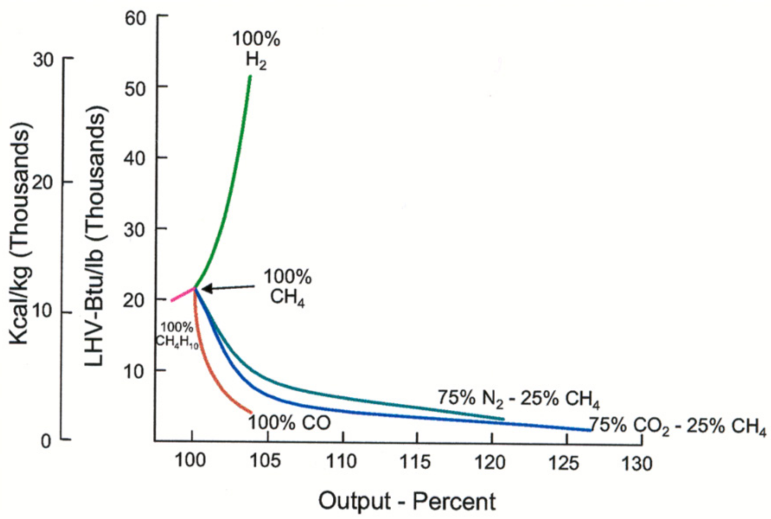


Figure 9-4. GE single shaft performance vs. fuel quality (Courtesy of GE)

Each line on the graph has a curve-linear relative function showing the mixing ratio. For example, half way the line from "100% CH4" to "75% N2-25% CH4" 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

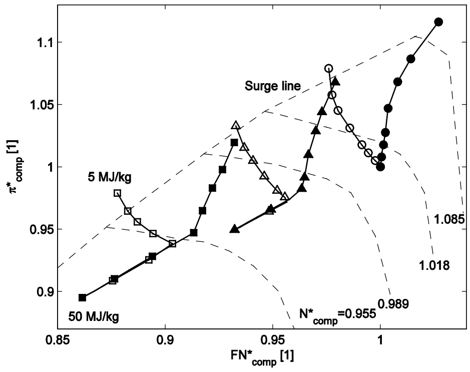
The impact from having a large amount of inert ballast through the turbine has to be analyzed with the specific performance deck[48]. A normal gas turbine has more or less the same flow at the compressor inlet as at the turbine outlet since the fuel flow balances over-board leakage. The engine pressure 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 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

[48] Off-design matching program

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 N2 content whereas an indirect has CO2. An unfortunate effect of the ballast is shown in the compressor map where filled symbols are N2 and the circles are CO2. 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 (N2) 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[49] for low-calorific handling. It is not uncommon for OEMs to have in situ chromatographs for on-

[49] 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.

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.

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 30-40 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.

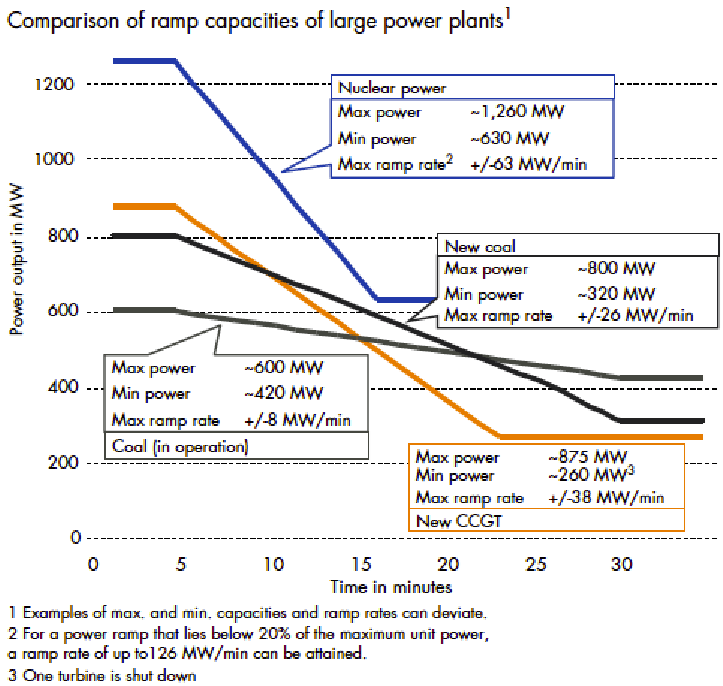


Figure 10-1. Ramp-rates for different production types (VGB)

The best performing "simple cycle" engine (GE LMS100) offers 100 MW within ten minutes at an efficiency of 44-45 percent. The time from barring to synchronization is eight minutes, leaving only two minutes for the loading. 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.

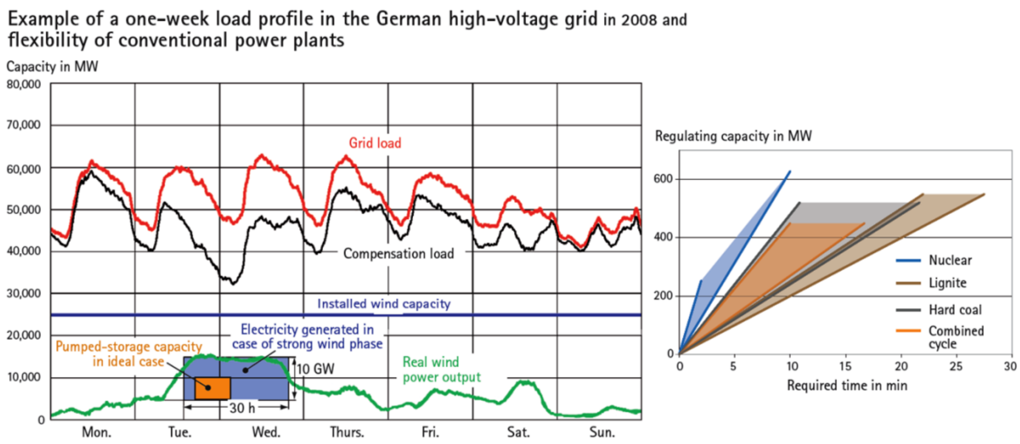


Figure 10-2. The German grid and production capacities (VGB).

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.

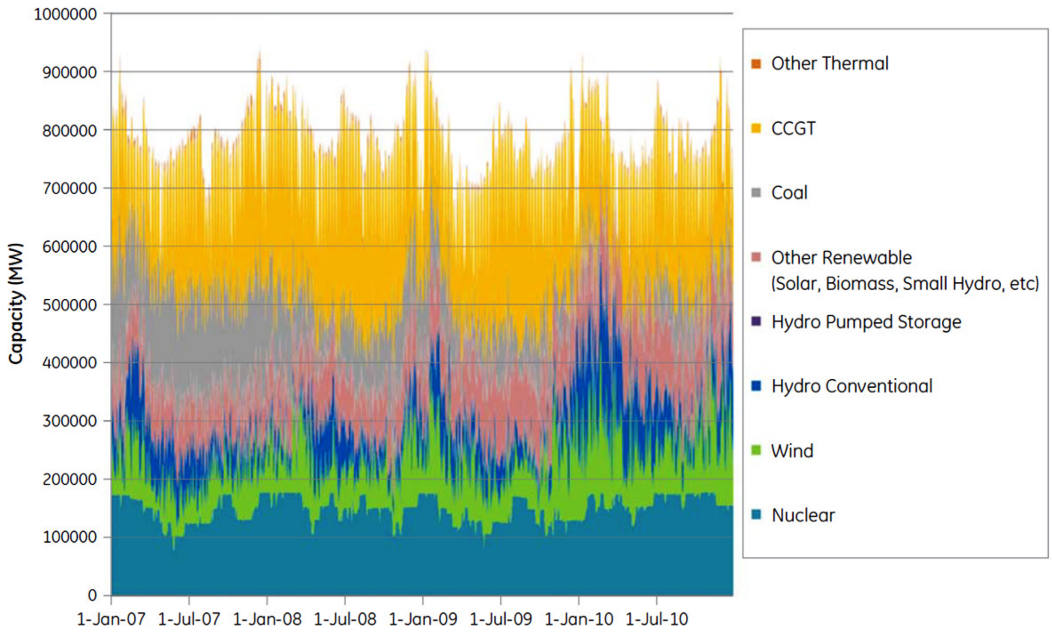


Figure 10-3. Spanish power production mix from 2007 to 2010 (courtesy of General Electric)

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 10-4. Example from Spain 2010 (Courtesy of Klimstra et al., Smart Power Generation, Wärtsilä)

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[50]. 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-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.

[50] 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.

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 45 percent followed by: Pratt & Whitney FT4000 (41 percent) 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.

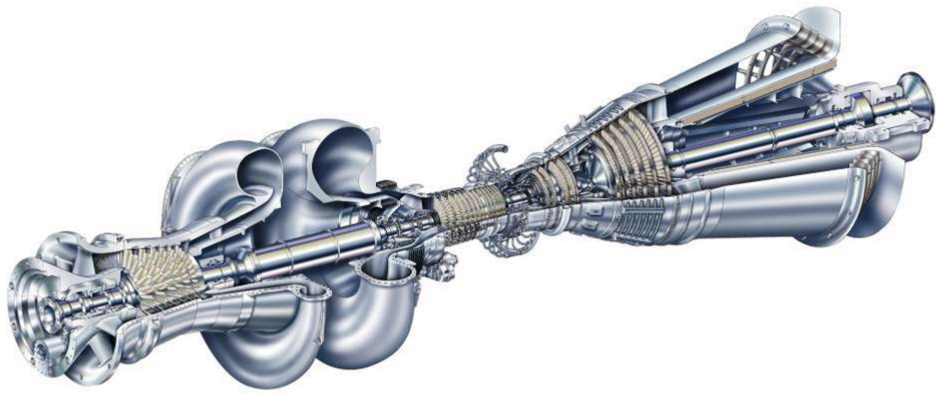


Figure 10-5. General Electric LMS100 (Courtesy of General Electric)

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

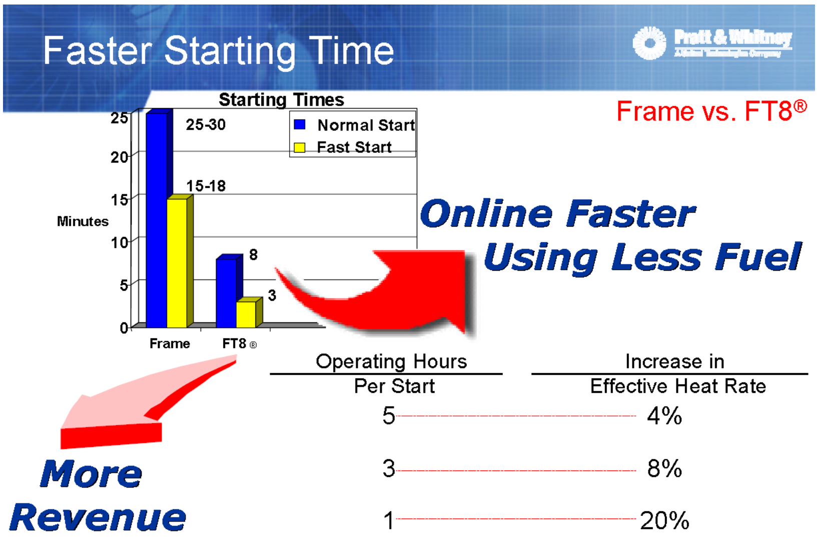


Figure 10-6. Start time influence on engine performance (Courtesy of P&W)

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 Alstom, General Electric, 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[51]. 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.

[51] 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.

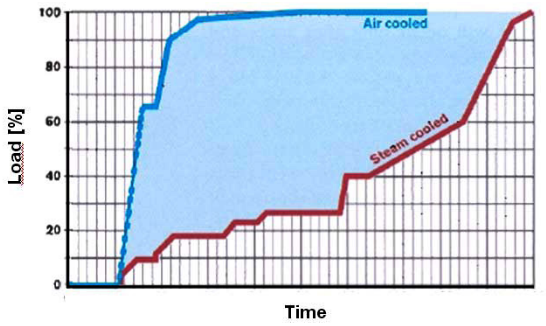


Figure 10-7. Plant start-up times (air- vs. steam-cooled)

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). The mentioned figures are valid only if the turbine is started at the nominal admission temperature. The cure is to use the de-superheater sprays to cool the steam to a temperature that allows the steam turbine to be started with controlled rotor and casing stresses. This approach is clearly visible in the OEMs start-up curves where the power continues to increase after the valve(s) wide open condition (VWO). The start-up time is typically defined from gas turbine barring to the VWO-condition. The load ramp due to the temperature is one order of magnitude less than the ”normal” start.

A classic problem with gas turbines is high thermal stress concentrations between the blade and the platform in the first stage nozzle segments. The thermal stress is a consequence of both the start per se and the combustion system overall pattern factor (OTDF). It is indeed very hard to discuss the gas turbine in general terms with respect to start-up and ramp-rates because all designs have their own set of pros and cons. The hot turbine blading itself is most likely not the most limiting factor and one can expect to find limitations in thermal stress in disks and rotor design features. The latter is even harder to discuss in general terms since there is a plethora of available options with respect to rotor build. The compressor can be bolted with either a single or multi bolts, with or without serrations (like Hirth-teeth etc.) or welded into a single drum. The turbine section is typically built by bolting disks into a turbine package. The bolts may (but not necessarily) carry torque and curvic couplings/Hirth-teeth may be used for torque transfer in addition to the bolts. The same may be the dace for certain compressor designs. Thermal disk stress is probably the most limiting factor with respect to rapid start capability.

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 here is to have a part of the evaporator before the final superheater (or/and re-heater).

10.6.1 Common flexibility issues – the bottoming cycle

Most gas turbines can, regardless of the size, be started within 10-15 minutes and a significant part of the start-up time should therefore be accounted to the bottoming cycle. The actual gas turbine start-up time is in most cases longer due to either HRSG or steam turbine temperature gradient limitations. Some HRSGs also have heat soaking requirements during start-up. Most of the direct gas turbine related issues are described in the chapter seven.

The HRSG design "juggling act" is to find the right balance between hoop stresses (hence creep) versus the temperature gradient induced LCF. The hoop stress level for a certain creep lifing sets the wall thickness. A thicker wall is more vulnerable for the temperature gradients during handling and there is therefore a trade-off between creep- and LCF lifing (see figure 10-8).

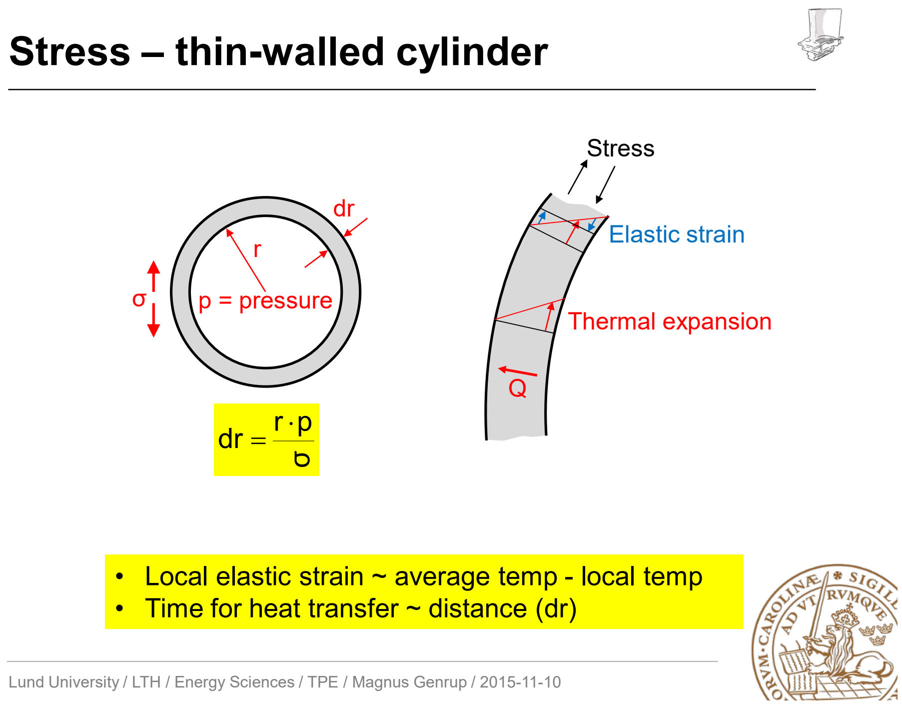


Figure 10-8. Thermal-induced elastic strain

A higher-grade material offers thinner walls for the same creep life, hence less sensitive to thermal stress. A typical operational life and cycle requirement could be summarized as:

- 225,000 hours of operation (30 years with a 15 percent down-time)
- 5 cold-cold starts per annum (or a total[52] of 150)
- cold starts per annum (or a total of 300)
- 20 warm starts per annum (or a total of 600)
- 100 hot starts per annum (or a total of 3000)
- 480 load changes per annum (or a total of 240 cycles per annum)
- 9 gas turbine trips per annum (or a total of 270)
- 5 emergency (forced) cool-down cycles per annum (or a total of 150)

[52] During its 30 year or 225,000 hours life-cycle

The most important feature is to have a stack damper to prevent from cooling down the boiler during barring operation. Most turbines require barring for extended periods after operation. The reason is mainly to prevent from uneven rotor temperature distribution. A barring failure typically renders in a distorted rotor. Issues related to a bent rotor may be vibrations during start-up, blade and central seal rubbing. There are even cases when the rotor seizes in the central casing – effectively prohibiting any barring/operation for several days. The barring speed is typically quite high to prevent from either blade attachment rubbing or ingestions into ”inward” rotor bleeds (i.e. the centrifugal acceleration should be higher than the gravity in both cases). This means that the air-flow can be high with the associate HRSG cooling. A system with a smaller barring by-pass and stack damper is an effective remedy. The damper effectively prevents heat migration and thereby reduces thermal shock on LP sections.

The common perception that the steam drum is the major source for the start-up time requirement is questionable. Alstom has shown that this actually is the case for cold starts but the situation is different for hot starts. The thick drum and its connections to e.g. down-comers[53] and raisers are, indeed, a source for high stress concentrations during thermal transients.

Most state-of-the-art vertical HRSGs with natural circulation today have rather small circulation ratios[54]. The figure used to be 10-12 but can be reduced down to 2.5 (or even lower). One cannot discuss circulation ratios alone since there are several critical factors, such as under-carrying, pressure drops, design-specific details etc. The straightforward solution here would be either to use once-through technology, longer drums or higher grade materials. Both latter approaches result in thinner drum walls, hence less stress due to thermal gradients. A smaller drum would also have thinner walls – but also a reduced retention time. The retention time is defined as the time (at nominal steam production with interrupted feed water supply) from the normal drum level to minimum allowed where the gas turbine and plant trips. The standard time today has been reduced down to three minutes from earlier about seven. Fast-starting units could be pushed even further and figures as low as 1.5 to 2 minutes may be required. A smaller drum is more prone to mechanical carry-over because there is less space/distance (i.e. time) for water separation. The latter issue may pose limitations for the straight-forward solution with a smaller drum in concert with a higher normal water level. The drum itself and down-comer attachments should be close to the saturation temperature – i.e. a function of the pressure[55]. The steam turbine load is directly proportional to the admission pressure (at valves wide open) and the gradient is therefore also a non-linear function of the load gradient. On top of the saturation temperature, the transient itself is damped by the temperature mitigation by the upstream heating surfaces (i.e. the super- and re-heaters).

Drum hogging is another issue that poses high level of stress on the down-comers and raisers. The ”hogging” of the drum is due to different temperatures at the wetted bottom and the hotter upper part – causing different thermal expansion to the upper and lower part. Different expansion causes the entire drum to bend (i.e. hogging) and pose additional tensile stress to the thick piping and welds (especially at the center parts of the drum) – and, indeed, the drum per se.

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[53] A down-comer is typically a large tube that connects the drum to a header (or mud-box) and is an integral part of the boiler circulation system

[54] Ratio between produced steam and circulated water through the down-comers

[55] The saturation temperature can be approximated with the expression: $t(p) \approx 100 \cdot p^{1/4}$

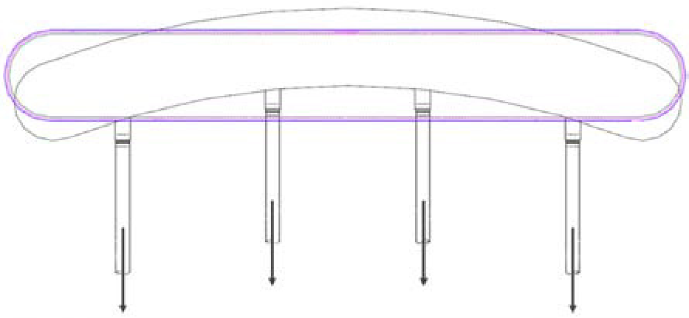


Figure 10-9. Drum hogging (Courtesy of Babcock and Wilcox)

Once-through technology, at the other hand, means that there are no drums (or typically no HP-drum). One could argue that this should be the technology for maximum flexibility. One draw-back is the more complicated cycle chemistry. The normal ”in-volatility barrier” that stops many impurities to enter the turbine is lost. Experience has, however, shown that most issues related to daily cycling manifests them self as stress problems at the super- and re-heater header (commonly called HARPs) attachments. These components are located closest to the hot gas turbine exhaust and experience the steepest ramps during the transients. The headers are typically thick-walled distributor tubes where the relative thin screens (or tubes) are welded into drilled holes. The super- and re- heaters also experience the highest stationary temperature levels since they are located at the HRSG inlet. Full penetration welds offer a solution to some of the issues.

- In summary - There are several options available for avoiding thermal fatigue issues in the steam generator:
- Stack dampers for maintaining boiler energy during barring operation
- Once-through technology (i.e. no HP-drum)
- The FastCirc™ separator by Babcock and Wilcox (i.e. no HP-drum) removes the need for a conventional drum.
- Higher material qualities for thinner walls
- Minimize the retention time
- Longer drum(s)
- Improved designs (e.g. Alstom HARP-attachments)
- Full penetration welds in header attachments

The exhaust temperature is determined by the compressor variable geometry, the firing level and a certain maximum level. The gas turbine exhaust temperature is maintained at the maximum level by the firing controller – whereas the setting of the variable geometry is used for controlling the power output. The control system is designed for optimum performance rather than maximum HRSG lifing and another strategy could perhaps provide some cure. One strategy could be a start procedure with more or fully opened IGVs for a lower transient temperature level. Such approach would certainly introduce both emission compliance and combustor dynamics issues. The main issue is carbon monoxide (CO) and un-burnt hydro-carbons (UHC) because of the lower temperature level. All modern burners have pilot burners for either low-load operation or flame stability and this feature could provide a cure for very high CO-levels. The word cure is used in a sense that the CO catalyst in the HRSG should be able to convert into ”harmless” CO2. There is, however, no information available in the public domain related to this. Mitsubishi has developed a catalyst that is less sensitive to cycling. The information is, however, at the time of writing limited.

The latest plants by all major OEMs have advanced steam data for high performance and very short start-up times. This is quite contradictory since advanced admission data per se should introduce slower start-up times. Advanced steam data results in thicker casings and slow starts and ramp rates for transient thermal stress. One can show that an increase in admission temperature from e.g. 565 °C to 620 °C should impose a factor of three with respect to start-up time. Both Siemens and General Electric seem to use over-sized sprays for starting the steam turbine at a lower temperature level – hence lower stress levels during start-up. The ”trick” is to have a perfect thermal match between the steam turbine rotor (and casings). The turbine is in most cases well-insulated and the drop in temperature is much lower than in the HRSG.

Typical steam turbine start-up definitions[56]:

Cold start	<160 °C
Warm start	160…450 °C
Hot start	>450 °C

The most critical part is the rotor low cycle fatigue issues due to temperature gradients. A state-of-the-art turbine governor typically has the capability of assessing the rotor thermal transient stress by actively controlling the start-up and loading gradient. The biggest issue is for the system is to actually know the actual rotor temperature at a certain initial time. Most systems utilize the casing temperature measurements as a base for evaluating the rotor temperature.

There are examples of e.g. hot-air pre-warmed steam turbines. The hot air is admitted into the turbine via dedicated valves after the ESVs and exhausted through the vacuum breaker. This strategy is ideal for larger size turbines since the hot-air effectively removes the risk for corrosion since the relative humidity will be negligible. Smaller units could perhaps use high-speed barring for feeding more energy into the trapped steam through ventilation. Another strategy could be to break the vacuum during barring and, virtue of the higher pressure, increase the ventilation work. One can show that the ventilation work scales with the speed cubed and is directly proportional to the pressure in the turbine. One drawback, however, is that the work scales with the radii to the power of five – hence a very large impact on the rear stages.

Most countries have legislation for worker safety (e.g. NFPA in the US) where there are firm rules for a certain number of purge volumes. The reason for having a purge requirement is to vent of combustible gases that may explode during start. The purging sequence takes time since there are both stipulated number of volumes and starter motor ramp capacity. The US NFPA 85-2011 provides a possibility of avoiding the purge if the fuel system can be designed without any possibility of fuel seeping. This

[56] VGB Guideline, “Thermal Behavior of Steam Turbines”

means two or several isolation valves and a de-pressurized and vented part between the isolation valves. *Caveat! The same holds for ammonia that is used in the NOx-catalyst and the same precautions should apply for that system as well.* The reasoning is only valid for a volatile fuel like natural gas and one should be very cautious with liquid fuels. The reason for the difference is that liquid fuel may cause high gas concentration within the turbine, ducting and boiled once evaporated during the subsequent start-up. The full purge cycle takes typically 5-18 minutes and requires as a minimum five volumes. NFPA also stipulates a minimum flow of eight percent of nominal. This means that the normal barring operation is insufficient for purging purposes.

The starter motor itself may have capacity problems with respect to the ramp rate.

One especially harmful case is the purge cycle after a trip (or similar short duration shut-down). Such event may cause condensation within the super- and re-heater tubes and resulting high temperature gradients and associated stress. The cure here is to have HARP-drains with sufficient capacity to prevent from flooding the tubes and headers. The purge flow is both lesser and colder than the flue gas during normal operation and this may cause additional stress.

10.6.2 Siemens

Siemens has introduced the FACY project which is an acronym of Fast Cycling as a response to market requirements for flexible operation. The main approach is the ”Start on the Fly” concept where the steam admission hold points were eliminated and the steam turbine is started as soon as the HRSG starts to produce steam – rather than bypass operation.

The first generation of the FACY lowered the start-up time from about 100 minutes down to less than 40 minutes. On example is the Dutch ”Sloe Centrale[57]” where the two F-class single-shaft turbine trains can be started within 30 minutes.

The main features of the FACY-concept can be summarized as:

1. Stack dampers and auxiliary steam supply for maintaining pressures and temperatures in the main components.
2. A fully automated start-up procedure without ant intervention during hot starts.
3. Optimized components and high-capacity fast-responding steam spray coolers for low thermal stresses.
4. Different optimized start-up concepts for various needs – i.e. lifing versus time requirements. There are three options available for the utility: Fast, Cost-Effective and Normal.
5. Start-on-the-fly – the steam turbine is started in parallel to the gas turbine

The usage of an auxiliary steam supply means that the HRSG can be pressurized and that the gland system can be in operation. The steam turbine barring gear must anyway be in operation to prevent from rotor hogging and this means that vacuum can be maintained over-night. One could argue that barring the turbine at higher pressure could increase the ventilation heat (as mentioned in the previous section), hence having

[57] EdF and Delta

a warmer rotor. The drawback, however, is the risk of having oxygen and corrosion in the turbine. The condensate pumps may also be in operation, enabling the condensate polishing plant to bring the chemistry into the specification faster.

Siemens uses Benson™ once-through technology for the HRSG HP-section for rapid start capacity. One could argue that this choice should reduce the start-up time. Experience has, however, shown that there are other - of equal importance - critical parts in the boiler. Siemens is not consistent since they are using drum-type of boilers in their 60 Hz Flex-Plant™ 10 plant. Both General Electric, Alstom and Mitsubishi are using drum-technology – with the same level of flexibility.

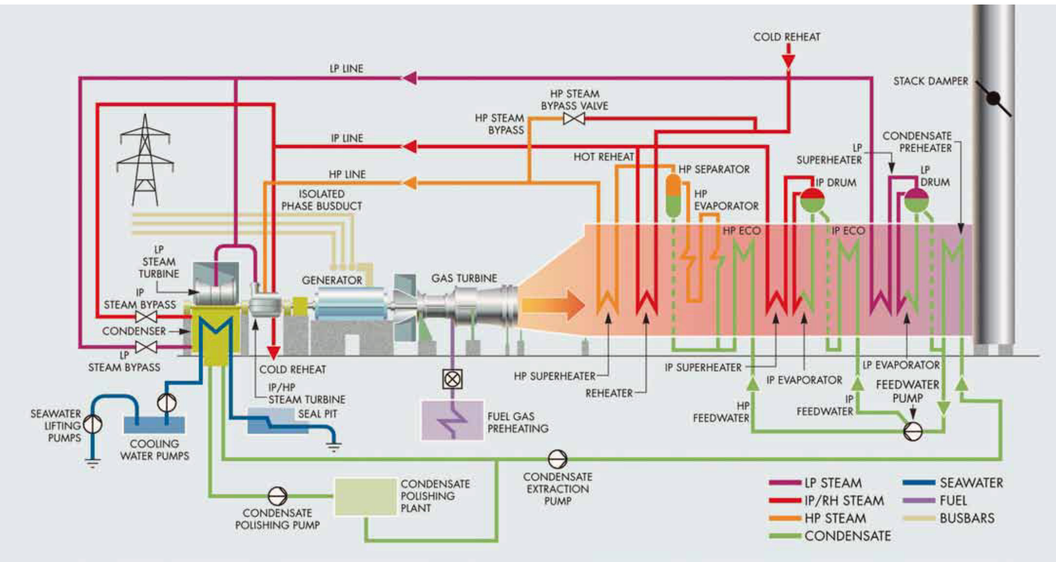


Figure 10-10. Benson™ once-through HRSG (Courtesy of Siemens)

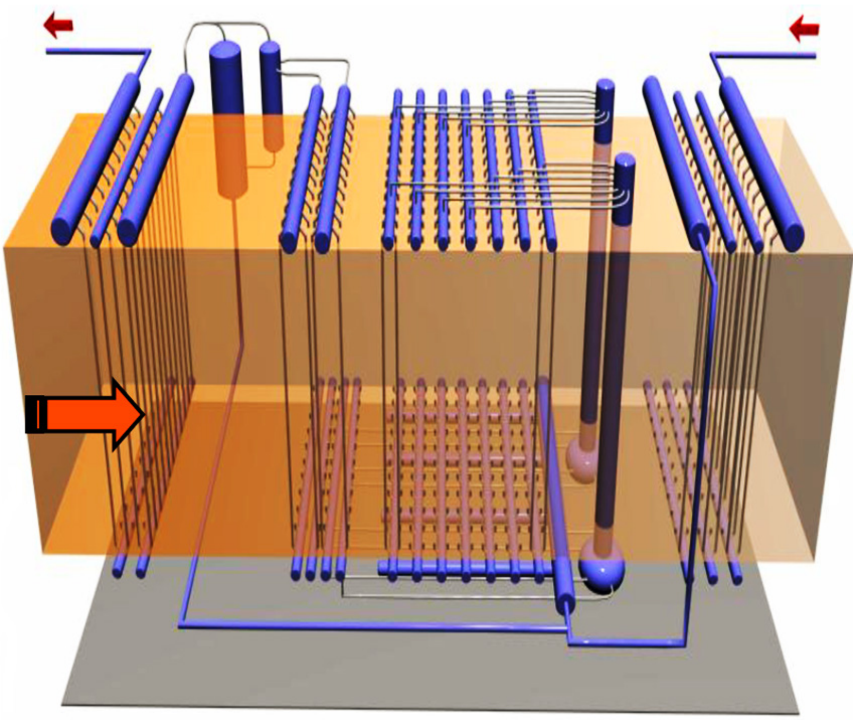


Figure 10-11. Benson™ once-through HRSG (Courtesy of Siemens)

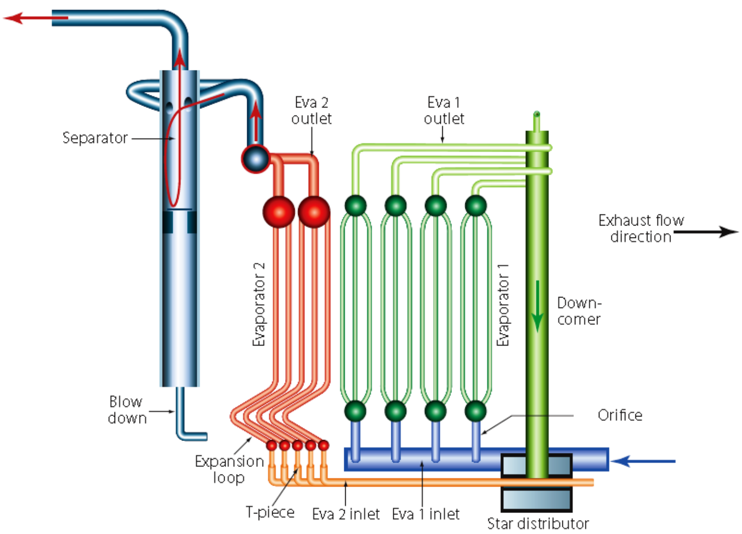


Figure 10-12. Benson™ once-through HRSG (Courtesy of Siemens)

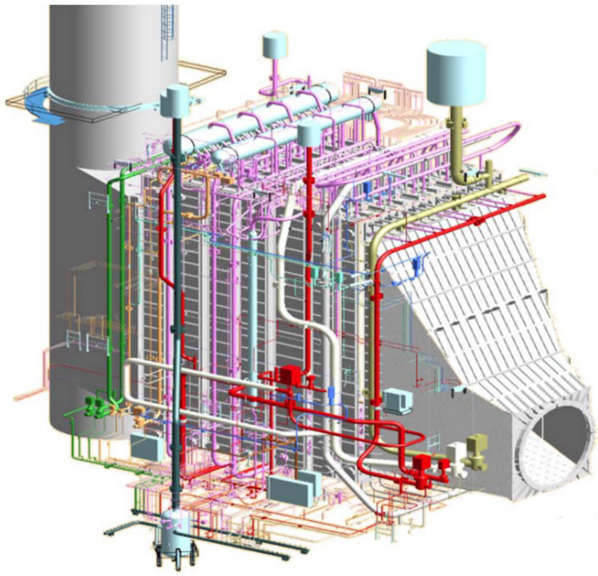


Figure 10-13. Benson™ once-through HRSG (Courtesy of Siemens)

The figure shows a cross-section of the Benson HRSG and the ”classic” start-up and low-load separation bottle is visible to the left just before the final superheater(s). The actual transition from the liquid to the vapor phase takes place within a more or less large section depending on the gas turbine load level. The start-up and low-load bottle could be seen as the ”latest” point for evaporation. The Benson technology differs from the Sulzer-type where the transition takes place the steam /water bottle at sub-critical conditions, similar to a forced circulation boiled (except for there is no drum). One issue with once-through technology applied to HRSGs is the need for a two-phase down-comer, when transferring from the first evaporator section to the second (see figure 10-9 and 10-10). Down-flowing two-phase flow may potentially cause instabilities and even harmful water hammering. This is not an issue for conventional steam boilers because of the tall furnace with continuous upward flow.

The control system is more direct with respect to lifing and empirical start-controllers seem to have been replaced with e.g. stress-controllers. The previous practice was to use a set of start-up and loading curves that were functions of the difference between the admission temperature and the casing. A more modern approach is to control the rotor temperature gradient (hence stress) in a more direct manner.

Siemens has two approaches for rapid start where the operator can choose between ”Cost-Effective” and ”Fast”. The former is still faster than a normal start but poses less maintenance penalties. The choice is driven by the dispatch needs and revenues from the non-spinning reserve asset – i.e. can be boiled down to an economic question.

The ”Start-on-the-Fly” concept enables the turbines to be started in parallel by removing the hold-points for certain process parameters. The conventional way has been to start the gas turbine and hold it on a certain load, with both the HP and LP by-pass open and wait for certain steam parameters, before starting the steam turbine.

The largest 50 Hz machine SGT5-8000H can be started within 30 minutes from hot condition (i.e. after 6-8 hours’ over-night shutdown). The maximum ramp rate is 35 MW/min.

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[58] 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.*

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[59], 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.

10.6.3 Alstom (Ansaldo)

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[60] 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.

Alstom has developed a HRSG concept that is called OCC™ and the acronym stands for "Optimized for Cycling and Constructability". The concept offers both cycling capability and is less labor intense once delivered for site erection.

[58] 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.
[59] Not necessarily the worst component
[60] No available information on base (hot, warm, cold)

The HRSG is a conventional three-pressure level with reheat and is an all-drum type of boiler. The boiler can be furnished with a duct-burner for additional steam production and flexibility.

The main feature of the OCC™ is the stepped thickness design where the tubes are attached in single rows to each (mini-) header. Each header is then connected to the lower and higher manifolds to close the circulation system. This means that the header material thickness can be greatly reduced. Alstom shows that the ratio between tube and header thickness is on the order of three – in contrast to a conventional design where one could expect to find ratios of approximately eight.

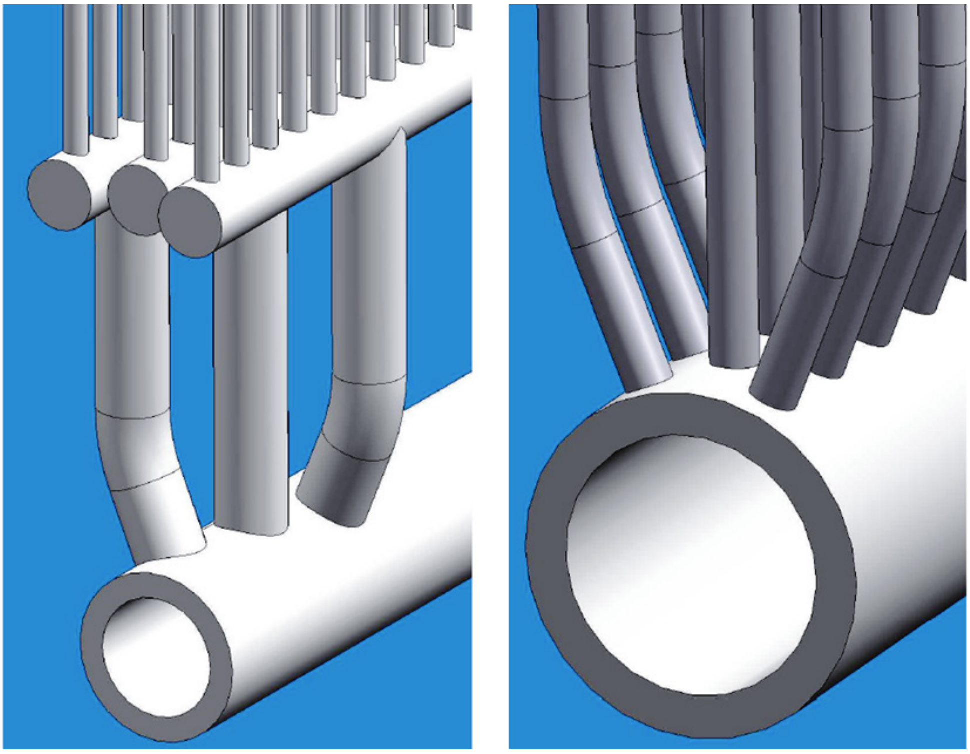


Figure 10-14. Alstom OCC™ design (left) and a conventional multi-row (right). Courtesy of Alstom

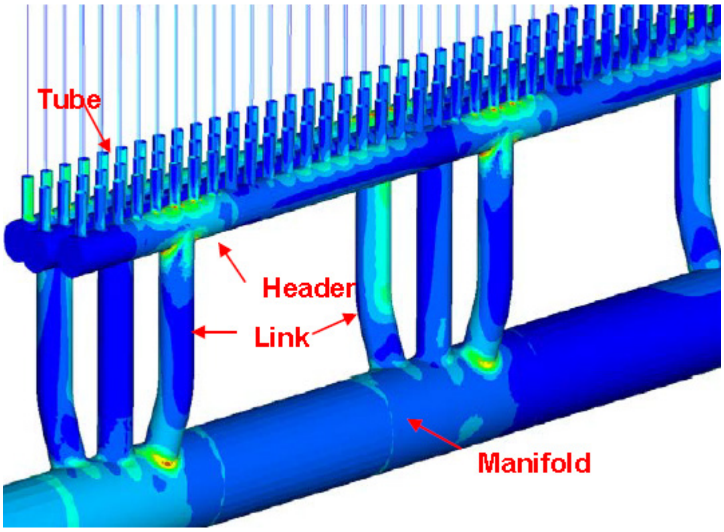


Figure 10-15. Thermal stresses at start-up - Alstom OCC™ design (Courtesy of Alstom)

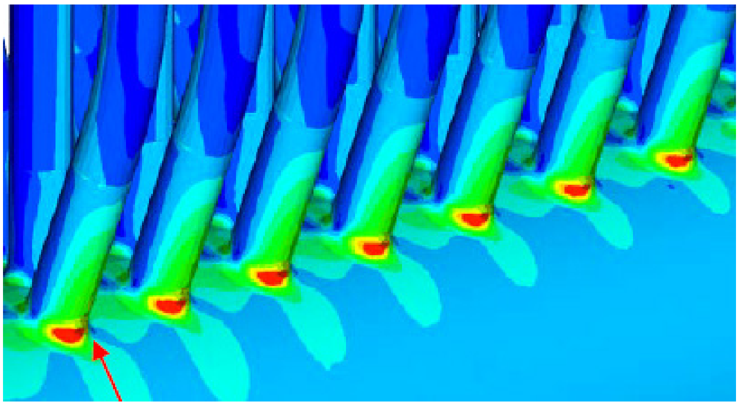


Figure 10-16. Thermal stresses at start-up – Conventional multi-harp design. (Courtesy of Alstom)

The reduced wall thickness also reduces the thermal induced stress by the same amount - hence a factor of three lower level. The single row concept results also means less risk of crack initiation by virtue of the much lower number of welds. A weld is always a potential location for crack initiation and the vulnerability is lower. A single row of tubes (and welds) is much easier to inspect.

The same stepped manifold approach has been used for the risers and the number of drum attachments have been reduced from 72 down to 8 (i.e. a factor of nine). The drum is always shipped to the site for welding and this greatly simplifies the drum erection.

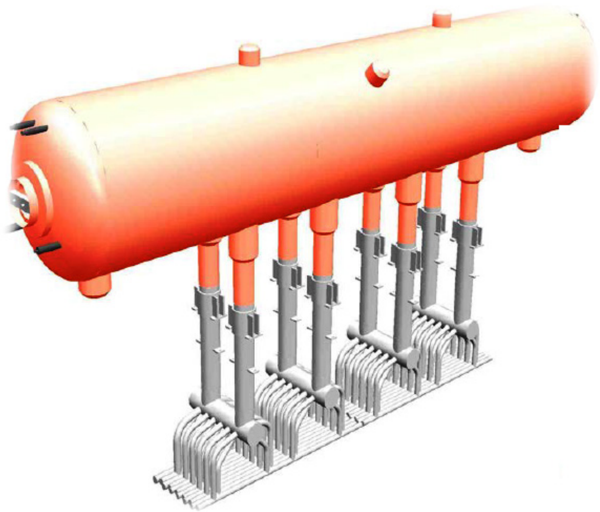


Figure 10-17. Alstom OCC™ drum attachment design (Courtesy of Alstom)

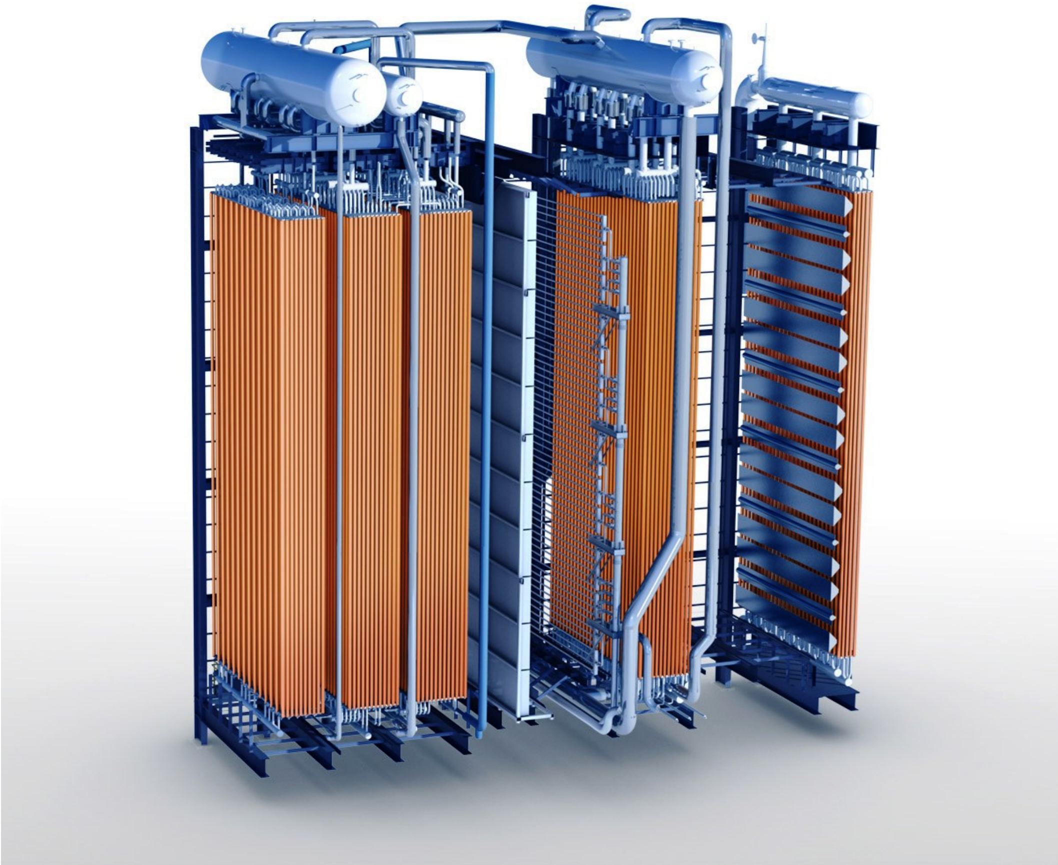


Figure 10-18. Alstom OCC™ general design (Courtesy of Alstom)

Alstom has presented lifing figures where the most critical components are shown. Their analysis shows the same principal features as discussed in the earlier section. Their analysis is based on three operational concepts during an assumed 20 year plant life[61]:

Event	Cycling	Intermediate	Base load
Cold starts	200	200	125
Warm starts	1,000	900	200
Hot starts	4,000	1,000	375
Equivalent hr's	100,000	130,000	200,000

Each start-up cycle has its own set of temperature gradients and detailed modeling shows the most critical components life usage as:

[61] Typically 30 year life cycle for a GTCC

Start-up regime	HP-drum	RH manifold	HP manifold
Cold starts	10%	2%	3%
Warm starts	18%	17%	12%
Hot starts	2%	50%	39%

The Alstom analysis shows that the HP-drum LCF-lifing due to cyclic operation is much lower than the super- and re-heaters in the front end of the HRSG. The presented figures are only valid for the Alstom OCC™ concept and cannot be extrapolated into any other type of HRSG.

10.6.4 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.

10.6.5 Babcock and Wilcox

Babcock and Wilcox has introduced a conventional circulation boiler without a drum. This design combines the benefits from not having a drum with the simplicity associated with a normal circulation boiler. Babcock and Wilcox (B&W) is a prominent boiler OEM company, without gas turbines in their portfolio. The US-based company was founded already in the 19th century by George Babcock and Stephen Wilcox. B&W introduced and patented the first successful circulating water tube boiler in 1856. This was then, indeed, a shift of paradigm since boilers of that era was based on fire tube technology (cf. steam locomotives). B&W has then provided many conventional and nuclear ”firsts” like the USS Nautilus and the NS Savannah. The company is active in many fields of the power industry – ranging from smaller boilers to nuclear reactors and large utility ultra-supercritical once-through boilers.

Babcock and Wilcox has introduced a novel concept where the horizontal drum has been replaced by two vertical separators. The vertical separators could be seen as two partly over-sized downcomers with ”normal” steam-water separation. This approach provides the same functionality as for the conventional drum with respect to time/space for efficient droplet separation and retention time. The smaller diameter also results in thinner walls – hence less prone to thermal stress. The issue with drum-hogging is also effectively solved. At a first glance, the boiler bears a resemblance to a once-through boiler but it is still a circulation boiler. This means that the B&W boiler could be seen as having the benefits from a once-through design without the ”additional” complexity such as tube cooling- and chemistry issues. The former is the efficient cooling effect from having circulation ratios[62] ranging from 3 to 10 evaporator. The latter is the natural impurity ”barrier” in the drum because many impurities like salts etc. have little or no tendency for vaporous carry-over. The efficient separation barrier greatly simplifies the chemistry and makes usage of non-volatiles such as Phosphate-dosage possible. A detailed description of the cycle chemistry is outside the scope of this report and the reader is referred to the specialized power plant literature.

[62] Based on mass fraction

The table below shows the calculated / expected LCF-fatigue life usage factors for the vertical separator and the HP-drum.

Start-up regime	Vertical separator	HP-drum	Design cycles (during 30 years)
Cold starts	1%	10%	200
Warm starts	6%	32%	1,170
Hot starts	18%	140%	4,680
Total	25%	182%	6,050

The presented figures translate into a fatigue life of about 16.5 years whereas the vertical separator would exceed the 30-year design life-span.

The B&W boiler design has the same approach as the Alstom design with single-row harps. This provides an efficient approach for avoiding thick-walled manifolds where the temperature level is at its maximum value – on top of the steep gradients. Their analysis show that cold-start fatigue life usage for e.g. the secondary superheater header reduced to 8.6 percent from 42 percent, when changing from multiple to ”stepped” design. The overall life usage is shown in the table below.

Start-up regime	Single-row	Multi-tube (conventional)
Cold starts	8.6%	42%
Warm starts	2.2%	263%
Hot starts	14.6%	1052%
Total	25%	1,357%

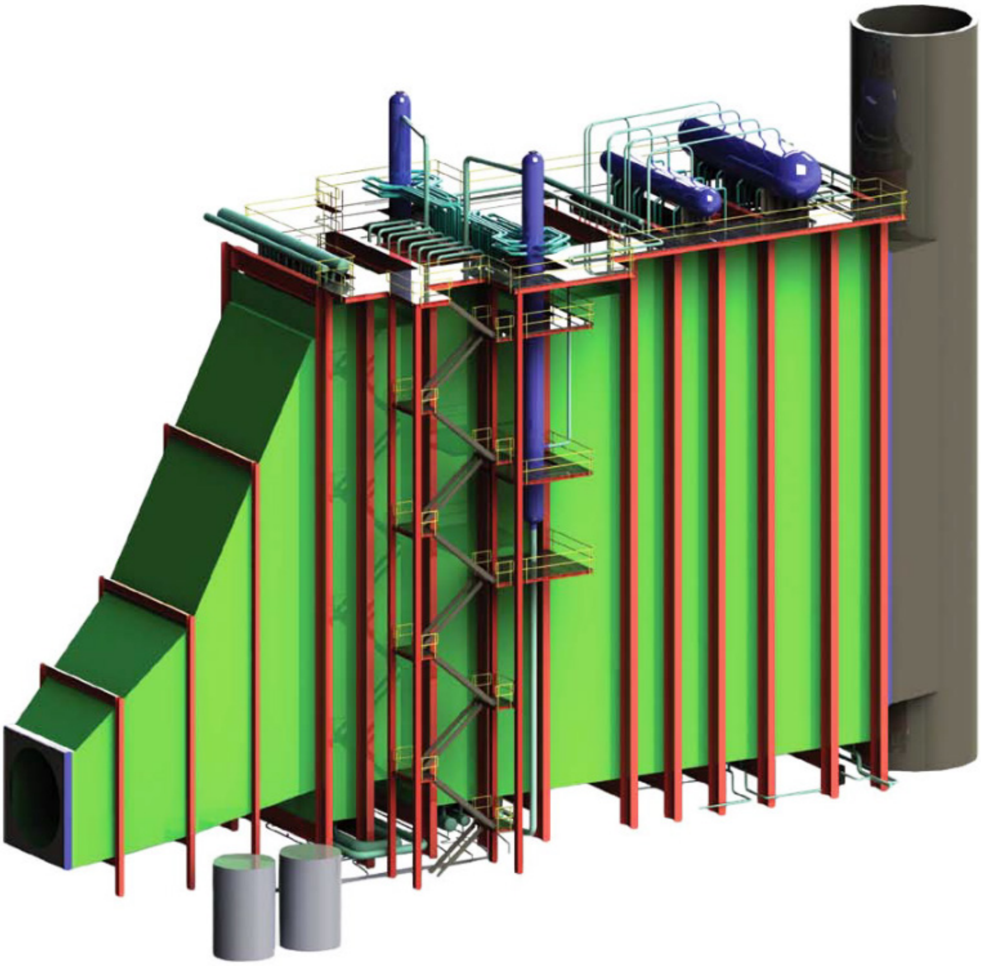


Figure 10-19. Babcock & Wilcox drum attachment design (Courtesy of Babcock & Wilcox)

Figure 10-20. Babcock & Wilcox drum attachment design (Courtesy of Alstom)

10.7 QUICK-FIXES FOR ENHANCED FLEXIBILITY

It is quite hard to discuss in general terms how to enhance the flexibility for a certain plant. Most new plants are offered with a lower than 30 minutes hot-start capability. The key success factors have already been discussed in the previous section and only a brief summary will be given here for completeness.

The gas turbine main limitation is typically the hot-end components, such as the combustor, liners, transition piece (a.k.a ”smiley’s”), first stage nozzle assembly, first rotor and turbine disk. The rotor itself may pose limitation in how fast the engine can be started and loaded. This is mainly driven by the rotor design and different type of rotor bolting, serration, etc. have different characteristics. Some OEMs have welded compressor rotors that will provide some relief, but the turbine disks are still bolded to form the complete rotor assembly. It is therefore quite hard to enhance the flexibility of a certain gas turbine since it is the very core of the engine that has to be redesigned. Most aero-derivatives such as GE LMS100, PW FT4000, RR Trent, GE LM6000, RR RB-211, GE LM2500 et al. have, by virtue of its parent engines, this flexibility from built-in start.

The heat recovery steam generators (HRSGs) have two typical sources for long start-up times, namely: superheater headers and the HP-drum. The former is limited to a handful headers / HARPS and it may be possible to modify the HRSGs to the stepped designs that are discussed in the previous section (cf. figure 10-11). This is probably the lowest hanging fruit when it comes to enhance the flexibility of a certain plant. The steam HP-steam drum is probably not feasible to change for e.g. the novel concept by Babcock and Wilcox due to prohibitive costs and complexity. Another option could be to remove the drum and redesign for once-through. This is, indeed, a major modification and less feasible the previous mentioned approach where the circulation principle is kept. The general perception seems to be that the drum is not the limiting factor for hot-starts. N.B. this is not the case for cold- and warm starts where the drum typically is the biggest limitation.

The steam turbine start-up time is most likely to be limited by the rotor thermal stress. The typical state-of-the art has a quite sophisticated stress controller that sets the appropriate start-up ramp and loading gradient – whilst maintaining the rotors (and casings) lifing. The thermal induced stress is a combination of the temperature, the temperature gradients and expansion coefficients. The temperature level per se influences the stress level and the quick fix is to install over-sized steam coolers in the superheaters and start with de-rated admission temperatures.

In summary, modifications of the superheater headers and large spray coolers seems to be the simplest and less costly way towards enhanced hot-start flexibility.

Appendix I. Previous developments 2012-2014

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

APP. I.1 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.

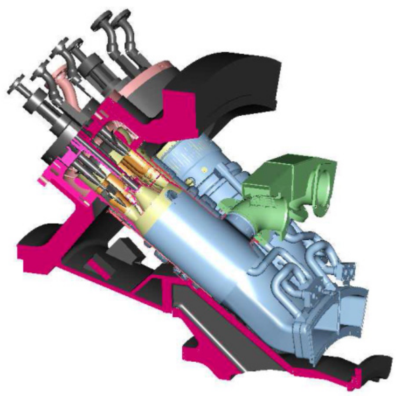


Figure A I-1. MHI steam-cooled liner and transition piece (Courtesy of MHI)

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.

APP. I.1.1 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

APP. I.1.2 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 (i.e. lower at the hub and tip sections) with a cooled liner and transition piece (a.k.a smiley).

APP. I.1.3 AIR COOLED G-CLASS ENGINE (M501GAC / M701GAC)

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.

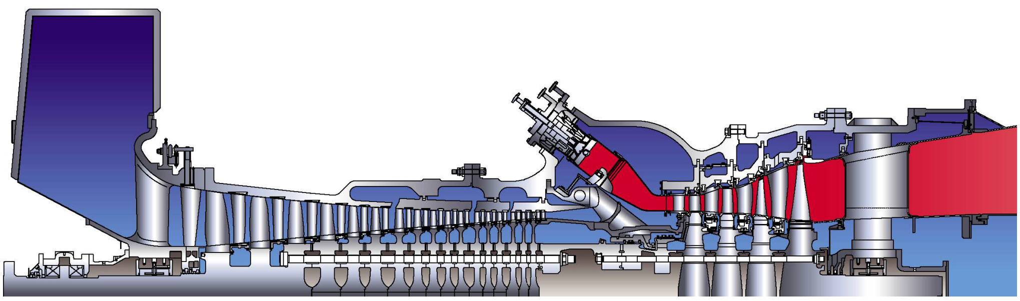


Figure A I-2. MHI M501GAC

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.

MHI has also introduced the M501GAC FAST for daily cyclic duty. The main modification is slightly higher emissions and larger turbine clearances. The latter should have some performance penalty on the gas turbine – that should be at least partly recovered in the bottoming cycle. The gas turbine is started (from ignition) to base load in ten minutes. The gas turbine ramp rate is >50 MW/min, equivalent to 20%/min.

APP. I.1.4 THE 460MW 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 2016 for 60 and 50 Hz, respectively. The 50 Hz 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.

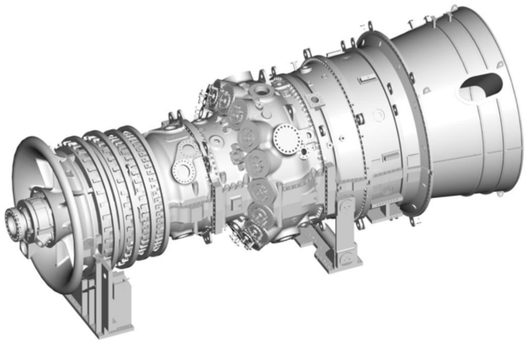


Figure A I-3. The latest MHI 1,600°C J-class engine

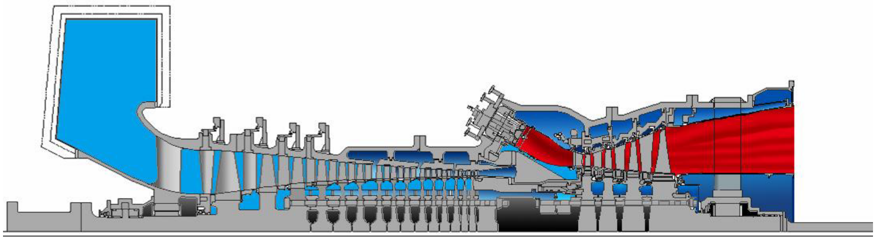


Figure A I-4. Cross-section of the latest MHI 1,600°C J-class engine

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 an H-class 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 TBC'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 10 ppm$_v$ 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 February 2011 and had accumulated (per December 2013) 10,000 actual running hours and 116 starts. The passing of the 8,000 hour mark means that the engine is legible for commercial operation insurance and hence an important milestone. 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. The total sale is 19 units until May 2013.

It was also revealed by MHI (PowerGen Int'l 2012) that they are working on an air-cooled J-class engine. The engine is scheduled for test at the T-point site 2014. The power in simple cycle is 310 MW and 450 MW for combined cycle in contrast to the present 60 Hz rating at 327 and 470 MW for simple- and combined cycle, respectively.

An eventual 50 Hz machine follows the scaling rules with an increment of a factor of $(60/50)^2=1.44$. No further information is available the time of writing.

APP. I.1.5 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 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 at 1,500°C 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.

APP. I.1.6 THE MHI-FLEET

The total number of MHI D, F, G and J-class are 591 units and the sales per December 2012/2013 (in bold) are:

60 Hz	M501J	**17**	M501G	66	M501F	**73**	M501D	26
50 Hz	M701J	**2**	M701G	11	M701F	**124**	M701D	91
	Total	**19**	Total	77	Total	**197**	Total	117

The G-class fleet has steam cooled combustor liners (see previous section for information), 14 and 3 units of the M501G and M701G, respectively are converted into fully air-cooled GAC units.

APP. I.1.7 INDUSTRIAL SIZES

Mitsubishi Power Systems acquired the Pratt & Whitney energy segment (Power Systems) in December 2012. This means that MHI will extend its portfolio into the mid-size range with the units:

- FT4000 Swiftpac 120 MW
- FT8 Mobilepac 25 MW
- FT8 Swiftpac 30-60 MW
- FT8 Swiftpac Combined Cycle 165 MW (49.3 % effici'y)

The Turboden™ organic cycle (ORC) part of UTC/PWA Power Systems was also acquired by MHI. The current power range is 280 kW to 12,000 kW.

APP. I.2 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 or Kraftwerk Ulrich Hartmann[63] (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 shown efficiency (in-situ) in the world currently.

The engine was, when launched, 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.

APP. I.2.1 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

[63] Renamed in 2011

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

APP. I.2.2 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.

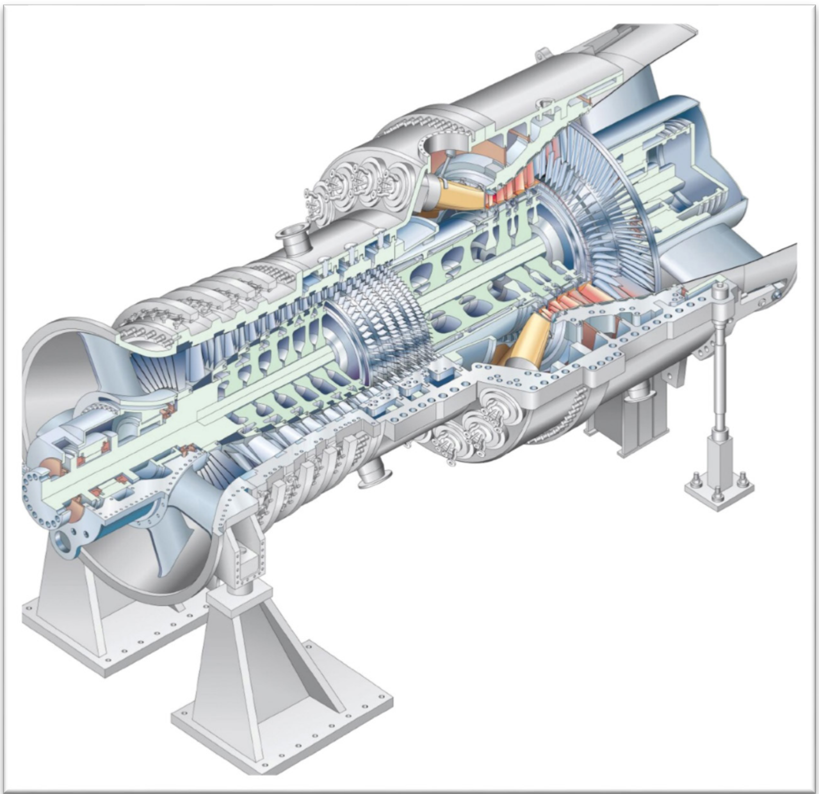


Figure A I-5. Siemens SGT5-8000H (Courtesy of Siemens press service)

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 one to three 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.

APP. I.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.

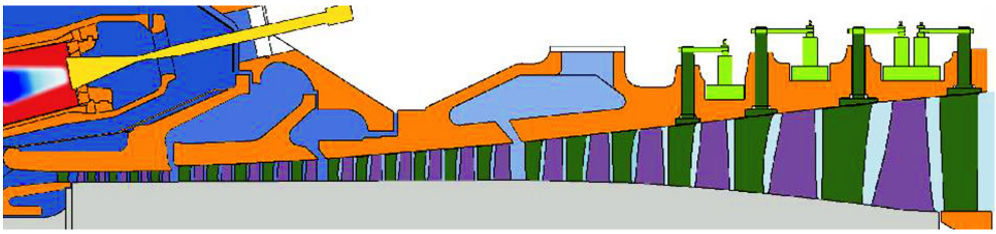


Figure A I-6. Alstom GT26 compressor section X-section[64].

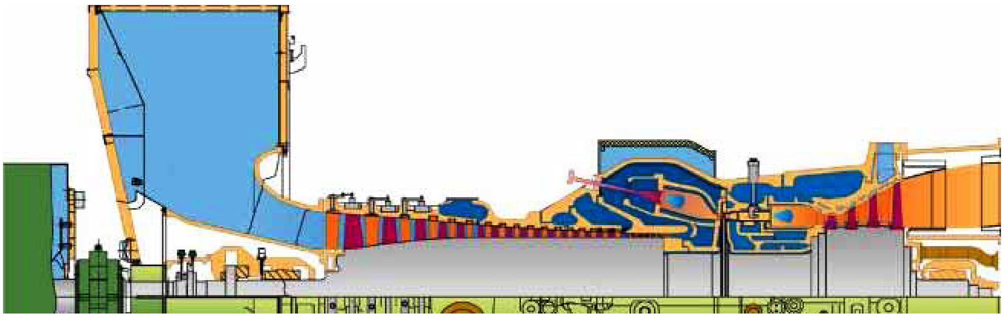


Figure A I-7. Alstom GT26 X-section.

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

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.

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

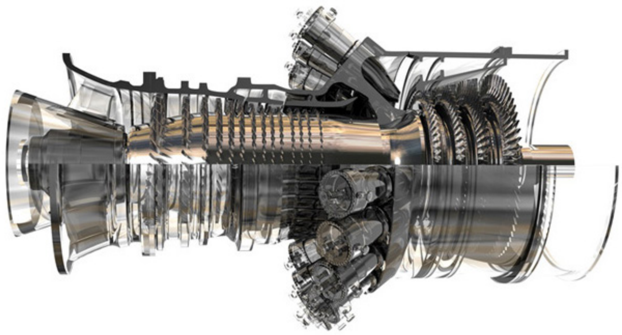


Figure A I-8. General Electric 9FB.05 (Courtesy of General Electric[65])

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). It's 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 ppm$_v$@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.

[65] 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

APP. I.5 HITACHI H80

The Hitachi H80 was originally introduced 2010 and is rated at 99.3 MW with an efficiency of 36.5 percent. The engine was initially designed as a replacement/refurbishment for older General Electric F7E/EA based combined cycle plants. This means that the exhaust temperature and mass flow is kept at the same level as for the mentioned older units. The newest rating is from 2013 and offers 112 MW with an efficiency of 38.2 percent. The firing level has been increased to 1,300 °C from the introduction rating at 1,260 °C.

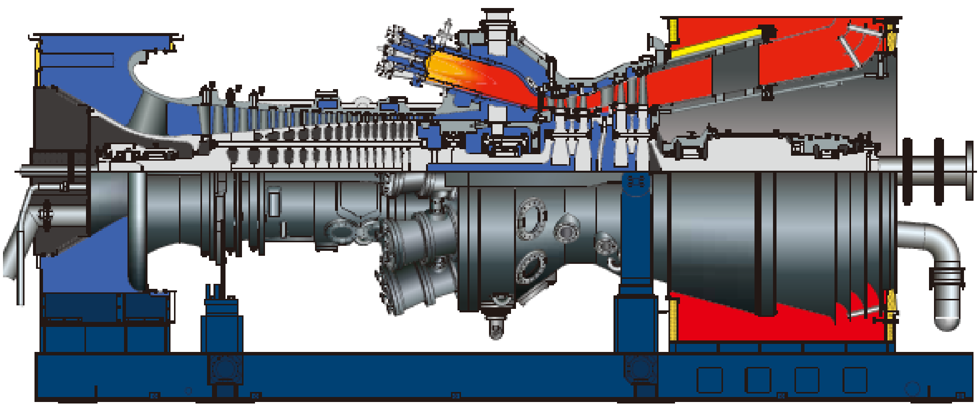


Figure A I-9. Hitachi H80

The Hitachi H80 engine was then (and certainly still is) a quite radical change from typical design practice since it is, despite, of its size a twin-shaft unit. The reason for not having a conventional configuration could be explained by the apparent scaling approach and avoiding a gear due to footprint reasons. The parent H25 unit has a speed of 7,280 rpm and scaling to the desired exhaust mass flow yields a speed level of 4,580 rpm. The H25 parent engine appears to have a fairly standard single-shaft conservative level compressor design. The 3000 min^{-1} compressor flow value is approximately 670 kg/s. The pressure ratio was 17 with a stage count of 17, which translates to an average value of 1.18 – again a conventional single shaft level because of its heritage. Without going into the detailed theory, the compressor speed is more-or-less set by the stress level in the last turbine stage for a single-shaft unit.

The latest rating from 2013 reveals an upgraded compressor with a higher flow capacity and increased pressure ratio. The latter is normally the main reason for increased efficiency and the pressure ratio was increased to 19.3 from 17.0. The level of the increased pressure ratio is approximately 14 percent and this magnitude cannot be achieved by only increasing the firing level and adjusted compressor turbine swallowing capacity. The engine flow is slightly higher at 308 kg/s (from 289), but not as high as one would have expected from a zero-stage. A zero stage would increase the flow capacity by approximately 20 percent and add about a factor of 1.2 to the pressure ratio. It also common practice to have a slightly increased compressor speed in order to restore the velocity triangles in the "old" stages. The zero-stage should typically add some 20…25°C and one would typically counteract by increasing the physical speed in order to maintain the aerodynamic speed. Instead, it appears that Hitachi has high-flowed the compressor front (by either re-staggering or by increased annulus flow path) and increased the rear stage loading capacity.

EIS	2010 (Introduction)	2013
Power output (GT)	99.3 MW	114 MW
Efficiency (GT)	37.5 %	38.2 %
Firing level	1,260°C	1,300°C
Pressure ratio	17.0	19.3
Average stage pressure ratio	1.18	1.19
3000 rpm flow	670 kg/s	720 kg/s
Exhaust flow	289 kg/s	308 kg/s
Exhaust temp.	538°C	546°C
Power output (GTCC)		156/323* MW
Efficiency (GTCC)		53.4/55.3* %

*2+1 configuration

The reasons for adopting a twin-shaft unit are normally stage count and part-speed torque characteristics. The main reason for the reduced part count is the elevated speed level. The twin-shaft unit also has a more favorable off-design performance that is driven by the engine matching. The combustion system off-design emission control is harder to achieve since it is, practically, not possible to control the engine airflow by the variable geometry. The underlying reason is that closing the IGV and VSV introduces incoming co-rotation to the front stages. This means that less work will be absorbed in the front, whilst the engine matching will keep the compressor turbine at its nominal work. The result is, more or less, the same flow at a higher speed level. The only practical means of controlling the unit is by the fuel flow (firing level) and this will have an impact on the exhaust temperature and to some extend the flow (through the compressor speed level). This will imply a rather large hit on the cycle efficiency since both the gas turbine and the steam cycle will operate a lower temperature level. The discussion can get quite involved and the reader is referred to the relevant literature.

The cure, however, is to use a single-shaft engine that offers a very good flow turn-down by means of the variable geometry. A single shaft unit will always operate (loaded) at synchronous speed and the inlet volumetric flow is set by the IGV to a great extent. This means that the firing can be maintained at a high level – at reduced flow. The exhaust temperature is actually lower at part load. It is not possible to operate at nominal firing since the exhaust temperature will be too high. The high-firing strategy reduced issues related to CO-emission at part-load. The single-shaft unit has a less complicated mechanical design with its beam rotor with only two radial bearings and a single trust bearing. One draw-back, however, is the last turbine stress level that typically render in elevated exit Mach number (hence loss). The cure is to use an effective exhaust diffuser for recovering some of the kinetic energy. This is straightforward since the single-shaft engine normally utilizes cold-end drive.

The strategy behind the 2013 Hitachi H80 shaft concept is unclear to the present author. Simply, since it does not incorporate the pros of having a twin-shaft in terms of reduced stages whilst keeping the normal set of cons for a twin-shaft in combined cycle operation. The power turbine first guide vane is cooled and this could also be seen as not follow "common practice". This is driven by the combination of pressure ratio, firing level and the compressor power requirement. This is probably explained by the intent as a "drop-in" replacement for the mentioned GE frames. The power output is in the same range as for the General Electric LMS100 but the efficiency is several points lower.

One could consider to improve the combined cycle operation by introducing a variable geometry power turbine. This feature would add another freedom in controlling the engine matching (read speed) for improved off-design performance. The gain in performance should, however, be balanced with the risks associated with having movable parts in the hot section of a gas turbine. The combination of the pressure ratio and firing level of 1,300 °C offers a rather low steam cycle temperature(s) and this will have an impact on the cycle performance.

APP. I.5.1 COMPRESSOR

The 17 stage compressor delivers a pressure ratio of 19.6 and an approximate flow of 308 kg/s. The mass flow translates into a 3,000 rpm flow of 717 kg/s. This level is quite low for a twin-shaft unit but is probably a result of scaling without a zero-stage.

APP. I.5.2 COMBUSTION SYSTEM

The combustion system is derived from the Hitachi H25 unit and offers 15 ppm@15%O_2. The system uses a diffusion flame pilot and a ring-shaped main.

APP. I.5.3 TURBINE SECTION

The turbine is configured as "2+2" and this follows the typical practice for industrial gas turbines. The compressor turbine uses state-of-the-art cooling and TBC technology. The power turbine is un-geared and operates at either 3,000 or 3,600 rpm for 50 and 60 Hz, respectively. The first power turbine guide vane is cooled and this is not following current practice. The underlying reason is most likely the combination of the lightly loaded[66] compressor turbine and a fairly high firing level.

[66] The compressor power requirement sets the power turbine duty, hence absolute enthalpy drop.

APP. I.6 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.

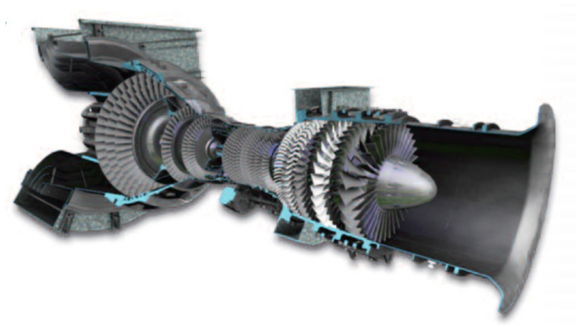


Figure A I-10. Pratt & Whitney FT4000

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.

The Pratt & Whitney energy segment (Power Systems) has been acquired by Mitsubishi Power Systems in 2012. This means that the MHI portfolio will be completed with engines in the 120 MW and downward bracket.

APP. I.7 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.

APP. I.7.1 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[67] 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.

APP. I.7.2 COMBUSTION SYSTEM

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

APP. I.7.3 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.

APP. I.8 KAWASAKI L30A

The latest Kawasaki engine was introduced in 2012 and is rated at 30.9 MW at a shaft efficiency of 41.3 percent. This twin-shaft engine is the largest in the KHI production range. The previous recent KHI engines are all single-shaft units, the L30A is, however, a natural step into the mechanical drive market. A single shaft unit is impractical for such applications due to its torque characteristics. One can show that a geared single shaft would have a lower speed level, than the gas generator, and therefore require a higher compressor stage count. The underlying reason is found in the last turbine stress level and available annulus area. The previous product portfolio was from 0.2 to approximately 18 MW with a total sale of more than 10,000 units. This efficiency level is the highest in its power class. The power turbine speed is 5,600 min^{-1} and a gear box is required for alternator operation. The genset rating is 30.1 MW and 40.1 percent efficiency.

[67] The 3000 rpm value is defined as: $\dot{m}_{@3000} = \dot{m}_{@n}(n/3000)^2$ 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.

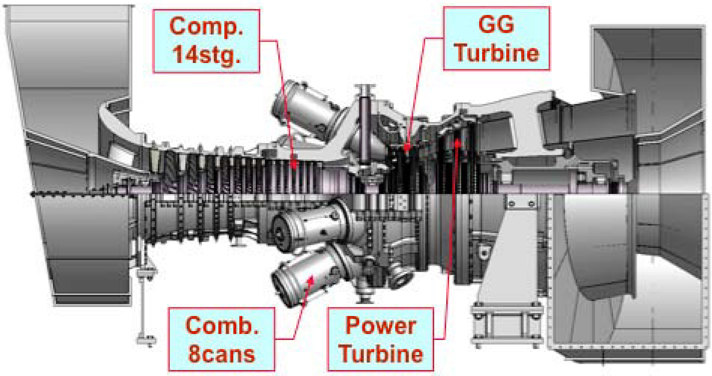


Figure A I-11. Kawasaki L30A

APP. I.8.1 COMPRESSOR

The pressure ratio is 24:1 with 14 stages resulting in an average stage pressure ratio of 1.25:1 per stage. The gas generator design speed is 9,330 min^{-1} and this translates together with a flow of 86.5 kg/s into a specific flow of 837 kg/s (@3000 min^{-1}). The compressor utilizes state-of-the-art three-dimensional blading.

APP. I.8.2 COMBUSTION SYSTEM

The DLE system is based on individual cans and is based on the system that was introduced for the smaller power M7A fleet. The M7A units are capable to run with less than 9 ppm NOx. The L30A has eight cans and is following the common trend with cans rather than annular systems. Cans offers true burner scalability with the same dynamics at the expense of higher pattern factors. This could potentially introduce turbine LCF-lifing issues with large temperature gradients. The figures below show the features of the burner and liner and test data from a combustor rig (i.e. not in the engine). The very low level is a result of the two main stages (main and supplemental).

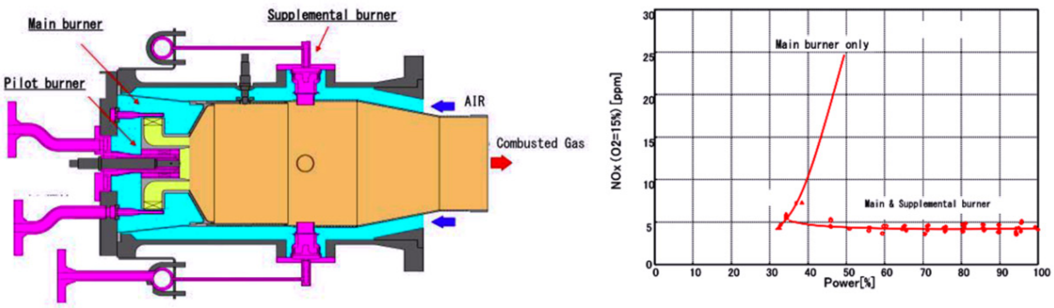


Figure A I-12. Kawasaki combustion system

The engine test shows that the engine is capable of running at less than 15 ppmv NOx over a 50 percent to 100 percent load range. The corresponding CO emission over the same load range is 25 ppmv.

APP. I.8.3 TURBINES

The turbine section has a conventional two and three stage for the compressor- and power turbine, respectively. The choice of 9,330 and 5,600 min^{-1} explains (or drives) in the stage configuration and the turbine inter-duct. A properly designed turbine inter-duct should have a quite low loss level. The L30A has the first power turbine nozzle into the duct (i.e. negative hade) and this feature should reduce the loss. The compressor turbine has un-shrouded blades for small running clearances. The power turbine blades are shrouded with inter-locking for both performance and vibration-free operation over the speed range. The design speed range is 2,800 min^{-1} at 50 percent load to 5,880 min^{-1} at 105 percent load. The maximum allowable speed is 6,440 min^{-1}. The unit has to be geared for synchronous 50 and 60 Hz due to shaft torque limitations.

APP. I.9 GENERAL ELECTRIC LM6000PG AND PH

The newest member of the GE "Flex" series is the uprated LM6000PH engine. The LM6000 engine is a true aero-derivative with high commonality (90 percent) with its parent engine CF6-80C2[68]. The latest upgrade is based on technology and materials from the CF6-80E, GE90[69] and LMS100. The engine is rated in DLE-version (PH) at 47,926 MW at an efficiency of 40.7 percent. The exhaust flow is 136 kg/s and the temperature is 476°C. The standard engine start-up time is ten minutes but can be reduced down to as low as five. The maximum load ramp rate is 50 MW/min.

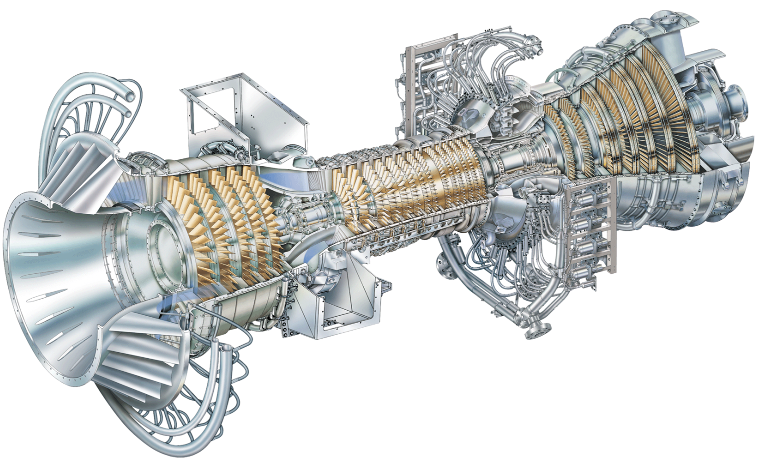


Figure A I-13. General Electric LM6000PD Sprint™

The GE LM6000 has a twin-shaft compound configuration where the low-pressure compressor is driven by the low-pressure turbine at constant speed for power generation. The engine can be either cold- or hot end drive depending on the application. The inner spool is "free" and the speed is varying with the load. This arrangement requires six variable stages on the high-pressure compressor for stability over the load range due to the fix LPC-speed. The low-pressure compressor has a single variable stage for start-up compressor stability and mass flow control.

The introduction of the CF6-80E technology resulted in a higher firing level for both higher power density and recoverable exhaust energy.

[68] Boeing 747
[69] Boeing 777

APP. I.9.1 COMPRESSOR SECTION

The LPC-compressor has five stages and it is only the first that is variable. The compound configuration means that the LPC is spinning at constant speed (3,930 min^{-1}) and the mass flow can therefore be controlled for fast start-up. The HPC has 14 stages where six are variable for part-speed (or load) operation. The total pressure ratio is 33.4.

APP. I.9.2 COMBUSTION SECTION

The engine has either DLE-technology or conventional SAC with water injection.

APP. I.9.3 TURBINE SECTION

The two-stage high-pressure turbine follows GE practice with a cylindrical un-shrouded design. The low-pressure turbine has five shrouded stages.

Appendix II. 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.

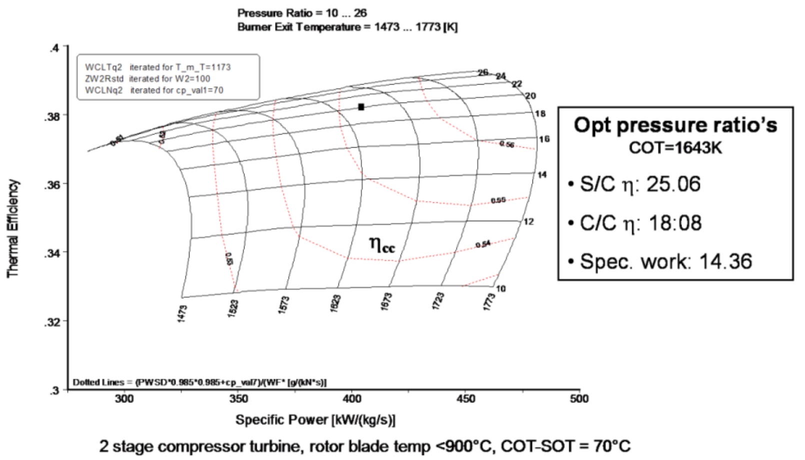


Figure A II-1. Example of cycle performance

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[70] and packing flows[71]. 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. II.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).

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

[70] 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).
[71] Packing flows are used for preventing from hot-gas ingestion and disk face cooling (design dependent)

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.

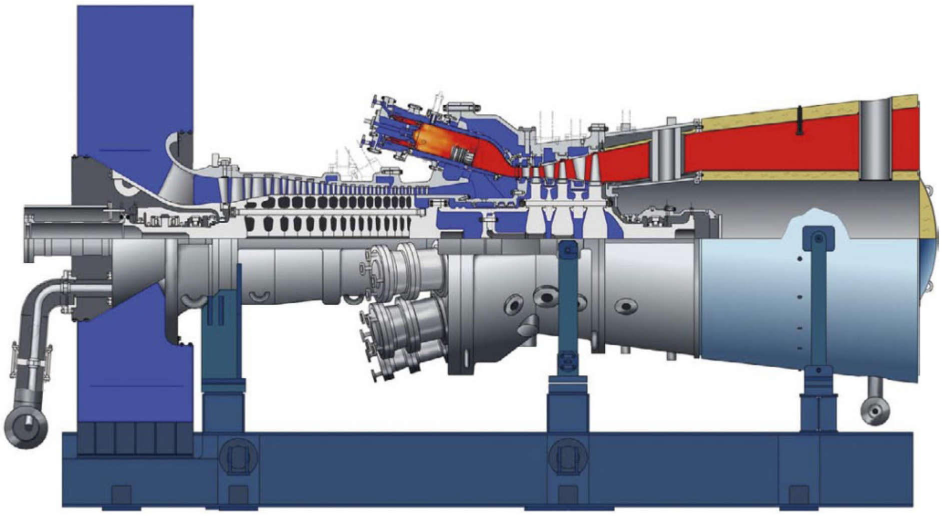


Figure A II-2. Hitachi H25 (Courtesy of Hitachi)

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.

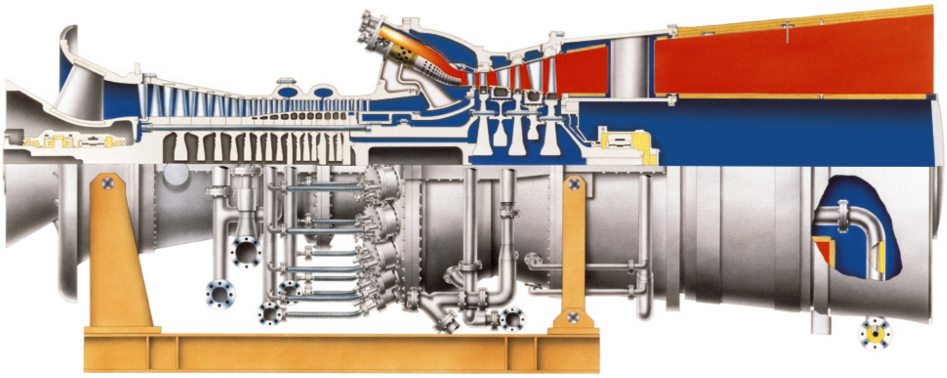


Figure A II-3. General Electric Fr7/9 Gas turbine

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 publication showing a QFD analysis[72] 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).

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

Based on General Electric's GER 4194 Eldrid et al.

Lund University / Energy Sciences / Thermal Power Engineering / Magnus Genrup

Figure A II-4. 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. II.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

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

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.

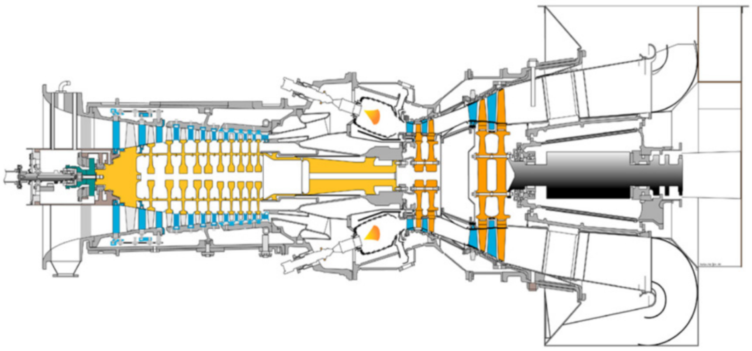


Figure A II-5. Example of an industrial twin-shaft gas turbine (Siemens SGT-700)

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. II.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.

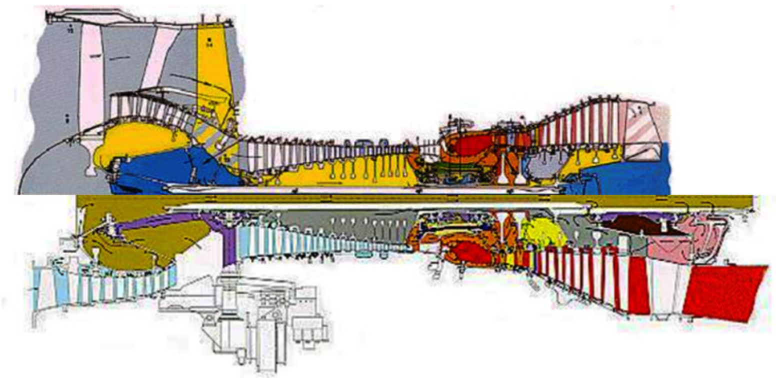


Figure A II-6. CF6 aero-engine and derivative LM6000

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.

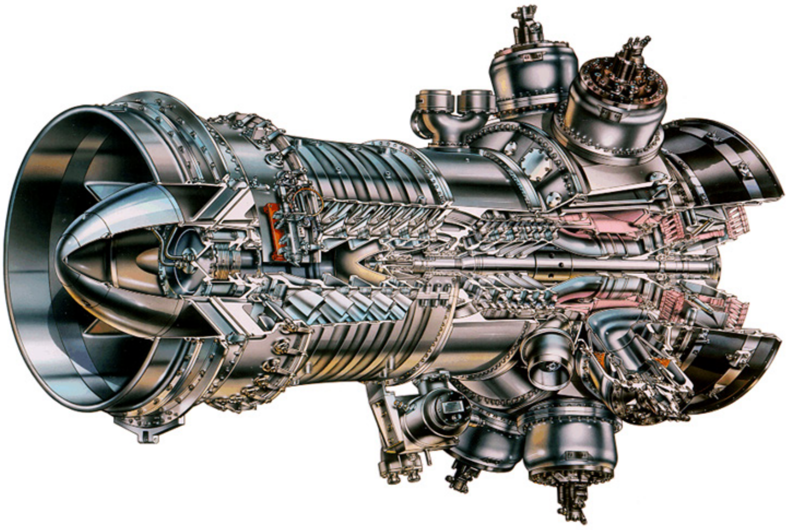


Figure A II-7. Rolls-Royce RB211 gas generator

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.

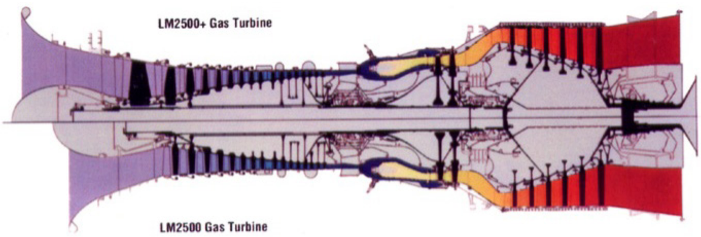


Figure A II-8. General Electric LM2500 vs. LM2500+ (Courtesy of General Electric)

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.

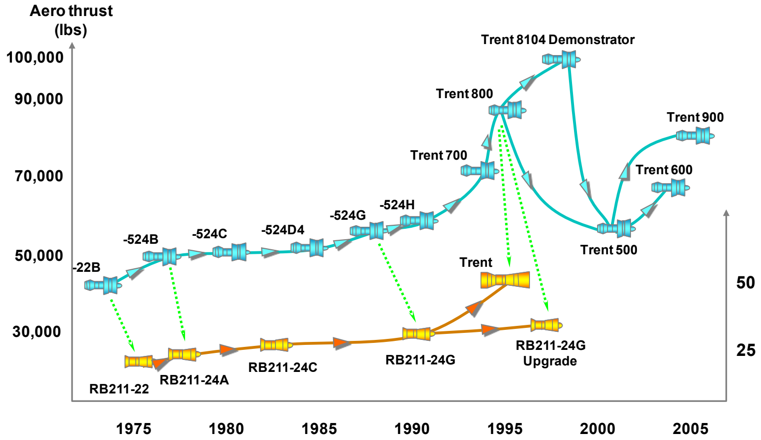


Figure A II-9. Rolls-Royce aero-derivatives and parent engines

APP. II.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.

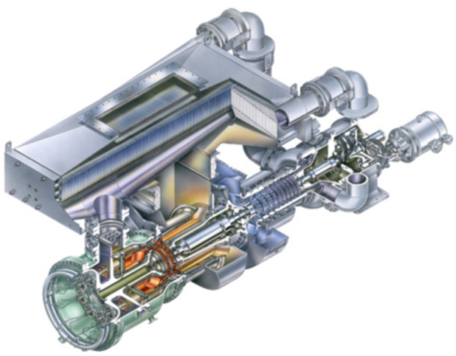


Figure A II-10. Solar Mercury 50 recuperated engine.

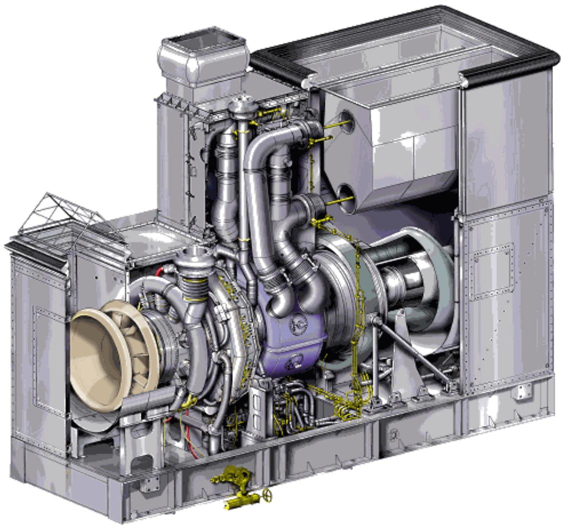


Figure A II-11. Rolls-Royce WR-21 ICR (Courtesy of Roll-Royce)

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[73] 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[74] 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 has been presented at European Turbine Network (ETN) in 2006 (and web published) and is showing a comparison between different cycle options.

[73] Rolls-Royce WR-21 has both inter-cooling and exhaust heat recovery with variable power turbine geometry.
[74] The presented numbers are based on GTW. There are claims of 59.5 and the target is 60 percent.

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.

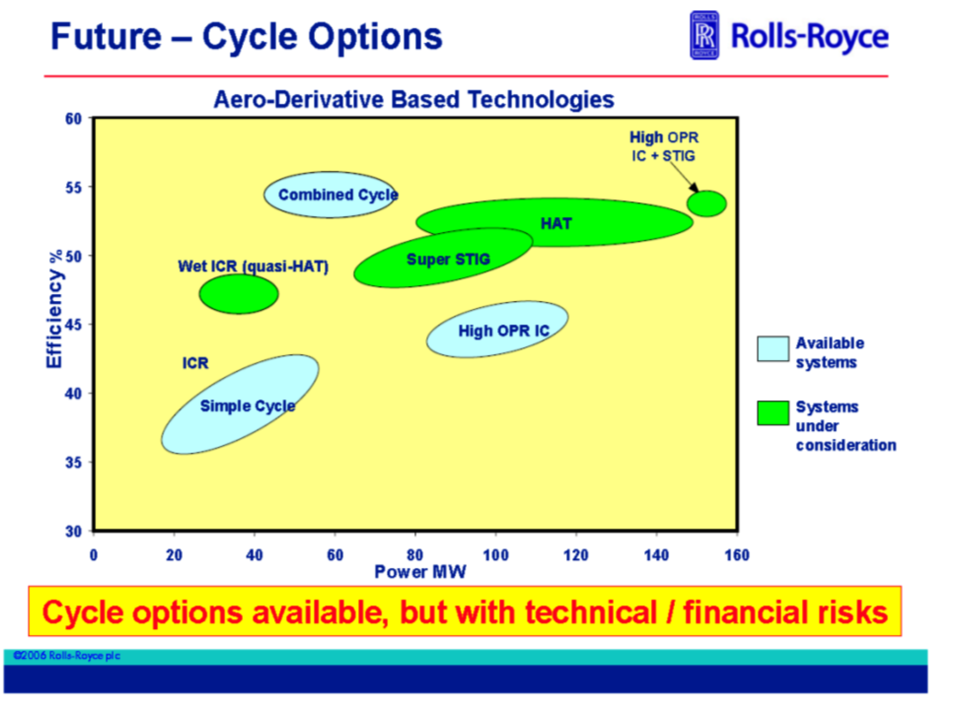


Figure A II-12. Rolls-Royce cycle view

APP. II.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:

1. Increased firing
2. High-flowing or zero staging

3. Linear scaling
4. 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[75]. Aerodynamic speed[76] 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).

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

[76] Reflects compressor velocity triangles based on Mach number

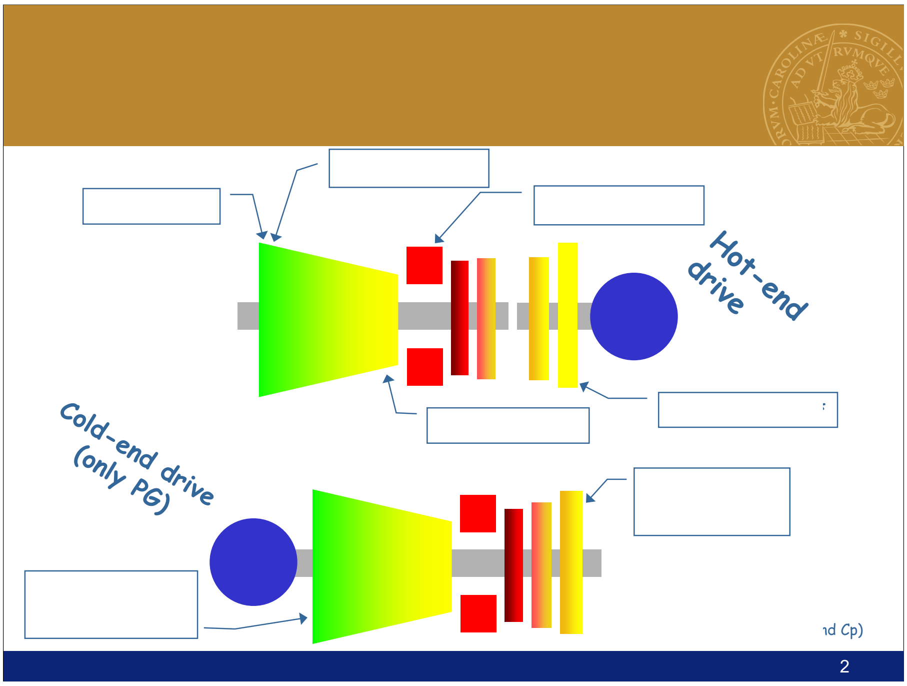


Figure A II-13. Typical evolution steps

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.

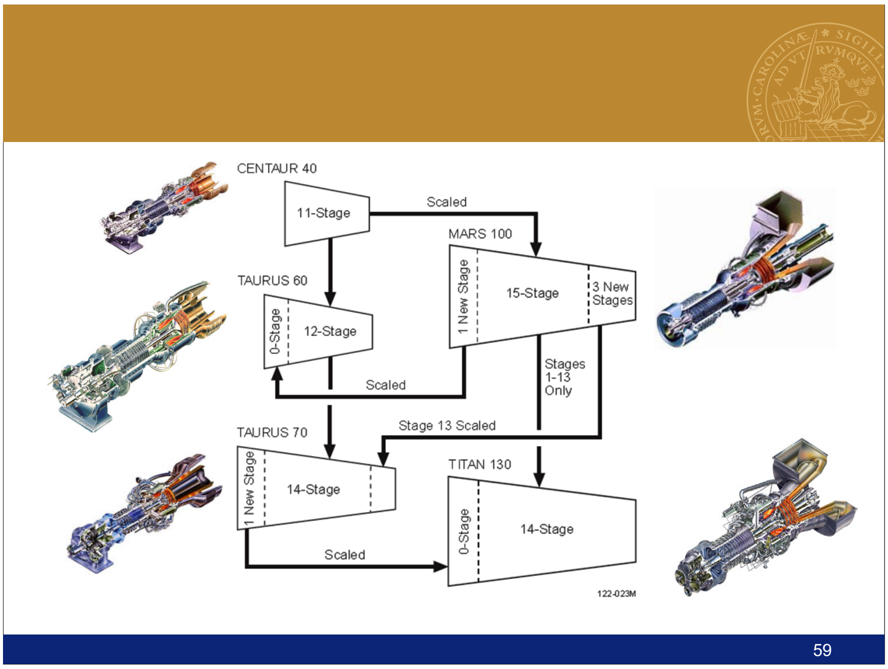
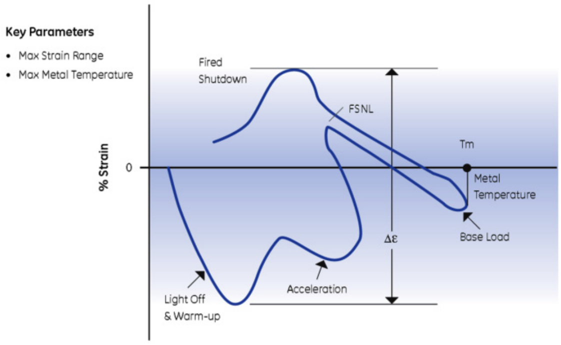


Figure A II-14. Solar compressor family

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. II.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 exist 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. II.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…60×10^6. 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 $\sqrt{(E/\varrho)}$. 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. II.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)

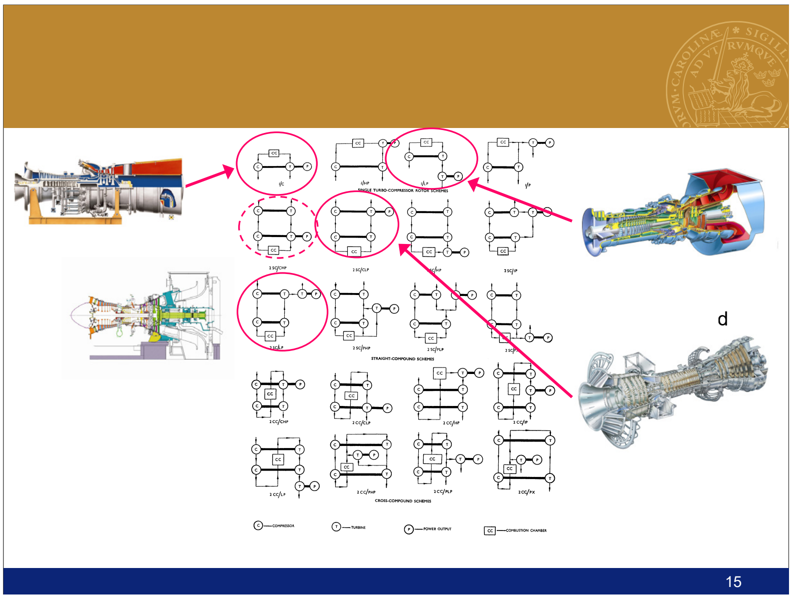


Figure A II-15. The Mallinson and Lewis shaft configuration study

GAS TURBINE DEVELOPMENTS

2012–2015

I Sverige är gasturbinens roll idag begränsad till en handfull anläggningar för kraft- och värmeproduktion och störningsreserven. Basproduktionen är normalt en mix av kärnkraft, vattenkraft, industriellt mottryck, kraft/värme och vind. Globalt ser situationen annorlunda ut och det finns i storleksordningen 50 000 körbara enheter. Gasturbinens är ofta en del i en kombianläggning, som idag kan uppnå verkningsgrader kring 62%, och detta är den högsta möjliga nivån för alla termiska produktionsslag. Utöver den höga verkningsgraden, har gaskombin mycket stor flexibilitet och kan t.ex. startas under 30 minuter, lastomfång mellan 40...100 %. Detta gör att gaskombin blir en förutsättning för höga nivåer väderberoende och förnybar kapacitet i näten – eftersom ingen annan teknik erbjuder samma flexibilitet och verkningsgrad för balanskraft utom vattenkraft.

Behovet av balanskraft ökar med ökande nivåer av vind framförallt, men även sol, eftersom kapacitetsfaktorn är cirka 30 % som årsmedel. Gasturbinen kan även göras bränsleflexibel och kan då bli en del ett helt förnybart system genom att antingen elda biogas och/eller vätgas. Det sistnämnda gör att gasturbinen kan variera mellan att vara en del av basproduktionen, men även del av ett vätgasbaserat energilager.