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TECHNOLOGY UPDATE ON GAS TURBINE DUAL FUEL, DRY LOW EMISSION COMBUSTION SYSTEMS

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ABSTRACT

A challenging issue in the gas turbine industry is to develop a practical dual fuel (DF), dry low emission (DLE) combustion system. Especially for the onshore-based power generation systems, and liquid DLE for aeroderivative engines used for marine propulsion.

A novel mid-size (3MW) gas turbine is being developed mainly targeted for marine propulsion, where a dual fuel DLE combustion system aiming at single digit NOx emission figures has been explored. As a part of this development, the present technology available from different gas turbine manufacturers has been surveyed. Status of the different techniques applied in dual fuel DLE combustors today and their achievements are presented, including the available information on fuel injectors, cooling schemes, combustion air distribution, noise control and combustor performance. The techniques utilized and explained are such as flame temperature control (water/steam injection), staged combustion, lean premixing and lean prevaporized premixing, rich-quench-lean-burning (RQLB) and catalytic combustion. These are also documented for the different concepts commercially available, describing both advantages and drawbacks. Conclusions are made towards the dominating trends for the different parameters mentioned above, and how they affect the final combustor design.

A survey of the dominating parameters for low emission combustion systems is presented.

INTRODUCTION

To reduce the environmental impact from gas turbines as well as diesel/gas engines it is a strong incentive to develop low emission combustion systems. In this course, dual fuel low emission systems are offered, giving customers the opportunity to run on liquid or gaseous fuel without any physical change to the machinery or auxiliaries. However, combining these two issues into a single system, which separately are very complicated matters, needs considerable research and development.

Manufacturers have chosen different technologies in order to achieve their targets. These technologies are summarized and described based on information published in papers and available literature. As an outcome, a description on how different parameters influence important issues such as performance and emissions is useful.

NOMENCLATURE

CO	Carbon MonOxide
CO2	Carbon DiOxide
D	Droplet diameter [m]
$\dot{m}_{combustor}$	Combustor Massflow [kg/s]
NH_3	Ammonia
NO_x	Oxides of Nitrogen (NO and NO_2)
T_{inlet}	Combustor Inlet Temperature [K]
T_{AD}	Adiabatic Flame Temperature [K]
UHC	Unburned Hydrocarbons
v_R	Combustion air velocity relative to liquid (m/s)
V	Volume [m^3]
We	Weber number
ρ_{comb_air}	Combustion Air Density [kg/m^3]
σ	Fuel droplet surface tension [kg/s^2]
τ	Residence Time [s]

MAIN CHALLENGES

Today's emission legislations for combustion engines require the most recent knowledge and technology available. The combustor can be considered as the vital part of the hot components of a gas turbine. The effects downstream of one combustor could be detrimental, if not significant resources are put into the combustor development. Distorted temperature profiles in the combustor outlet, high CO/UHC/NO_x emissions, noise and instabilities will lead to an increased rate of engine deterioration.

The main challenges and requirements for the development of a low emission combustion system are:

- Life and integrity – selection of materials and appropriate coatings, heat transfer and temperature distribution, static and cyclic load, engine rating, combustion generated noise
- Cost – materials and coatings selection, sensors, mechanical and control complexity, part-count, manufacturing and building
- Emissions – fuel preparation, fuels (dual fuel options), noise and space envelope, range requirements
- Safety – flashback, auto-ignition, start/restart, durability, integrity
- Operability – engine cycle, acceleration and deceleration requirements, start/restart requirements, ambient conditions

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The items are described with emphasis on the limiting issues. All of these reflect the complexity of the combustion process:

- The heat release from the highly exothermic chemical reaction
- The product formation as a result of the mixing, chemical reaction and the resulting high temperature
- The pressure pulsations due to the flame front variation and the heat release location
- The wide variety of operating conditions due to the load changes for the gas turbine – stability range

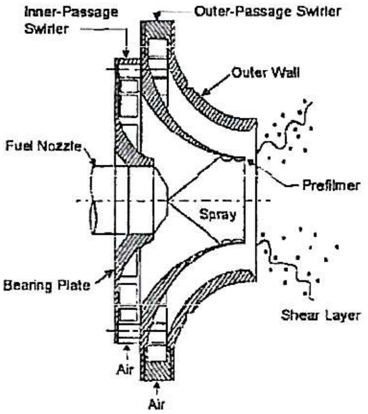

Figure 1 Counter rotating swirlers, with fuel injection into the shear layer [1].

Fuel Injection.

An essential issue is the design of the fuel injector, and important items to be considered are;

- Dual fuel capability
- Atomization and vaporization of liquid fuel
- Distribution of fuel
- Separate inlet systems for liquid and gaseous fuels
- Starting/ignition
- Coking

Figure 1 depicts a fuel injector for liquid fuel. The shear layers created in the airflow by merging separate swirling flows, such as from each side of a vane, is an efficient location for fuel injection. The high turbulence being achieved in such a layer enhances the mixing of fuel and air. The figure shows some of these elements; spray distribution into turbulent zones and double radial swirlers.

Evaporation (liquid) and mixing of fuel.

Homogeneous mixing of fuel and air in the premixer is essential for sufficient ignition delay time margin of the fuel-air mixture. Non-homogeneous regions in the fuel-air mixture also have a significant consequence for NO_x formation, as described in [2]. Non-homogenous regions will result in high temperature peaks occurring in local zones. Thus, more NO_x will be formed (documented in the next section). On the other hand, local rich zones might stabilize the combustion process. A lean uniform mixture is vulnerable for small disturbances that can result in instabilities.

For gaseous fuels the main items are to distribute and mix the gas and air properly. For liquid fuels, the droplets must be atomized properly and evaporated. The droplet break up process can be split into 3 parts; fuel film, ligaments, droplets. The droplet sizes is a function of the Weber number (We) of the fuel spray, which is defined as the ratio of the disruptive aerodynamic forces of the spray to the surface tension force that holds the droplet together [3]:

$$We = \frac{\rho_{comb\_air} v_R^2 D}{\sigma} \tag{1}$$

The larger We, the faster the evaporation and atomization can occur.

For low emission application, the aspect of fuel bound nitrogen needs to be considered for liquid fuel, whereas this is more or less non-existing for most of the gaseous fuels. Fuel bound nitrogen can give from 0.5 ppm to 10 ppm NO_x depending on the fuel nitrogen content.

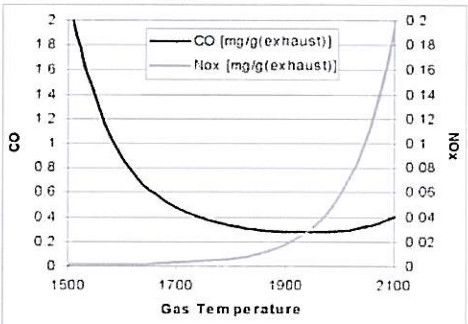

Figure 2 NO_x and CO versus combustion temperature, results from Chemkin Collection (Release 3.6.2), GRI-Mech 3.0, [4], ($T_{inlet, mixture}$=600 K, $P_{combustor}$=15 bar), τ = 8 ms.

Emissions.

The formation of NO_x and occurrence of CO occur in different temperature regimes. NO_x is formed mainly through the thermal (Zel'dovich) NO_x route, which becomes significant at temperatures above 1850 K. CO is present at lower temperatures, due to lower reaction rates and less oxidation of CO to form CO_2. Thus, the window of opportunity for low NO_x and CO is a quite narrow range for both equivalence ratio and thus combustion temperature, as shown in Figure 2. The two other main mechanisms for formation of NO_x are Prompt NO_x and N_2O (nitrous oxide). These mechanisms are not very temperature sensitive. At lean conditions, such as in most DLE concepts, the nitrous oxide route is dominant, whereas Prompt NO_x is more likely to occur at fuel rich conditions.

Emissions of UHC are a result of local fuel-rich zones. All of the fuel will not be combusted and UHC will occur in the combustion gases. In gas turbines it is seen that UHC emissions in general follow the same pattern as CO emissions.

Details on the different pollutant mechanisms are given in for instance [3] and [12].

Noise and instabilities.

Combustion generated noise and acoustic phenomenas have been a major obstacle for further development of low emission combustion systems. Recent publications from for instance Rolls-Royce [5] and Siemens [6] show examples of how this problem can be addressed. The main issue is to understand why instabilities occur and whether their main drivers are thermodynamic, aerodynamic or chemical effects.

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Unfortunately, no generic models have so far been generally approved for predicting combustion instabilities and acoustics.

Operating conditions.

For gas turbines in cyclic operation, such as in marine and aero applications, it is important to control emissions while maintaining performance and efficiency over the operating range. Onshore gas turbines for power generation are mostly operated on the same load. Thus, more effort needs to be put into maintaining performance, efficiency and emissions at rated power for marine and aero gas turbines. For a combustion system that is designed with special emphasis for operation at full load, difficulties at part load due to the changes in fuel equivalence ratio, velocities, pressure and temperature will occur. Advanced control systems are required.

A summary is made of the basic obstacles and how these can be addressed.

EMISSION CONTROL TECHNOLOGY SURVEY.

Some of the technologies to achieve low emissions have a substantial track record and have been field proven. Emission control technologies applied in gas turbines for achieving low emissions are:

- Water or steam injection (wet low emissions, WLE)
- Lean Prevaporized Premixing (LPP), liquid fuel.
- Lean Premix (LP)
- Lean Direct Injection (LDI)
- Staged combustion, with two or more steps of fuel injection or air introduced in a parallel or serial manner.
- Variable Geometry (VG) - constant equivalence ratio.
- Rich – Quench – Lean – Burn combustion (RQLB).
- Catalytic combustion.

There are different technologies and limitations for gas turbines depending on the size of the application. For instance, a gas turbine used for onshore power generation does not have restrictions on weight and size similar to a gas turbine for marine propulsion in a lightweight catamaran or similar.

A short description of the different technologies is included below.

Wet low emissions, WLE.

WLE has direct consequences for NO_x emissions, as thermal NO_x production is a consequence of high flame temperatures. Water or steam injection has mainly been a feasible solution for power plants where it can be provided as a retrofit package. However, WLE is still being developed for new combustion systems and has a certain share of the gas turbines in the power generation market. Due to the lower temperature achieved with WLE, the limitation is the CO and UHC emissions due to the lower reaction rate at lower temperatures. An optimum for the water-fuel ratio (WFR) is reached at approximately unity. As seen in Figure 3, which shows actual measured emissions for testing of a conventional combustor, the CO curve starts to increase rapidly when the WFR rises beyond 1. The NO_x emissions still decrease. With steam injection the steam-fuel ratio (SFR) can be as high as 2-3, which also increases the power output of the gas turbine in addition to reducing emissions. Cheng [7] has developed a promising low emission combustion system utilizing steam/natural gas premixing called Cheng Low NO_x (CLN). CLN has proved single digit NO_x and CO simultaneously and has a substantially lower pressure drop in the combustion chamber compared to DLE-systems [8].

With water injection the power increases and the efficiency decreases due to the energy needed for evaporation of water.

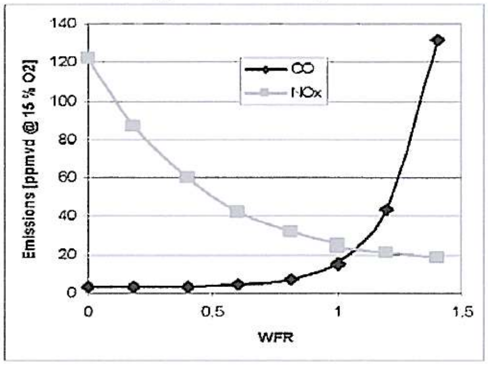

Figure 3 CO and NO_x emissions vs. water-to-fuel ratio (WFR), P=8 bar, T_{inlet}=720 K. Rig test of a typical conventional combustor.

Lean premixing (and prevaporisation).

Lean Premix (LP) and Lean Premixed Prevaporised (LPP) combustion introduces the ability to run a combustor at leaner conditions and thus lower combustion temperatures. Efficient premixing is achieved by high turbulence in the mixing region, longer residence time in the premixer will benefit the mixedness before entering the primary combustion zone. A way of achieving this is by introducing swirl in the injector. Another is to introduce the fuel into a high velocity throat, for instance a venturi as on the Rolls-Royce Ulstein Turbine LPP [9] concept and the OPRA OP16 [10] gas turbine (figures are shown in Table 1). Stabilizing the flame from a lean premix system can be achieved by a swirling flow giving a recirculation in the primary zone, such as dump diffusors, or with a larger volume downstream of the venturis. A uniform mixture of fuel and air is vulnerable for small disturbances that may result in instabilities. Thus, a major problem when designing combustors with lean premixing has been keeping the combustion generated noise levels at a satisfactory low level. Most of the manufacturers of gas turbines with lean premix combustion systems today have published information on controlling instabilities, such as:

- RR RB211 and Trent combustors [5], [11] – carefully monitoring the primary zone temperature and fine-tuning the fuel distribution according to this.
- Siemens KWU [6] – Active Instability Control (AIC), modulation of the fuel control valves, which reduces the pressure amplitudes and thus the noise.
- General Electric (GE) and Pratt & Whitney – Helmholz Resonators.

Lean Direct Injection.

Lean Direct Injection is based on an injector system where fuel is injected into the flame tube in an arrangement that gives a high rate of mixing. Typical, for efficient mixing in LDI, the mixing process must occur at shorter time scales than the chemical. This can be achieved with a Damkohler number (Da) less than unity, where Da is defined as the ratio between the turbulent time scale and the chemical time scale. This can give satisfactory mixing as with a lean premix module and thus achieve low NO_x emissions. A more detailed description is given by Correa [12]. With a highly turbulent diffusion

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flame like this, LDI is a probable design for future development for aero gas turbine combustors. A typical design proposed and patented by CFDRC is described in [13] and a more complex design is described in [14].

Staged lean combustion.

Staged combustion can be separated into serial (SSC) and parallel (PSC) systems. For gas turbines in general, it is difficult to achieve low emissions over the entire operating range. Thus, staged combustion is introduced with special emphasis on achieving good emission performance at lower power settings. The varying equivalence ratios throughout the operating range give variations in flame temperatures and combustion efficiency.

SSC can be exemplified with the RR RB211 and Trent combustion systems, which will be described later. A primary, secondary and main zone has separate fuel injection locations. Thus, the flame temperature can be controlled carefully by an efficient fuel distribution and control system. On the other hand, PSC is a system where multiple burners or injectors are placed in parallel. One such system is the GE LM6000 and Heavy-Duty DLN combustors. In a PSC system, some of the burners can be run at optimum conditions at part load, whereas more burners are ignited at higher power settings.

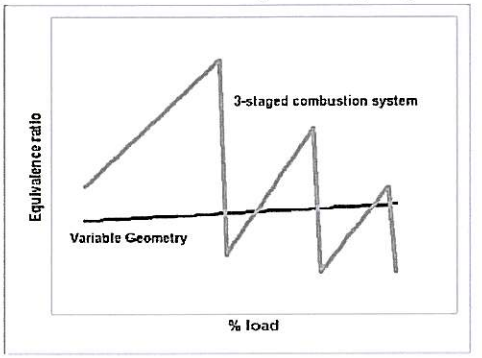

Figure 4 Comparison of Variable Geometry and 3-staged combustion system.

Variable geometry.

Staged combustion can also be achieved by Variable Geometry (VG) devices in the combustion system. A VG system consists of an adjustable unit that can change the distribution of the combustion air, such that optimum local fuel air ratios can be achieved through the operating range of an engine, for instance a valve. A VG system can be considered as a staged combustion system with an infinite number of stages. An example is described for the OPRA OP16's combustion system in [10].

Figure 4 shows the advantage of VG compared to a typical three staged combustion system. As the air can be bypassed the primary zone in a VG, all the fuel can be introduced in a single stage and still maintain a near-constant equivalence ratio. If the number of stages in a staged combustion system is increased the red curve would approach the VG curve.

Rich-Quench-Lean-Burn combustion.

Rich – Quench – Lean – Burn combustor design separates the combustion process in two zones; a fuel rich zone where the intention is to burn all the oxygen in the air and a second leaner zone where the remaining fuel will be burnt. The intention is to stop the rather slow chemical reaction of NOx formation with the sudden quenching between the two zones, as indicated in Figure 5. The fuel rich zone will produce unburned hydrocarbons (UHC), CO and unburnt fuel fragments due to the rich mixture. In the second zone, addition of excess air will result in a leaner mixture of fuel and air and thus low combustion temperatures. On the other hand, a rich mixture in the primary zone produces soot, and the sudden addition of air between the two zones can lead to low oxidation rates of the fuel fragments/soot. The main obstacle is to achieve an efficient transfer from the rich zone to the lean zone, mixing the excess air without burning it between the two zones is a complicated matter.

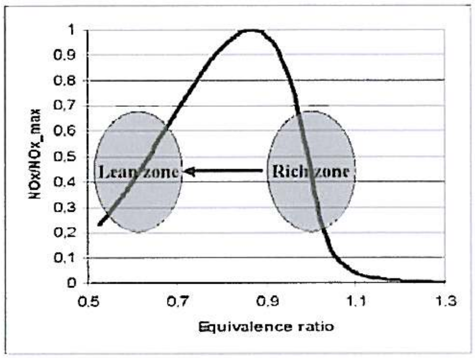

Figure 5 The RQLB combustion process, Chemkin Collection (Release 3.6.2), GRI-Mech 3.0, [4], ($T_{Inlet, mixture}$=300 K, $P_{combustor}$=15 bar), equilibrium.

Catalytic combustion.

The main objective for developing catalytic combustors is to be able to initiate a heterogeneous chemical reaction between the fuel and the air at a lower temperature. It is usually achieved by introducing a ceramic or metal structure coated with noble metals. The chemical reactions take place on the surface of these metals. Since running at lower temperatures, catalytic combustors have the ability of operating at much leaner conditions than DLE combustors. It is possible to control the temperature very accurately and optimize the reaction process. The disadvantage is the lifetime for the catalyst at the required temperatures and the cost of the catalyst materials, especially if the reaction turns homogeneous and the temperature increases significantly. Also, the thermal shocks, considering gas turbine (GT) cycles, is a significant challenge for the catalyst material. The catalytic combustors are quite demanding in size. Some examples of catalytic combustion development are described in [15] for Solar and [16] for RR Allison. These combustors prove single digit NO_x emissions in laboratory scale. Companies such as Catalytica Energy Systems (Xonon) and Precision Combustion have developed prototypes of catalytic combustion systems in cooperation with gas turbine manufacturers. Commercial operation with less than 2.5 ppm NO_x with Xonon combustion system on a Kawasaki GT is described in [17]. The cost of producing an efficient catalytic combustor increases rapidly as it needs some sort of heating device to ignite the mixture of fuel and gas. Some concepts being developed today are using a premixed first stage burner before the catalyst to ensure high enough temperature for the catalyst to ignite.

The strategies mentioned above are mainly focused on DLE issues (except WLE), regardless of DF capability. It is generally acknowledged that lower emissions are easier achieved with gaseous fuels than with liquid fuels due to complication of the atomization

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and vaporization process in addition to mixing issues, and the fuel-bound Nitrogen in liquid fuel.

LOW EMISSION COMBUSTION SYSTEMS.

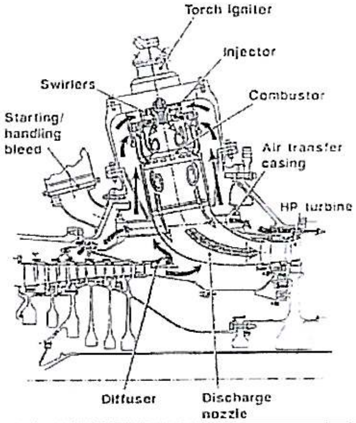
Different examples of combustor design will be described more in detail when going through the different concepts applied for the existing DF DLE combustion systems. Table 1 describes combustion systems, strategies, cooling technology and includes achieved emissions levels. Also, references for more detailed descriptions of the concepts are given.

Table 1 Gas Turbine Combustion Systems – mainly focused on commercial experience and operation.

ROLLS-ROYCE PLC.

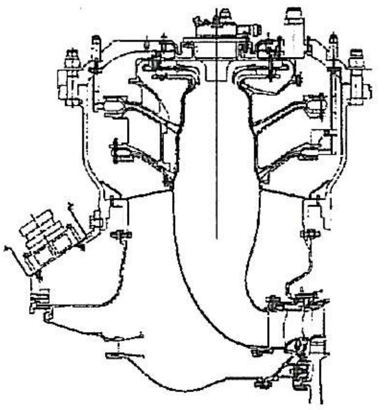
RB211-DLE, 27 MW, ~36 % EFFICIENCY

Strategy	Two-staged Serial Premixed combustion, Double Mixed flow swirlers, Flame temperature control inside flame tube due to the staged fuel injection
Cooling	Transply material flame tube, effusion cooling for the most recent, Thermal Barrier Coating (TBC)
Emissions	$NO_x < 25$ ppm, CO < 50 ppm (gas fuel only)
Noise	Pulsations experienced, could be dealt with by adjusting the fuel distribution and schedules for the primary and secondary fuel inlets.
References	Running experience from [20] and [5]



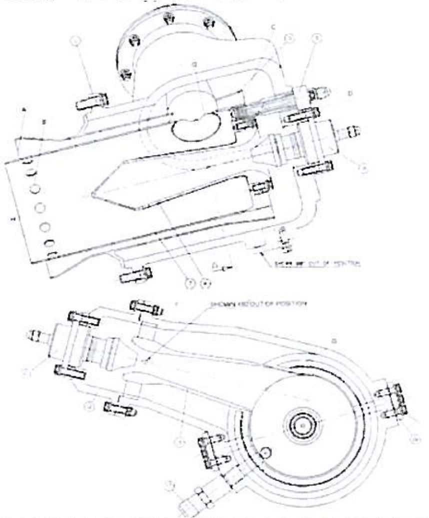
TRENT, 50 MW, ~38 % EFFICIENCY

Strategy	Three-staged Serial Premixed combustion, Double Mixed flow swirlers, Flame temperature control inside flame tube due to the staged fuel injection
Cooling	Effusion cooling, TBC
Emissions	$NO_x < 25$ ppm, CO < 5 ppm (gas fuel only)
Noise	A similar noise problem as for RB211
References	Description in [11]



ROLLS-ROYCE ULSTEIN TURBINE AS, EURODYN, DF LPP COMBUSTOR, 2.6 MW, ~32 % EFFICIENCY

Strategy	Reverse flow tubular combustors, two-staged Lean Premixed Prevaporised combustion, Dual Fuel injectors, Venturi premixers ensuring good atomisation Good discharge coefficient and pressure recovery
Cooling	Convection cooling, TBC
Fuel	Dual fuel
Emissions	Liquid: 20 ppm NO_x and CO Gas: single digit NO_x
Noise	Controlled and limited by modifying venturi inflow conditions and optimizing the venturi design
References	Experimental data from [9]



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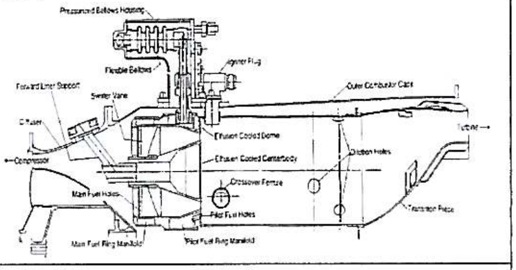
ROLLS-ROYCE COORPORATION, 501KB7, LE4, 5.2 MW, ~32 % EFFICIENCY

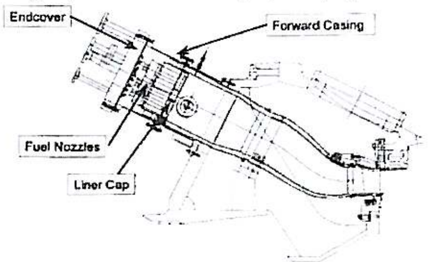
Strategy	Lean Premix combustion, Swirl Premixer, high residence time for efficient fuel/air mixing, high velocity premixer outlet to prevent flashback
Cooling	Effusion and convection cooling, TBC
Emissions	$NO_x < 25$ ppm and $CO < 50$ ppm (gas fuel only)
Noise	No noise reported.
References	Running experience from [18] and [19]



GENERAL ELECTRIC

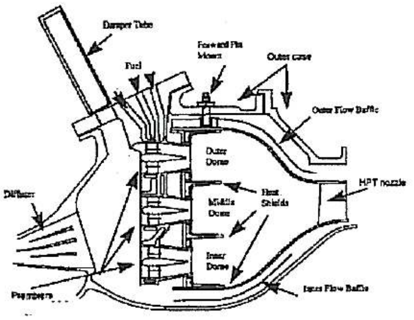
HEAVY DUTY ENGINES, DLN 2.0+ AND 2.6, 26-255 MW, ~33-36 %

Strategy	6 premix burners, Parallel staging, Water/steam injection capability for liquid fuel
Cooling	Impingement and convection cooling.
Fuel	Dual fuel
Emissions	Liquid: 42 ppm NO_x, with steam/water injection Gas: 25 ppm NO_x (9 ppm on MS7001 engine)
Noise	Not reported
References	Operating experience shown in [21] and [22].



LM2500/6000 DLN-1/2, 45 MW, ~42 % EFFICIENCY

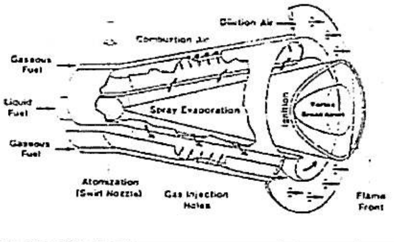
Strategy	75 Double Counter Rotating Swirler in an annular combustion chamber, three rows.
Cooling	Convection cooling with ribs
Emissions	$NO_x < 25$ ppm, CO~10 ppm (gas fuel only)
Noise	Helmholz resonators
References	Development from [2]



ALSTOM POWER

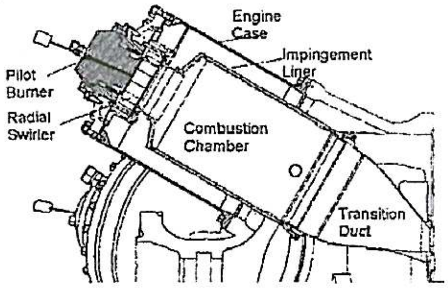
GT10-24/26, EV-BURNER, 30+ MW , ~36 % EFFICIENCY

Strategy	Premixed combustion, Conical Venturi-shaped premixer, liquid fuel injector in the centre and gas injected in the outer manifold of the cone. Applied into annular and silo-type combustors
Cooling	Film cooling, convection
Fuel	Dual fuel
Emissions	Liquid: < 40 ppm NO_x with water injection Gas: < 25 ppm NO_x and 30 ppm CO Liquid operation with Advanced EV-burner (AEV) below 25 ppm NO_x emissions shown in [24]
Noise	Pulsations within acceptable levels for EV and AEV burners, as described in [22]
References	Documentation from [25], [26] and most recently in [27]



CYCLONE/TEMPEST/TYPHOON, 4.3-13.4 MW, ~33-35 % EFFICIENCY

Strategy	Reverse flow tubular combustors, Pilot with Radial inflow Swirler, Gaseous fuel injection in swirler passage, Liquid fuel injection on the inner wall of the pilot
Cooling	Impingement and convection cooling
Fuel	Dual fuel
Emissions	Liquid: < 50 ppm NO_x and CO Gas: < 25 ppm NO_x and 50 ppm CO
Noise	Within acceptable level.
References	Described in [28], [29] and [30]



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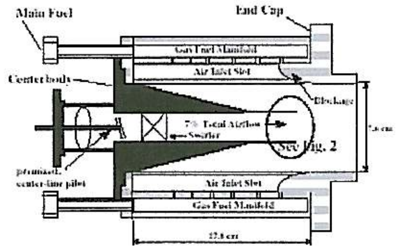
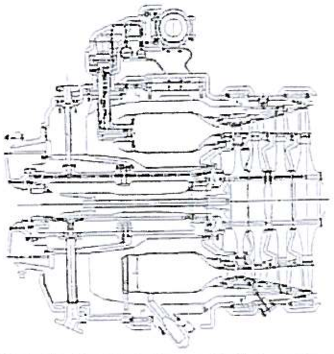
PRATT & WHITNEY (UTC), FT8, 25-60 MW, ~38.5 EFFICIENCY

Strategy	Reverse Venturi burners, annular combustion chambers, Three-staged fuel injection Pilot and Lean Premix zones
Cooling	Film cooling
Emissions	$NO_x < 25$ ppm, $CO < 30$ ppm (on gas)
Noise	Helmholz resonators
References	Documented in [31], [32] and [33] A triple venturi retrofit kit for the FT4 engine is described in [34]. Dual fuel capability and 25/50 ppm NO_x for gaseous and liquid fuel respectively.



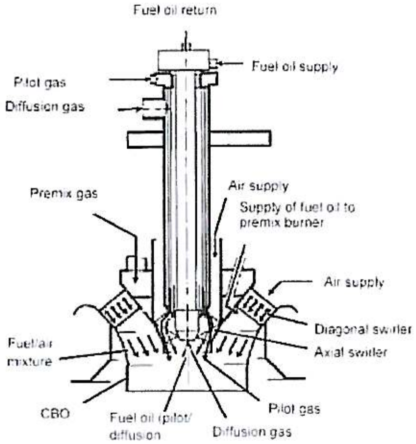
SOLAR TURBINES INC., SOLONO$_X$, 5-14 MW, ~34 % EFFICIENCY

Strategy	Air-bleed injector giving low emissions at part load, no staging, Annular combustion chamber, No dilution, 60 % of combustion air through injector
Cooling	Film cooling
Fuel	Dual fuel
Emissions	Gas: $NO_x < 25$ ppm and CO< 50 ppm, With modified Dual Fuel design; Liquid: < 100 ppm NO_x, Gas: < 80 ppm NO_x
Noise	Avoided with fuel injector modifications
References	Described in [35], [36] and [37] Catalytic pilot development with NOx levels below 5 ppm at full load documented in [38]



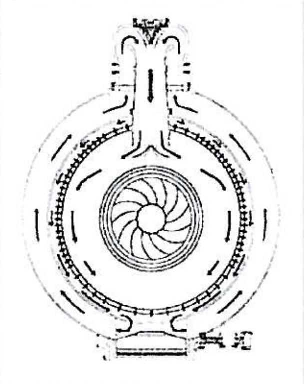
SIEMENS, KWU HYBRID BURNER, 67-265 MW, 34.7-38.5 % EFFICIENCY

Strategy	Swirl Premix burner, Annular or silo-type combustor Gaseous fuel injected before swirler vanes, Liquid fuel low emissions achieved by water or steam injection
Cooling	Film cooling
Fuel	Dual fuel
Emissions	Liquid: 42 ppm NO_x Gas: NO_x< 15 ppm and CO < 10 ppm
Noise	Active Control System for noise reduction
References	Experience from [39], [40], [41], [42] and [43] Some work done for noise reduction on a Siemens Westinghouse combustion system is described in [44]



OPRA TURBINES, OP-16 COMBUSTOR, 1.6 MW, ~28 % EFFICIENCY

Strategy	Variable Geometry combustion system – air valve Venturi premixer Achieving a relatively constant equivalence ration throughout the entire operating range
Cooling	Impingement/convection cooling
Fuel	Dual fuel
Emissions	Liquid: $NO_x < 25$ ppm and CO < 9 ppm Gas: $NO_x < 10$ ppm and CO < 6 ppm
Noise	No noise or acoustics reported
References	Development documented in [10]



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PARAMETERS AFFECTING PERFORMANCE OF LOW EMISSION COMBUSTION SYSTEMS.

Some of the main parameters for low emission combustion are discussed here. As detailed geometrical and operational data from the gas turbine manufacturers are unavailable, a generic gas turbine in the power range up to 3.0 MW has been chosen as an example for the parameters discussed in the following section.

A combustor for this gas turbine can be described as in Figure 6. This combustor has a relatively large premixer with a swirl generator (10 % of entire combustor) before the primary combustion zone (PZ). This utilizes sufficient residence time and high turbulence for good premixing of fuel and air. For lower power settings, a pilot fuel injector is present in the primary zone of this combustor to maintain stable combustion at partial load. In the last part, the reacting mixture is diluted by air to achieve the design temperature distribution before the gas enters the turbine. The walls of PZ and dilution zone (DZ) are cooled by effusion cooling. The air distribution is given in Table 2.

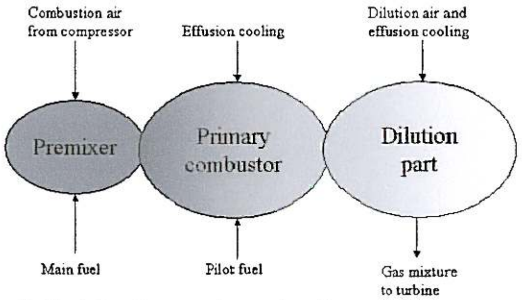

Figure 6 Sketch of general combustion system.

Table 2 Air distribution in combustor.

Premixer	Effusion cooling	Dilution air
45 %	10 %	45 %

Figure 7 indicates the equivalence ratio in this typical combustor. The pilot is a diffusion type fuel injector giving a stable flame at the lower equivalence ratios, while the main fuel inlet (premixer, ref. Figure 7) is inside the premixer volume.

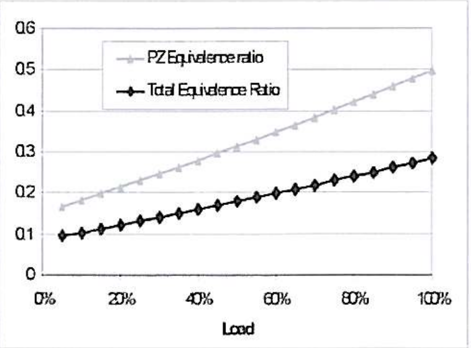

Figure 7 Fuel schedule for combustor.

In Table 3, six examples for typical gas turbine combustion systems are shown. The gas turbines are applied with reverse flow combustors with varying number of combustors due to the size of the gas turbine. The geometry of the flame tubes is fixed. They are scaled with regards to matching the premixer and primary zone residence time, and thus varying the diameters and mixing volume of the premixer. The combustors' boundary conditions are partly taken from specific combustors, and partly from averaging combustion systems for gas turbines in the defined power range.

Table 3 Exemplified Gas Turbines.

	GT-1 v1	GT-1 v2	GT-3 v1	GT-3 v2	GT-10 v1	GT-10 v2
Number of combustors	2	1	6	3	16	8
Pressure ratio	6	6	18	18	22,5	22,5
Inlet temperature (K)	550	550	748	748	850	850
Premixer res time (ms)	5,87	5,72	4,31	4,21	4,75	4,63
PZ res time (ms)	2,71	1,35	8,17	4,09	9,84	4,92
Massflow (kg/s)	3,60	3,60	10,81	10,81	28,82	28,82
Massflow/combustor (kg/s)	1,80	3,60	1,80	3,60	1,80	3,60
Efficiency (%)	27,1	27,1	32,0	32,0	34,8	34,8
Combustion Power (kW)	3772	3772	11315	11315	30173	30173
GT Power (kW)	1131	1131	3394	3394	9052	9052

Primary Zone temperature and residence time.

The PZ temperature is the most important parameter for controlling the NO_x formation. The bulk of the combustion process takes place within the primary zone, and this is where combustion products are formed. Thus, carefully monitoring the temperature here is crucial for low emissions. This is a reminder of the narrow window of opportunity as shown in Figure 2.

For the six gas turbines described in Table 3, the adiabatic flame temperature will vary with the combustor inlet temperature. The change in flame temperature is approximately half the change in inlet temperature. The equivalence ratios profiles are kept constant and will not give flame temperature variations for these six gas turbines.

Also, as shown in Table 3, the PZ residence time is varied significantly with temperature and density. It is calculated using equation (2).

$$\tau_{PZ} = \frac{V_{PZ}\,\rho_{comb\_air}}{\dot{m}_{PZ}} \cdot \frac{T_{inlet}}{T_{AD,PZ}} \qquad (2)$$

The wide variation in PZ residence time indicates questionable emission levels gas turbines with 1, 3 and 8 combustors. The low PZ residence time in GT-1v2 may result in high CO emissions whereas the long PZ residence time in GT-10v1 may give high NO_x emissions.

The air distribution in the combustor has effects on the premixing module and the cooling of the flame tube. With 47 % of the air introduced in the premixer, proper mixing of fuel and air can be achieved, hence a more efficient and uniform combustion process. However, with less air for cooling of the flame tube, this requires extra attention. As the combustor geometry dictates the air distribution, the temperature distribution will be controlled by the fuel distribution in the different zones. As shown and discussed earlier, combustor zone temperatures control (e.g. RR Trent combustion system) can further control issues related to noise, thermal load on material and the total performance of the combustor.

The primary zone is mainly dictated by the premixer outlet airflow. At full load, all fuel is injected in the premixer, thus a uniform lean mixture for combustion can be achieved. To avoid the risk of instabilities that could cause flashback, a high velocity throat at the premixer outlet is applied. The amount of air in the PZ should consider the increased inlet temperature of the combustor. Thus, for different conditions and power ratings more air should be introduced to maintain a stable equivalence ratio. Then, the performance and

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emissions can be expected to be at equal levels for the two different sizes of the combustor.

Controlling the airflow in the primary zone with Variable Geometry is shown for the OPRA OP16 combustion system [10]. The MF_{PZ} is varied from 0,48 to 0,88 for gaseous fuel, and from 0,62 to 0,88 for liquid fuel. It is the found that the NO_x emissions can be seen in connection to the variations of MF_{PZ} for this gas turbine.

Premixer residence time vs. autoignition risk.

All of the combustors described in have a premixer part where the fuel is mixed with the air. With a uniform mixture of fuel and air entering the PZ, the advantage of an airflow-controlled flame becomes clearer. Then, a way to calculate the premixer residence time follows:

$$\tau_{PM} = V_{PM}\left(\frac{\rho_{comb\_air}}{\dot{m}_{combustor}}\right) \qquad (3)$$

This equation is similar to equation (2), without the temperatures, since there is no combustion in the premixer.

The residence time in the premixer affects the uniformity of the mixture of fuel and air. A uniform mixture is desired for lean premixed combustion. But on the other hand, for liquid fuel operation there is the risk of autoignition at elevated pressure and temperature. Autoignition characteristics have been studied different combustor designs both for gaseous and liquid fuel. For instance, Røkke and Wilson [9], conclude that autoignition margins are very dependent on geometry, flow conditions and fuel injection and not only the chemical effects. However, it is essential to avoid any stagnation points or recirculation zones within the premixer. Then, a mixture of fuel and air would have a significantly longer residence time than the design τ_{PM}. A certain recommendable residence time cannot be pointed out, since this is clearly dependent on the geometry and design of each combustor. Mixtures with gaseous fuel are unlikely to autoignite as documented in [45].

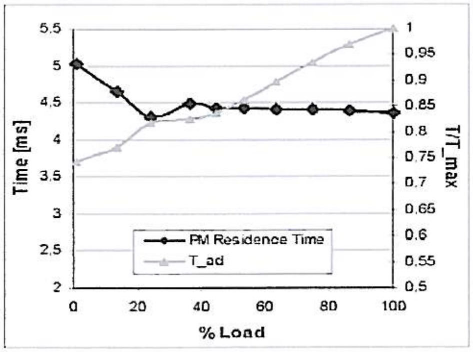

Figure 8 Premixer residence time and combustor flame temperature, according to equivalence ratio shown in Figure 7.

The combustor used here has a relatively large volume in the premixer ($V = 435{,}6$ cm^3), residence time and the associated flame temperature is shown in Figure 8 (based on the premixer flow only). The residence time is contained within 4 and 5 milliseconds. As more fuel is introduced, the premixer equivalence ratio increases and the flame temperature increases. The flame temperature is found using the Chemkin Equilibrium (v. 3.6.2) code based on the equivalence ratio in the primary zone.

Flame stability.

At lower loads a pilot inside the flame tube will keep the flame stable. Since the pilot flame is a diffusion flame, higher local temperatures will decrease the possibility for high CO emissions. The PZ equivalence ratio increases up to 0.48 @ full load. The NOx formation rates are then kept at a satisfactory low level. The Chemkin Equilibrium code will not give proper results for the level of NOx/CO emissions since a perfect mixture of fuel and air is assumed. The residence time in a Gas Turbine combustor is not long enough to achieve equilibrium. However, the calculated temperature level can give indications on the NO_x-production rates. It is aimed to have a temperature between 1650 K and 1850 K, as indicated in Figure 2.

FUTURE STRATEGY AND DEVELOPMENT.

In the early nineties, Correa [12] predicted the future strategy to be lean premix designs, a prediction that has been fulfilled. The advantages were said to be the following;

- Lean premix combustion is virtually pressure independent,
- Turbulent mixing increases uniformity of mixture,
- NO_x is mainly formed from the thermal NO_x mechanism.

However, if the targets for NO_x emissions are below 10 ppm, other mechanisms than the thermal need to be considered. The prompt/Fenimore process and the nitrous oxide mechanism (N_2O) can contribute with 1-5 ppm. When burning liquid fuel, fuel bound Nitrogen also will contribute with some ppms.

The common message for all concepts described in Table 1 and in the referenced documentation, is to achieve efficient premixing of the fuel and air and evaporation (in case of liquid fuel) before entering the flame tube of the combustor. Efficient premixing is most commonly achieved with for instance axial swirlers, radial swirlers, venturis and/or combinations of the mentioned. Swirler arrangements introduce high air velocities and zones with complex flow interactions resulting in high turbulence and shear forces and thus good mixing conditions. With an efficient premixing process in the combustor, the ability to run on a lean fuel to air ratio lowers the peak combustion temperature and decreases the combustor cooling requirements. The disadvantage of running on lean conditions is the marginal blowout limit, and the risk for high CO/UHC emissions at low combustion temperatures. Hence, operation of a low emission combustion system requires accurate fuel control systems, which is particularly challenging for transients.

Lean Premixing.

The Rolls-Royce combustors in Trent and RB211, introduces fuel in the swirlers, thus premixing fuel and air in the swirling channels before entering the flame tube. Similar strategies are applied in the RR LE4 design with fuel injection in the axial swirler. Alstom and PW have tangential inflow of premixed fuel and air where fuel is introduced before the main zone. If technologies for dealing with combustion instability and acoustics are developed, thus finding reliable methods of solving them, lean premixing is the most obvious future strategy. Noise is the main obstacle this strategy today. For the future, single digit NO_x emissions targets are achievable for stationary gas turbines. Commercial systems today guarantee < 25 ppm (9 ppm for GE Frame 7 DLN). For aero engines with higher

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pressure ratio and higher demands for stability, LDI seems as the future strategy with the stable diffusion type flame.

Catalytic combustion

Catalytic combustion for GT applications has proven emissions close to zero in laboratory tests. Catalytic concepts have still not entered the commercial market in a gas turbine package. However, orders have been made and several publications are available on catalytic systems being tested together with gas turbines. The challenges for catalytic combustion are limited temperatures on the catalyst and the large size of the combustors. The advantage of catalytic combustion is the ability to burn the fuel/air mixture at a low temperature and then avoid the NO_x formation. But as the efficiency for gas turbines increase, the turbine inlet temperature (TET) increases proportionally. It must then be considered whether efficiency demands or environmental issues are to be prioritized. A combination of a catalytic combustor with a post-combustion process where fuel is injected downstream of the catalyst to increase TET up to specification, is an alternative future solution for combustion with NO_x emissions closer to zero. Xonon Cool Combustion is a design shown in [17] that has bypass air reintroduced downstream of the catalyst. In [46], the development of a GE combustion system where fuel is injected downstream of the catalyst to increase TET was described. The catalyst will then not be exposed to temperatures that are too high and thus have a longer life.

The time schedule for DLE combustion systems development goes far into the future, especially for aero engines where the development of low emission combustion systems must consider items as high altitude relight and operating pressures varying from 5 to approximately 40 bars.

The main problem and main challenge seems to be controlling the instabilities on present lean premix designs, this subject is clearly visible by the number on recent publications concerning pulsation control equipment for gas turbines. Noise and pulsations are phenomena that are difficult to simulate by theoretical studies and thus it is very difficult understanding why it occurs for some designs and not for others. Approaches for controlling the instabilities are such as active fuel control, Helmholz resonators and variable geometry.

The market trends and development costs, in general, need to be considered carefully. The european markets have a limit of 25 ppm NO_x, whereas certain states in the US have regulations down to 3-5 ppm NO_x. A gas turbine manufacturer can then consider the best available technology for lean premix systems today, and see that NO_x emissions below 5 ppm for these systems is a tremendous challenge that requires large resources in research and development. A clean-up technology such as Selective Catalytic Reduction (SCR) combined with technology giving 25 ppm NO_x emissions might then be a better solution to achieving the targets, at least in the near future. The SCR systems need to operate at low NH_3-slip (<3ppm) as well. An interesting alternative might be the CLN-system utilizing steam/natural gas premixing, but the system needs further verification in industrial applications.

SUMMARY/CONCLUSION.

The DLE combustion systems of the different manufacturers have been summarized in Table 1. The future trends are very much dominated by the legislations. Thus, governmental and international legislations and regulations may very much control the future development of combustion systems.

The most important parameters (such as temperatures, residence times, fuel/air distribution and mixing) in designing low emission combustors have been discussed in this paper.

Most manufacturers have published data showing single digit NO_x emissions in laboratory or prototype scale, however commercial operations of engines are rarely guaranteed below 25 ppm. Emissions down to the level proved in laboratory testing for catalytic systems are a great challenge. Thus, if environmental issues set legislations below 5 ppm NO_x in the future, alternative methods such as catalytic combustion systems or clean-up technologies in addition to DLE combustors are likely choices.

Concerning liquid fuel, the most common way of achieving low emission is by WLE technology. Dual fuel option is delivered by only a few manufacturers and still remains a challenge for the future.

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