25 Dual fuel engines

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25.1 What is a dual fuel engine?

In a Paper published in 1949[1], when the development of dual fuel engines was in its infancy, Ralph Boyer of Cooper-Bessemer wrote: 'Over a period of years there have been a number of engine developments referred to as Dual Fuel. Unfortunately this terminology has been used in many cases where more than one fuel is involved, but where the fuels could be gasoline versus fuel oil, gasoline versus kerosene, and many other combinations. The term Dual Fuel is therefore perhaps in need of some explanation.' He went on to describe the further confusion arising from the use of the term 'Gas Diesel'.

The situation today is even more confusing. Engines capable of using more than one fuel, and therefore entitled to be referred to as dual fuel engines, have been produced in all shapes and sizes, from the smallest industrial engine to the largest marine engine. Since this is a diesel engine reference book, we need not concern ourselves with spark ignited Otto cycle engines. However, there is a certain irony in the fact that the most common type of dual fuel engine, and the type with which we are primarily concerned here, actually operates predominantly on the Otto cycle when running on gas.

This "typical" dual fuel engine is a turbocharged medium speed diesel engine or between 200 and 500 mm bore, equipped with an additional natural gas fuel supply system. The gas is mixed with the combustion air by timed injection into the inlet ports, and the mixture is ignited by the injection of a small quantity of diesel fuel. Compression ignition of this "pilot" fuel then provides the ignition energy for combustion of the gas-air mixture. As we shall see, there are many variations on this basic system, and many different fuel gases which may be burned.

In recent times there has been increasing interest in a different type of dual fuel engine, known as the "Gas Diesel". This operates on the true Diesel cycle whether running on gas or fuel oil. The gas is injected directly into the combustion chamber and burns as it is injected, in exactly the same way as the liquid diesel fuel. A pilot injection of diesel fuel is still required to provide ignition. Although this approach overcomes a number of drawbacks associated with the conventional dual fuel engine, it requires the gas to be injected at high pressure (up to 350 bar), which brings problems of its own.

25.2 Combustion in dual fuel engines

In the conventional dual fuel engine, combustion is initiated by the compression ignition (auto-ignition) of the liquid fuel pilot. This then serves as the ignition source for the pre-mixed charge of air and gas.

Efficient pre-mixed combustion depends on turbulent flame propagation which in turn is dependent on the equivalence ratio of the gas air mixture. If the mixture is too lean flame propagation will be erratic, leading to misfire. Fuel unburned in the cylinder may then burn in the exhaust manifold. This provides more energy to the turbocharger turbine, increasing the air boost pressure and making the mixture leaner still. In these circumstances the engine can rapidly go out of control.

Conversely, if the mixture is too rich, detonation (or "knock") will occur. It has been shown[2] that, in dual fuel engines, detonation is due to the auto ignition of the premixed air and fuel in the neighbourhood of the pilot fuel sprays. During compression, pre-ignition reactions take place in the mixture. Then, once the pilot fuel ignites both pilot fuel and gas burn very rapidly and simultaneously. This causes excessive rates of pressure rise, high temperature and increased heat transfer often leading to engine damage.

The misfire and detonation limits are strongly affected by the properties of the gas, engine inlet manifold temperature (illustrated in *Figure 25.1*), and compression ratio. The relative susceptibility to detonation of fuel gases is often measured by methane number (see Section 25.3).

Engine control systems need to protect against misfire and detonation and ensure operation at maximum efficiency. In recent times the requirement to do this without exceeding NOx limits has led to increasingly complex and sophisticated engine management systems, such as that shown in *Figure 25.2*.

The requirement to operate within a limited range of equivalence ratio means that, as with spark ignited engines, some means of controlling air flow into the engine is needed. Methods of achieving this are discussed in Section 25.5. To minimize the risk of misfire the pilot quantity as a percentage of the total fuel may be increased. In practice this usually means that the pilot fuel quantity is kept constant throughout the load range, and light load running (below 25% of full load) is done on diesel fuel.

As already noted, the Gas Diesel engine operates on an entirely different principle. The gas is not premixed with the air so control of equivalence ratio is not necessary. Furthermore, it has been shown[3] that the combustion of the gas takes place by the diffusion process, and that the thermal efficiency, cylinder pressure and temperatures are all similar to normal diesel operation.

The fundamentals of combustion processes in dual fuel engines have been investigated in great depth by Karim and his colleagues at the University of Calgary[2,4,5,6].

25.3 Gas properties and their effects

A large variety of fuels are currently being used with medium speed gas engines and these are very dependent on availability and the location of the installation. Details of the major gas types are listed in *Table 25.1* and specific properties and characteristics are shown in *Table 25.2*.

The most important parameters which affect engine operation are as follows:

25.3.1 Heat value of a stoichiometric mixture volume

This will determine the amount of mixture required to achieve a given output. This is of particular importance for naturally aspirated engines as it will directly affect full load output. On a turbocharged engine this is not so critical and will only affect the turbocharger matching. This level remains similar for all hydrocarbon fuels but reduces if significant amounts of inert gas are present, as in digester gas or gasifier gas.

25.3.2 Net heating value (kJ/m^3)

This parameter indicates the volume of gas required for a given engine output. This will affect the design of the gas admission system. It is a major consideration when the engine is to be fuelled by a variety of gases or when the quality of the in-service gas supply varies significantly.

25.3.3 Anti-detonation properties

This is a measure of the auto-ignition properties of the gas ahead of the flame. It is dependent on the chemical constitution of the end gas and the local temperatures and pressures.

The chemical constitution of the end gas will be a function of the air–gas ratio and the gas constitution. As a general rule long-chain gases tend to have poor auto-ignition characteristics. This is reflected by the heavy hydrocarbon gases having poor detonation properties, while CO and CH_4 tend to have good anti-detonation properties. Various methods exist for assessing the

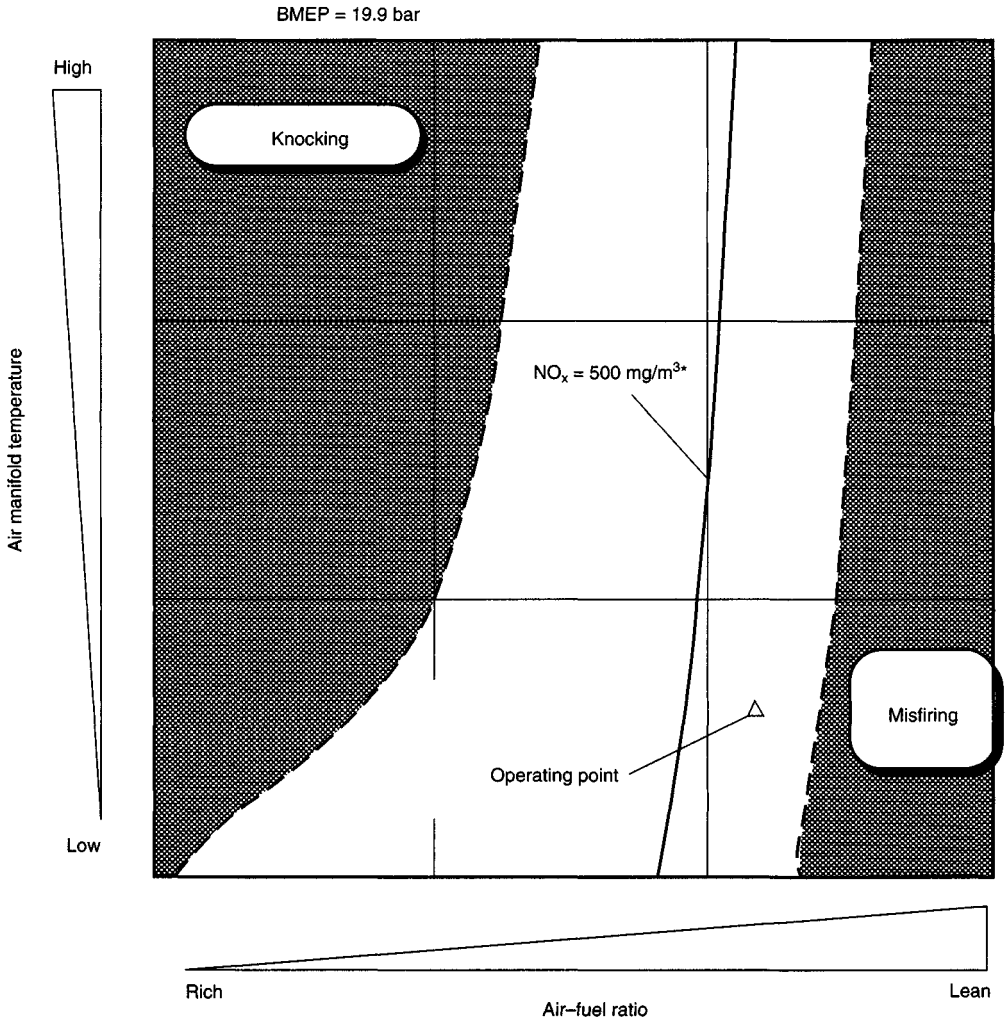


Figure 25.1 Effects of air manifold temperature and air–fuel ratio on the operating range between misfire and knock (MAN B&W)

*Referenced at 5% O_2

detonation tendency of different gases, the most common being the Methane Number rating system, which has some similarity to the octane rating system used for gasoline fuels. Higher numbers indicate greater resistance to detonation, referenced to pure methane which is given the value 100. An extensive test programme to investigate the effects of gas composition (on spark ignited gas engines) was jointly undertaken by Ricardo Consulting Engineers and Tokyo Gas Company in 1994[7]. This confirmed the validity of Methane Number as a general measure of the knock resistance of a gas fuel, but also showed that there can be significant differences in the actual BMEP achievable with gas mixes of similar Methane Number.

The local temperatures and pressures in the combustion chamber are dependent on a complex relationship between engine variables, such as engine speed, bore size, compression ratio, combustion chamber shape, etc. and combustion variables such as air temperature, start of combustion, flame speed, fuel heat input, etc.

25.3.4 Pre-ignition tendency

This is the tendency of the gas to ignite spontaneously owing to local high temperatures of 'hot spots'. These normally occur in the vicinity of the exhaust valve and can be minimized with correct engine operation and adequate combustion chamber cooling. As a general rule, gases with low resistance to detonation will also be susceptible to pre-ignition. Hydrogen in particular has very poor pre-ignition characteristics. Because of the likelihood of local hot spots caused by combustion chamber deposits, low ash content lubricants are normally recommended.

25.3.5 Flame speed

Gases which burn with very high flame speeds, such as hydrogen, promote very rapid rates of burning within the chamber. The laminar flame speed of hydrogen is typically seven times that of methane or natural gas[8], while digester gas burns at half the speed of natural gas. If any hot spots occur within the combustion chamber, backfiring through the intake valve can occur during the induction stroke. Gases with very slow burning speeds, such as those with a high level of inert gases present, can delay burning to such a degree that the combustion is incomplete when the exhaust valve opens. This can cause extensive heating of the exhaust valve which may promote backfiring in the exhaust, or pre-ignition. Combustion flame speed is also very dependent

Table 25.1 Details of gaseous fuels used in reciprocating engines

Gas type	*Chemical composition*	*Gas net heating value kJ/m³ (15°C, 1 atm.)*	*Comments*
Methane	CH_4	33 900	Pure methane is often used as a reference gas for engine operation. Good detonation characteristics.
Propane	C_3H_8	86 400	Poor detonation characteristics compared to methane.
HD-5	≈ 90% C_3H_8 + other paraffins	≈92 000	Stored as a liquid (under pressure)
Commercial propane (LPG)	(70–90%) C_3H_8 + C_3H_6	≈86 000	Can contain C_3H_6 and C_4H_8 if produced from chemical plant. Stored as a liquid.
Butane	C_4H_{10}	112 400	Poor detonation characteristics.
Commercial butane (LPG)	(70–90%) C_4H_{10} + C_4H_8 + C_3H_8 + C_3H_6	≈110 000	Stored as a liquid
Natural gas (Pipeline/CNG)	CH_4 (90%) + heavier hydrocarbons + CO + N + CO_2	≈34 000–42 000	Composition can vary depending on location but predominantly methane. Pipeline gas is generally cheaper than refined liquid fuels.
Coal gas/town gas	H_2(30–55%) + CH_4(20–50%) + CO(5–15%) + CO_2(4%)	≈16 000–19 000 (30 000 if CH_4 content is high)	Traditional method of producing gas from coal: low conversion efficiency (≈25%) and coal with a high volatile content is required.
Wellhead gas/field gas	CH_4(60–95%) + heavier hydrocarbons + CO + N + CO_2 + H_2S	≈ 30 000–45 000	Can contain small amounts of H_2S (≈1%) These must be removed.
Digester gas (sludge/landfill/ sewage/bio gas)	CH_4(≈60–70%) CO_2(30–40%) H_2S (< 1%)	≈20 000–23 000	Anaerobic digestion of sewage waste. Conversion efficiencies typically 50%. Good detonation tendency due to CO_2. Low heating value requires an increase in gas supply to maintain output relative to methane. The heating value of a stoichiometric mixture is lower than for natural gas which will limit naturally aspirated output to typically 85%. Gas available at pressures just above atmospheric and so is generally used with naturally aspirated engines.
Gasifier gas	N_2 (50–60%) + CO (10–25%) H_2 (5–15%) + CH_4 + CO_2	≈3000–6000 (varies depending on feedstock)	Produced from carbonaceous (wood. coal) or vegetation waste. Gasification method is generally used. Good detonation characteristics due to large amounts of inert gases present. The stoichiometric mixture heating value is typically 75–85% compared to methane which causes a subsequent reduction in achievable naturally aspirated full load output. Gases available at atmospheric pressure or just above. Gas conversion efficiencies are typically 50–60%.
Blast furnace gas (see Gasifier gas)	N_2(50–60%) + CO(20–30%) + CO_2(8–15%) + H_2(2–4%)	≈3000–4000	Produced by pyrolysis of ligneous materials.
Producer gas (see Gasifier gas)	N_2(50–55%) + CO (30%) + H_2(12%) + CO_2(4%)	≈ 3200–6 500	Air is drawn past incandescent fuels such as coke charcoal, wood, etc. High conversion efficiencies achievable (90% +).
Refinery gas	C_3H_8(18–34%) + C_4H_{10}(45%) +CH_4(4–12%) + C_2H_6(10–20%) +H_2 (<60%) + CO_2	≈32 000–56 000	Gas type very dependent on location.

on the air–fuel ratio (lean mixtures reduce flame speed) and on turbulence.

As can be seen from *Table 25.1* there are many different gaseous fuels available. These can be split into two major types. The first contains gases which are either naturally occurring or are readily available. These tend to be predominantly hydrocarbon gases with high calorific values. Natural gas is the most commonly used fuel gas, and has methane as the major constituent. Owing to the presence of certain heavier hydrocarbon constituents, the detonation resistance of some gases can be low. Even natural gas can be affected in this way. It normally has a methane number of between 80 and 100, but it can be as low as 65. The effect of gas composition on (detonation limited) power output has been investigated by Gillispie and Jensen[9].

Gases of the second type are produced from various carbonaceous/biomass materials[10]. The various methods of production include thermochemical methods (dry) such as direct combustion, gasification and pyrolysis. and biochemical methods (wet) such as anaerobic digestion of bacterial fermentation. All these methods require specialized plant to produce the gas, so the cost effectiveness of such plant must be considered against the conversion efficiency and the plant size. The type of fuel to be used also has a significant effect on prime mover selection. These gases contain varying proportions of inert constituents (CO_2 and N_2) which tend to reduce the gas heating value, improve the detonation tendency, reduce the combustion flame speed and reduce the stoichiometric mixture heating value.

25.4 Combustion systems

There are three basic types of combustion system used in dual

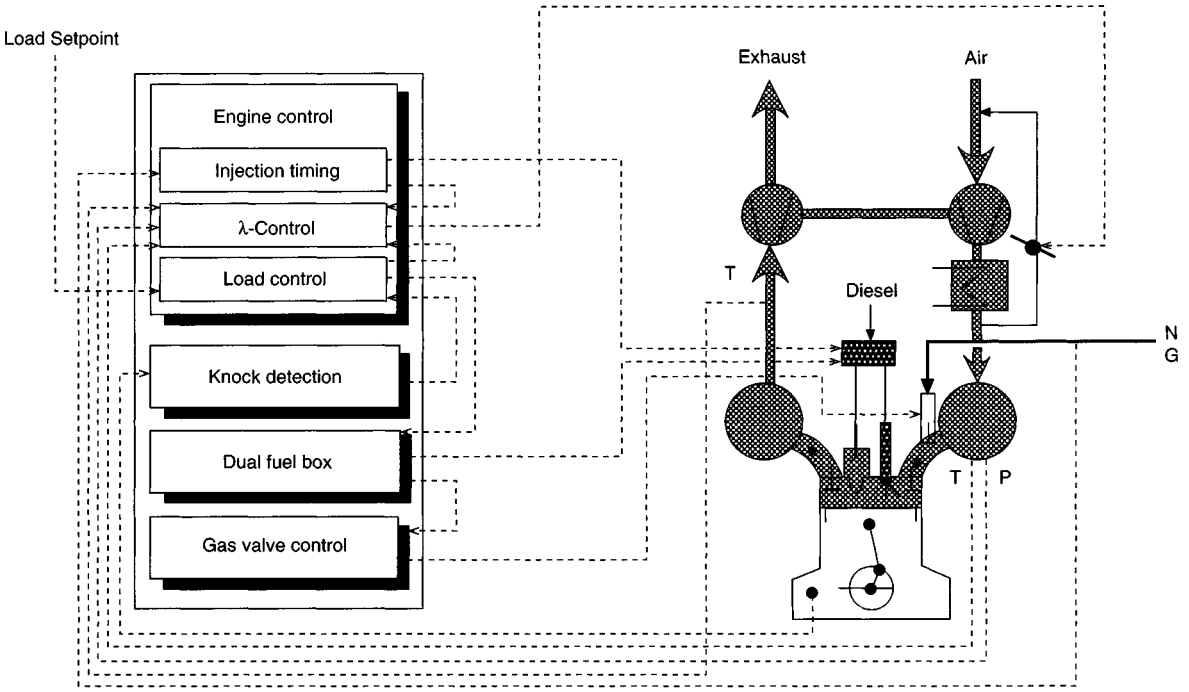


Figure 25.2 Engine management system (MAN B&W)

Table 25.2 Gaseous fuels—properties and characteristics

Gas type	*Chemical composition*	*Relative molecular mass*	*P* (kg/m³)*	*LCV (kJ/kg)*	*LHV* (kJ/m³)*	*Stoichiometric air–fuel* (kg/kg)	*Stoichiometric air–fuel* (m^3/m^3)	*Heat content of stoichiometric mixture* (MJ/m³)*	*Heat energy required to burn 1kg ideal air (kJ)*	*Methane Number (MN)*	*Minimum auto-ignition temp. in air (Ref. 2) (k)*
Methane	CH_4	16.04	0.6785	50 040	33 950	17.25	9.52	3.227	2900	100	813
Propane	C_3H_8	44.1	1.865	46 338	86 420	15.65	23.81	3.483	2960	35	723
Butane	C_4H_{10}	58.12	2.458	45 732	112 410	15.43	30.95	3.518	2964	10.5	678
Propene	C_3H_6	42.08	1.780	45 758	81 450	14.82	21.43	3.631	3087	20	733
Hydrogen	H_2	2.0	0.0853	119 810	10 220	34.18	2.38	3.023	3505	0	–
Carbon monoxide	CO	28.0	1.185	10 101	11 970	2.46	2.38	3.54	4106	73	–
Nat. gas	≈90% CH_4 + others	= 17.5	≈0.74	≈46 000	≈ 34 000	≈17.0	≈10.3	≈3.01	≈2705	≈90	–
Digester gas	≈65% CH_4 ≈35% CO_2	= 24.8	≈1.05	≈20 000	≈21 100	≈7.7	≈ 6.6	≈ 2.78	≈ 2600	≈ 130	–
Gasifier gas	≈55% N_2 ≈25% CO ≈10% H_2 + others	≈26.0	≈1.10	≈ 4700	≈ 5200	≈ 1.23	≈ 1.1	≈ 2.47	≈ 3821	≈ 130	–
Air	≈79% N_2 ≈21% O_2	28.96	1.225	–	–	–	–	–	–	–	–

*1 atmosphere 15°C.

fuel engines, each of which has advantages and disadvantages compared to the others. The performance of the three types in terms of BMEP and NOx emissions is compared with equivalent diesel and spark ignited (SI) gas engines in *Figures 25.3* and *25.4*. A description of the main characteristics of each type of combustion system, and some of their variants, is given in the following paragraphs.

25.4.1 The 'conventional' dual fuel engine

Early dual fuel engines were naturally aspirated and gas was mixed with the intake air by means of a carburettor. Today's turbocharged dual fuel engine, produced by Ruston, Allen and Niigata among others, uses timed gas admission into the inlet ports. Timed admission is used to prevent gas being lost in the scavenge air. The pressure of the gas supply to the injection valves needs to be a little above the boost air pressure, hence typical gas supply pressure is between 3 and 5 bar. The engine retains its standard diesel injection system, with perhaps some modification to the injection pump and nozzle to improve their performance at low fuelling rates. To prevent overheating of the nozzle due to prolonged operation with small quantities of fuel, cooled nozzles may be fitted. Depending on the quality of the gas fuel to be used, compression ratio may be lowered to avoid detonation, but must still be high enough to ensure reliable auto-ignition of the diesel pilot: 11 or 12 to 1 is normally considered to be the minimum. Maximum power output is typically 10-20% lower than the equivalent diesel engine. This is detonation limited and depends on the gas methane number.

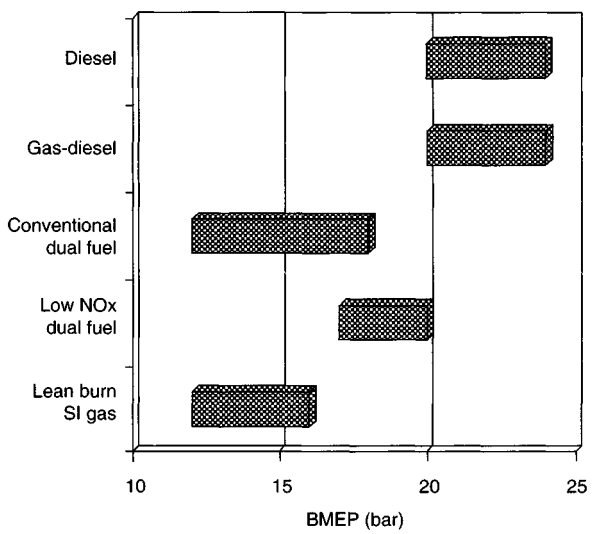


Figure 25.3 Typical BMEP range for various engine types

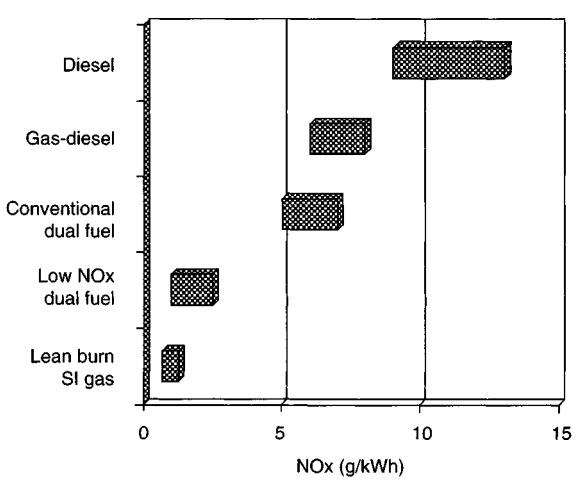


Figure 25.4 Typical NOx range for various engine types

For light load operation and start up, the engine is generally operated in the diesel mode only and, when at operating temperature, the diesel fuel is switched to the pilot fuel quantity (equal to or less than the idle fuelling level), the balance of the fuel requirement being provided by the gas. Any reduction in load at rated speed is then achieved by reducing the gas supply. A control system sometimes based on exhaust temperature is then used to regulate the air supply to maintain the required equivalence ratio.

The gas and air mixture is ignited by a finely atomized diesel fuel spray which is typically between 5% and 8% of the full load fuelling. By careful attention to the governing control system, the engine can, therefore, be operated at full load either as a conventional diesel engine or as a dual fuel engine. Furthermore the engine can automatically revert to full diesel operation if the gas supply is lost. This is beneficial if the full supply of gas to the engine cannot be guaranteed, as is often the case in dual fuel installations.

As these engines are normally turbocharged, equivalence ratios of 0.6 can be maintained to avoid excessive loss in output due to detonation. The high ignition energy from the pilot injection minimizes cycle-to-cycle variation in combustion, and the effects of changes in gas quality. However, any large change in air–fuel ratio may promote either detonation (richer) or misfire (leaner). Increasing the pilot fuel quantity may enable some extension to the lean misfire limit to be achieved owing to the higher ignition energy available and the reduced combustion period, but these effects are attenuated with increases in pilot fuel quantities above 15%. As pilot fuel quantities increase, NOx emissions will also increase owing to the greater amount of high temperature ignition sources present.

25.4.2 The 'low NOx' dual fuel engine

Lean burn gas engines have inherently lower NOx emissions than diesel engines, by virtue of lower peak combustion temperatures. However, in the case of the dual fuel engine, this is compromised to some extent by the pilot injection requirement. As a result, the Nox emissions of dual fuel engines lie somewhere between those of diesel engines and lean burn spark ignited gas engines (see *Figure 25.4*). Furthermore, very low values of equivalence ratio (of the order of 0.4) are required to achieve very low NOx levels. Lower equivalence ratio requires higher ignition energy, but in the dual fuel engine, this means more pilot fuel which itself is a significant contributor to overall NOx emissions. Several manufacturers, notably Cooper Energy Services[11], Coltec/Fairbanks Morse[12] and MAN B&W[13], have developed combustion systems which reduce the quantity of pilot fuel required, and hence the overall NOx emissions.

As described earlier, engines which use the normal fuel injection system to deliver the pilot fuel for gas ignition require at least 5% of the full load fuelling to achieve reliable combustion. This can be reduced to about 2% if a separate injection pump and nozzle, optimized for the pilot fuel quantity, are used. Higher injection pressures and smaller nozzle holes give better penetration and atomization, making more effective use of the fuel to provide a concentrated high energy ignition source.

Even better use of the small quantity of pilot fuel can be made by injecting the fuel into a prechamber. This amplifies and intensifies the available ignition energy, allowing pilot quantity to be reduced to less than 1%. Moreover, this can be

achieved together with equivalence ratios of 0.4 or less. However, a sophisticated engine management system is required to tread the fine line between detonation and misfire, while maintaining low NOx, as illustrated in *Figure 25.1*.

In the 1980s in the United States, increasing concern over engine emissions led gas engine manufacturers such as Cooper Bessemer and Fairbanks Morse to seek ways to overcome the relatively high NOx emissions from dual fuel engines. In the early 1990s, both companies published work on prechamber engines[12,14]. Subsequently, Fairbanks Morse have successfully applied this approach to their opposed piston, uniflow scavenged two-stroke engine. The general arrangement of this engine and its intake and exhaust systems is shown in *Figure 25.5*. Gas admission to a two-stroke engine cannot be made with the intake air, because a significant amount of this air escapes from the cylinder during exhaust scavenging. Gas must therefore be injected into the cylinder after the exhaust valves have closed.

The opposed piston arrangement means that the fuel injectors are located at the periphery of the combustion chamber. In the standard dual fuel engine, there are two liquid fuel injection nozzles and one gas injector. These are retained in the prechamber version, and a separate prechamber with pilot injector is added, as shown in *Figure 25.6*. Development of the prechamber system is described by Blythe[15]. The energy efficiency of the standard dual fuel engine is retained, with NOx levels comparable to the lean burn spark ignited engine (see *Figure 25.7*).

In co-operation with Fairbanks-Morse, MAN B&W have developed a prechamber dual fuel version of their 320 mm bore 32/40 engine. Development work focused on meeting the stringent German TA Luft limits for NOx emissions from gas engines (0.5 g/m^3 @ 5% O_2, equivalent to about 1.5 g/kWh) without exhaust gas after treatment. The main injector for diesel fuel is retained to permit operation on diesel fuel if required. A separate prechamber and pilot injector is fitted into the cylinder head near the outside of the combustion chamber, as shown in *Figure 25.8*. Fuel to the pilot injector is provided by a separate injection pump mounted alongside the main pump and operated by the same camshaft (*Figure 25.9*). An electronically controlled, hydraulically actuated gas valve is fitted to each inlet port as shown in Figure *25.10*.

The success of the prechamber approach is crucially dependent on getting the prechamber design right. The geometry of the prechamber profoundly affects the combustion process. The jet, or jets, of hot gas must have high energy and be spread over

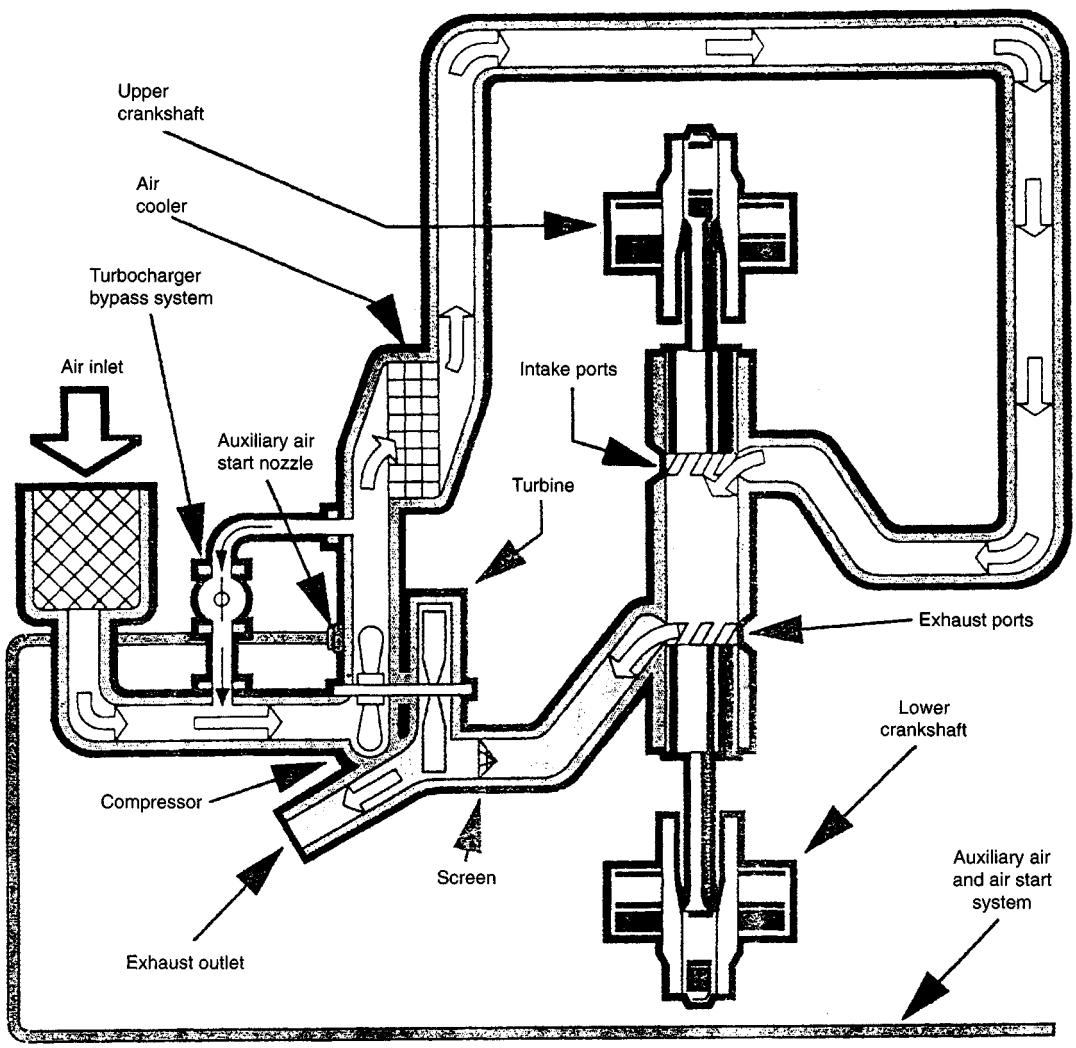


Figure 25.5 Fairbanks Morse OP engine—arrangement of intake and exhaust systems

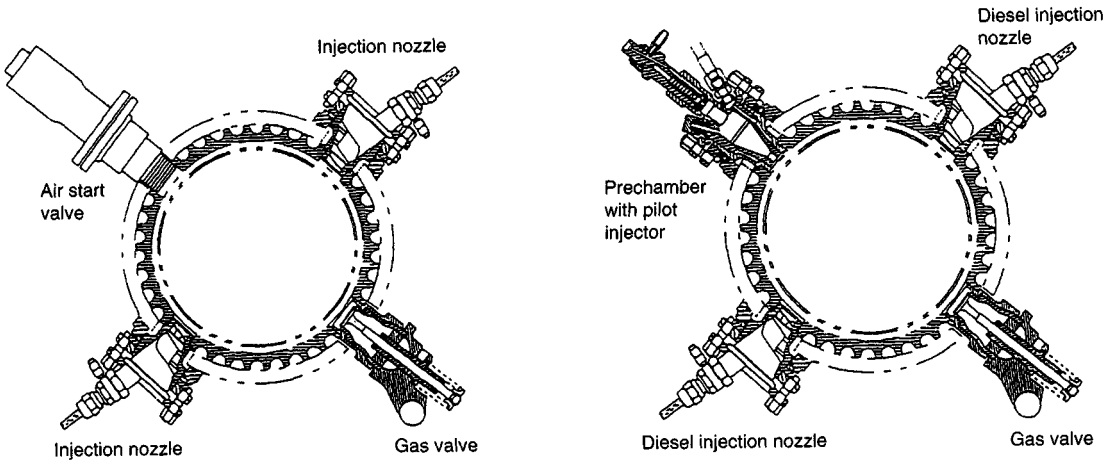


Figure 25.6 Fairbanks Morse OP engine—arrangement of fuel injectors

the combustion chamber space to ensure reliable and consistent ignition of a lean air/gas mixture of low flammability. Also the prechamber is subject to very high thermal loads, requiring careful optimization of geometry, materials and cooling to ensure adequate life.

25.4.3 The 'gas diesel' engine

A fundamentally different approach to burning gas in a diesel engine is taken by the 'gas diesel' engine. The concept is, quite simply, to replace the direct injection of liquid diesel fuel into the combustion chamber with the direct injection of gas fuel. In practice, of course, things are a little more complex than this.

Pilot injection of diesel fuel is retained to provide an ignition source. The required pilot quantity is 3 to 5% of full load fuelling, and NOx emissions are similar to the conventional dual fuel engine. There is no equivalent to the 'low NOx' engine. Because combustion is diffusive rather than pre-mixed, the problems of detonation and part load air–fuel ratio control are largely eliminated. The part load efficiency and full load BMEP of the diesel engine are retained and operation is largely unaffected by variations in gas quality.

However, there is a price to pay for these advantages. Direct injection of gas requires a high pressure gas supply (250 to 350 bar). The capital cost of the necessary gas compressor is high, and the power required to drive it can approach 5% of the engine output. Safety systems associated with the high pressure gas supply also add to the cost. The higher output and efficiency of the engine offset these costs to some extent, but the result is that the gas diesel is best suited to applications where high pressure gas is cheaply available, and diesel fuel is costly.

The main exponent of gas diesel technology for medium speed engines in recent times has been Wartsila Diesel of Finland. Development of a gas diesel variant of their 320 mm bore engine began in 1987. A paper presented in 1993[16] described the development work and field trials at three installations. The motivation for this development was a concept for power generation on floating oil production vessels called 'fuel off the well', the idea being that these vessels could produce their own energy from whatever gas or crude oil was available from the well.

A common central injector is used with twin nozzles (*Figure 25.11*). The gas injection is controlled electronically with hydraulic oil providing the actuating force for the gas needle. To prevent gas leakage through the needle clearance, sealing oil with higher pressure than the gas is applied in the clearance between the needle and the nozzle body.

The diesel fuel nozzle is designed to provide both good atomization with small pilot fuel quantities, and the same characteristics in full load operation on diesel fuel as a standard injector nozzle.

An important safety feature is an automatic quick closing shut-off valve in the high pressure gas line to each cylinder. If the gas injector needle does not close properly at the end of the injection period, the resulting pressure drop will immediately close the shut-off valve. The high pressure gas pipes are double walled. The space between the walls is continuously vented and equipped with gas sensors to detect any leakage. These and other safety systems are integrated into a comprehensive electronic control system. This automatically compensates for changes in fuel quality, and provides instantaneous switching from gas to diesel fuel if the gas supply is interrupted. Engines of 320 mm and 460 mm bore cover the power range 2 to 16 MW.

The gas diesel concept has also been applied to low speed two-stroke engines, notably by MAN B&W and Mitsui. Following the joint development by these two companies of a system for medium speed four stroke engines, the technology has been adapted to the MAN B&W MC range of engines[17,18]. These

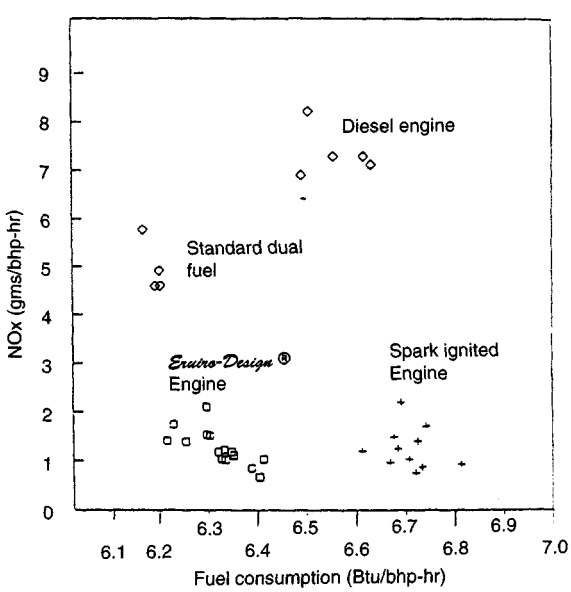


Figure 25.7 Fairbanks Morse OP engine—NOx emissions for different versions

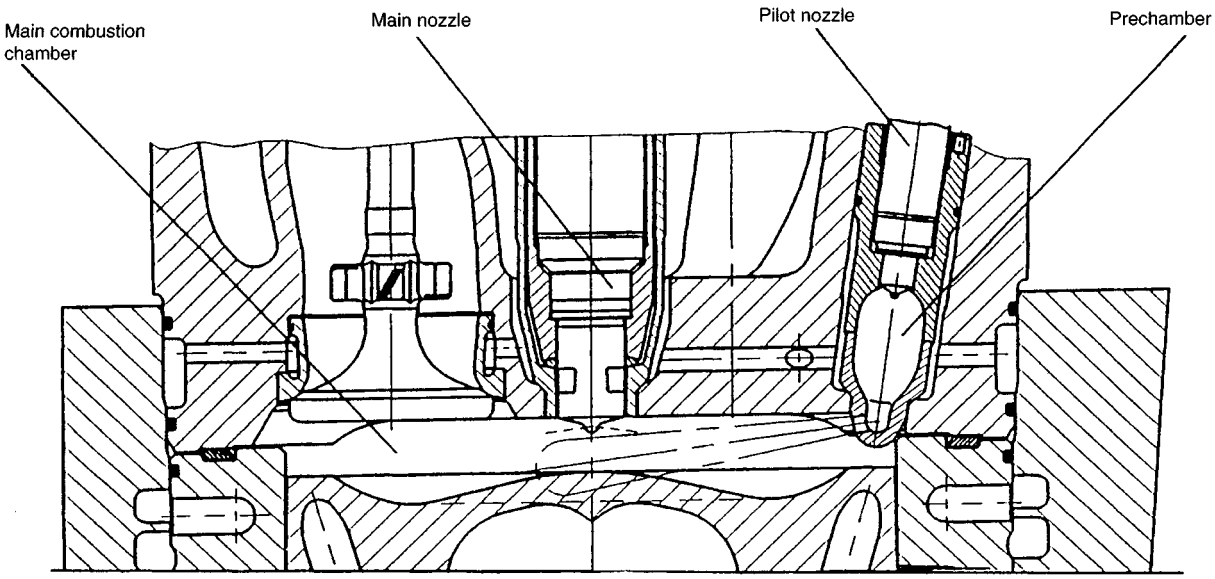


Figure 25.8 MAN B&W 32/40 DG combustion chamber (MAN B&W)

operate at speeds from 75 to 250 rev/min, with power output ranging from less than 10 MW to nearly 70 MW. A cross-section of one of these engines is shown in *Figure 25.12*, indicating the additional or modified parts for gas diesel operation. Separate injectors are used for diesel fuel and gas. The diesel fuel pump and injector may be either the standard items, where full load operation on diesel fuel alone is required, or smaller versions optimized for pilot fuel delivery, With the latter, the quantity of pilot fuel required to achieve stable gas combustion (and hence NOx emission) is reduced to about 8%. The pilot fuel may be either distillate diesel fuel or heavy fuel oil (HFO).

The ability of the gas diesel to burn gas that has poor detonation characteristics has been exploited for shuttle tanker propulsion. Shuttle tankers are widely used to serve offshore oilfields from which pipeline connections to the shore are not feasible. The shuttle tankers load their crude oil cargo either from storage facilities at the oilfield or directly from the production platforms and loading buoys at the oilfield. The crude oil cargo is then transported to oil refineries or terminals ashore close to the oilfield. During the handling of the oil, and in particular during loading, large quantities of the light components of the oil evaporate. These oil vapour are known as Volatile Organic Compounds (VOC). The VOC emission during the voyage is not normally significant due to the short sailing distance.

The high content of propane and the higher hydrocarbons in such gases lead to methane numbers which may be very low (close to zero) and which vary from one oilfield to another, making it unrealistic to utilize this in an engine employing the pre-mixed combustion system. Thus, the only technology available for using VOC as fuel is high pressure injection of the VOC directly into the cylinders.

In a joint venture with Statoil, MAN B&W have developed and adapted the MC gas injection systems for VOC use[19]. The normal (HFO) fuel system is retained for use when the VOC gas is not available.

The VOC utilization system is shown schematically in *Figure 25.13*. Because the tendency to release VOC is greatest during the loading of the crude oil, the VOC has to be converted to a form that can easily be collected and stored until the engine can use it.

The VOC and inert gases emitted from the crude oil tank are therefore conducted through gas pipes from the crude oil tanks to the VOC treatment and collection facility consisting of a cleaning system and a gas condensation system. The condensed hydrocarbons are separated from the lighter hydrocarbons and inert gas, and transported to a storage tank. Liquid VOC is taken from the tank, pressurized and supplied to the engine.

25.4.4 Other combustion systems

The General Electric 'H-Process'[20] is a hybrid of the dual fuel and gas diesel approaches, in an attempt to combine the benefits of both. The combustion chamber layout is shown in *Figure 25.14*. At light loads it operates as a gas diesel, with high pressure direct injection and diffusion burning, thereby avoiding the poor efficiency and other problems associated with operating dual fuel engines at part load. At high loads, however, all the gas is injected before TDC so that homogeneous charge combustion is obtained with associated low levels of NOx emissions. Because the gas is injected late in the cycle, pre-mixing time is limited and detonation problems can be avoided without the need to reduce compression ratio. Hence the same full load power and efficiency of the diesel engine can be achieved on gas. Full load NOx is about half that of the equivalent diesel engine.

Furthermore, because the gas is injected when cylinder pressure is lower than is the case for the gas diesel, the pressure of the gas supply can be correspondingly lower, reducing the amount of energy required to compress the gas and hence improving the overall system efficiency.

The EMD two-stroke diesel engine is widely used as a locomotive power unit in North America and elsewhere. In the interests of both utilizing alternative fuels and reducing NOx emissions, a number of dual fuel combustion systems have been investigated for this engine.

As described earlier, gas admission to a two-stroke engine must be made into the cylinder after the exhaust valves have closed. If the gas is injected as early as possible, this can be done at low pressure (about 7 bar) and conventional, homogeneous charge combustion results. Alternatively, the gas diesel approach of high pressure injection and diffusion combustion can be used. A number of variations on these themes were investigated in a research programme called "GasRail USA" in 1993[21].

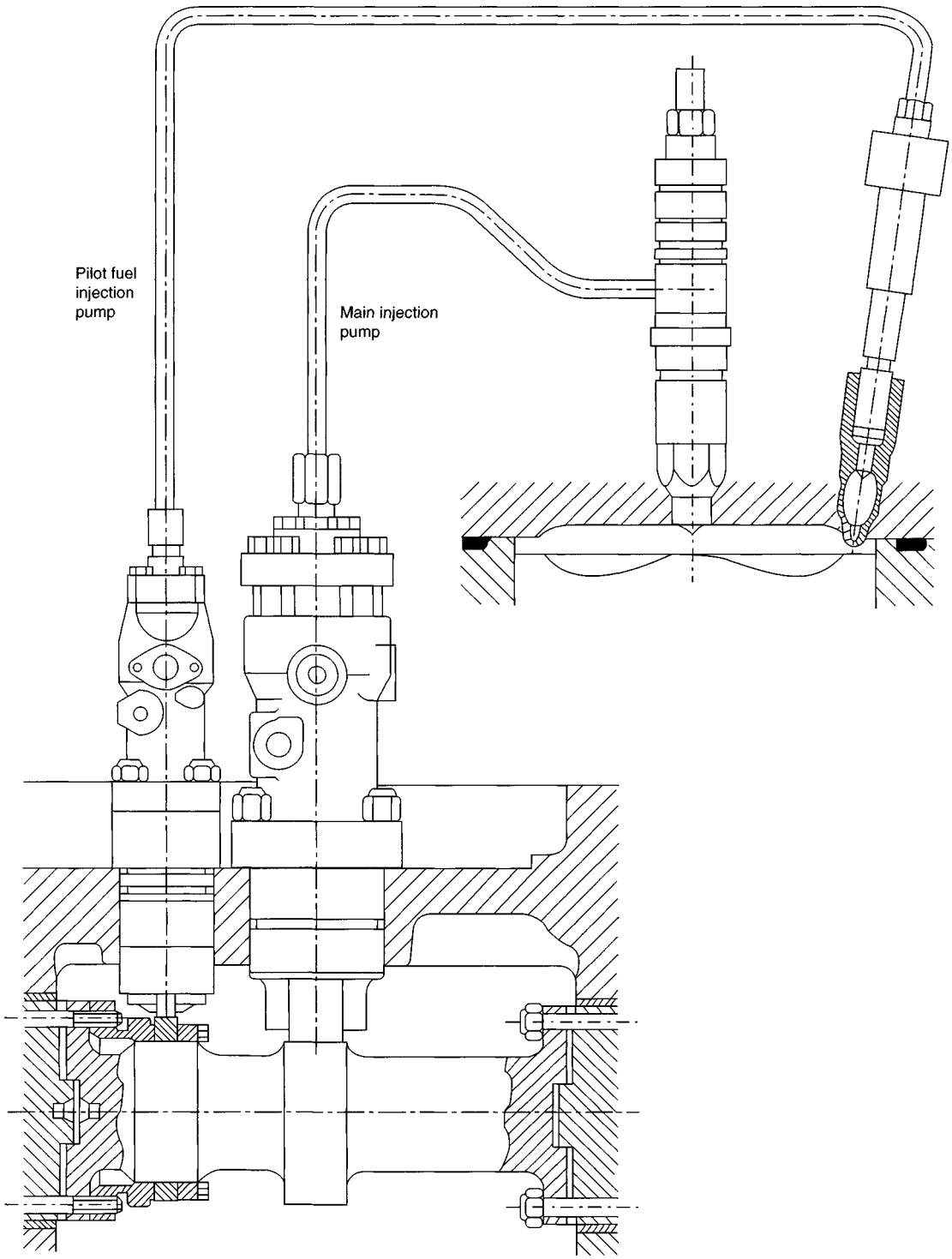


Figure 25.9 MAN B&W 32/40 DG fuel injection system (MAN B&W)

Subsequently, locomotive trials have been conducted using both the dual fuel and the gas diesel approach. The gas diesel was chosen for a trial with four locomotives[22] using liquefied natural gas (LNG). NOx emissions of below 5 g/kWh were reported. A beneficial side effect of using LNG is that the gas can be pumped to the required pressure whilst in the liquid state, then heated to turn it into a gas.

EMD two stroke engines used in stationary applications have also been converted to dual fuel operation using low pressure gas[23,24].

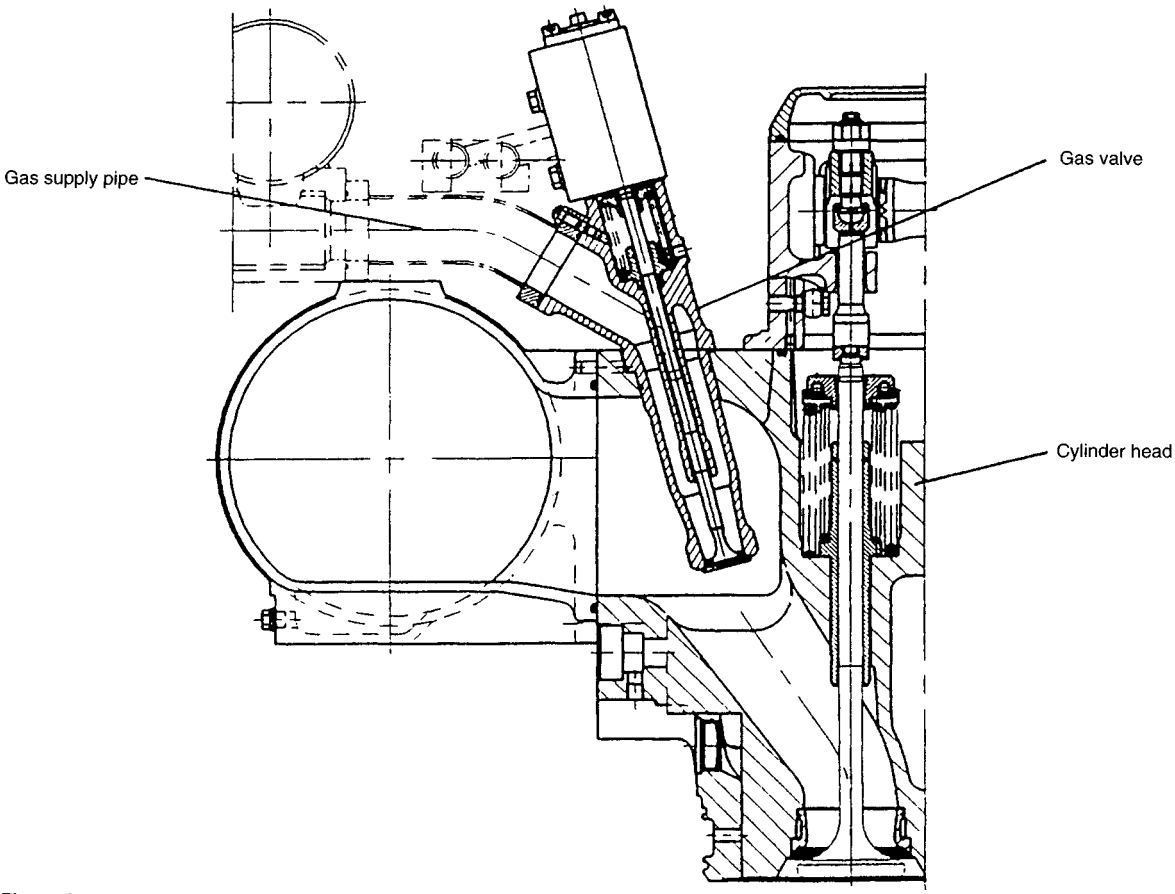


Figure 25.10 Location of electronically controlled hydraulic gas valve (MAN B&W)

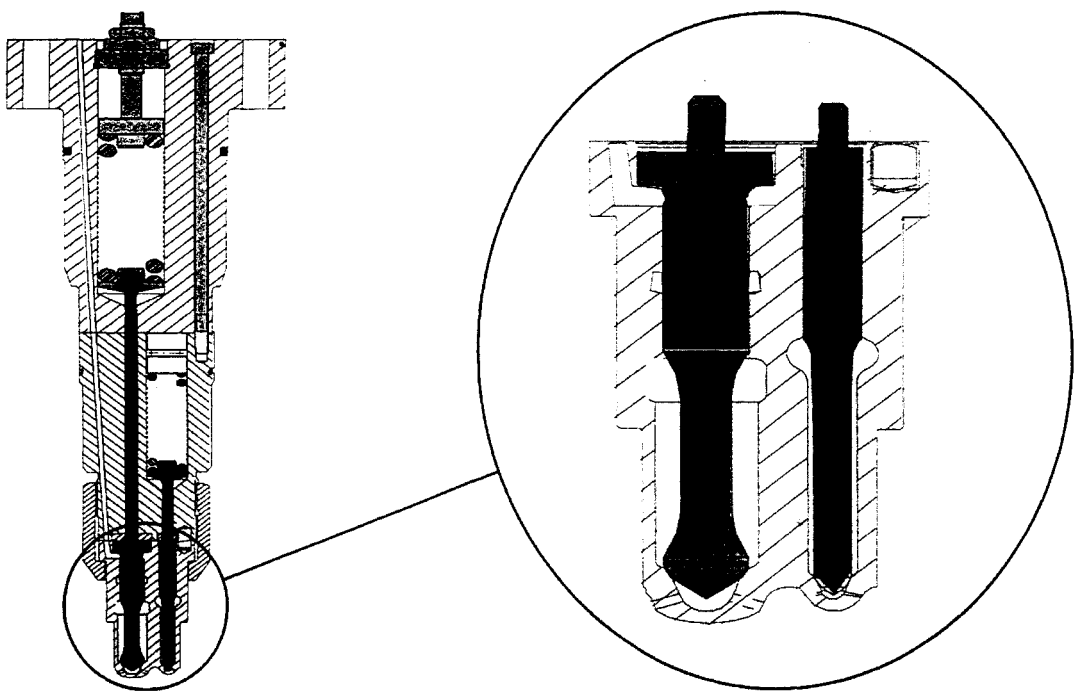


Figure 25.11 Twin needle combined gas and liquid fuel injector (Wartsila)

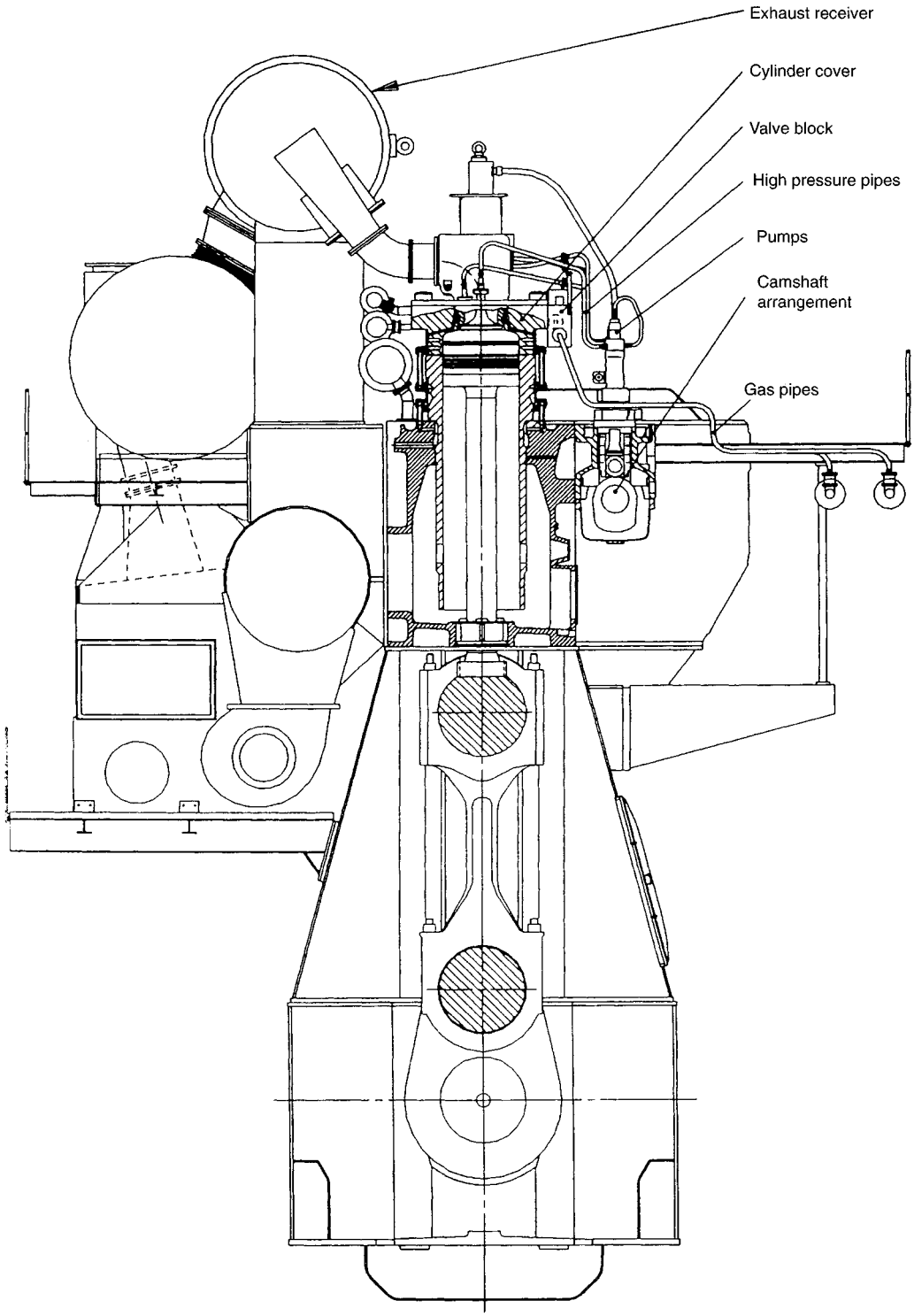


Figure 25.12 MAN B&W MC-GI dual fuel engine, showing additional or modified components (MAN B&W)

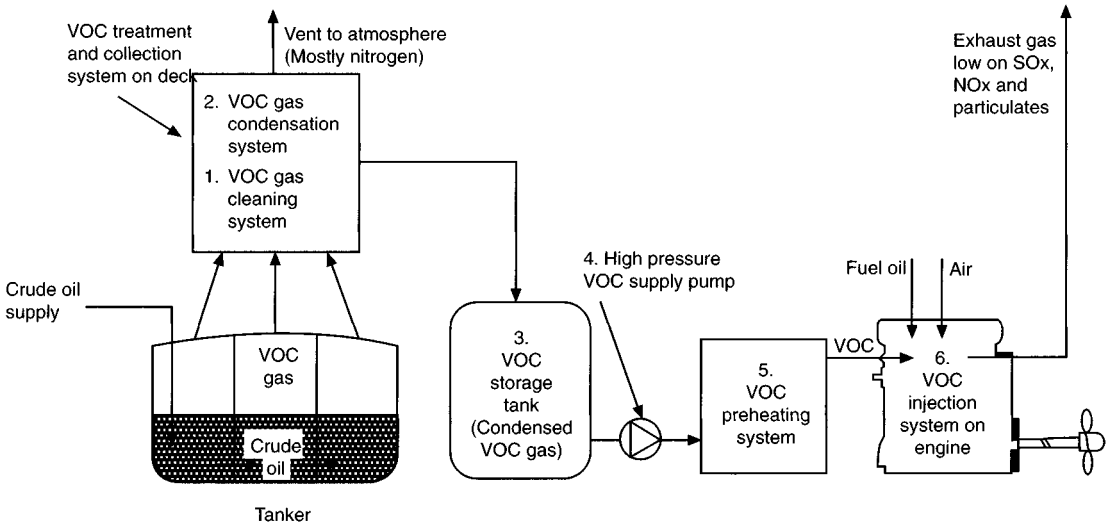


Figure 25.13 Schematic diagram of VOC utilization system (MAN B&W)

25.5 Air–fuel ratio control systems

With the exception of the gas diesel, dual fuel engines utilize pre-mixed combustion of the gas and hence require some form of air flow control to maintain air–fuel ratio below the misfire limit. There are several methods of achieving this.

25.5.1 Intake throttle

Such a system is necessary for engine operation over a wide range of constant air fuel ratios. As the throttle will act to increase the compressor discharge pressure and reduce the air flow, it will move the operating point towards surge. It is therefore important to match the compressor to give surge-free operation across the engine range while maintaining good turbocharger efficiencies at rated conditions.

25.5.2 Exhaust by-pass

This method is most suited to constant load control because a boost pressure signal can wastegate the turbocharger to maintain constant load. This system is also used by one American manufacturer to regulate air manifold pressure as a function of gas pressure (load) and engine speed.

25.5.3 Compressor by-pass

This can enable control to be applied without affecting the compressor operating line. It is widely used on dual fuel engines where the control of air supply at very low load is not required, because most dual fuel engines revert to diesel operation under these conditions.

25.6 Safety systems

The following paragraphs are derived from an MAN B&W brochure[25] describing the gas diesel variant of their low speed two-stroke MC engines.

All the normal safety systems incorporated in the diesel engine are retained for dual fuel operation; However, additional safety devices and systems are incorporated to avoid situations which might otherwise lead to failures.

There are numerous systems around the engine which have to be incorporated in an overall control and safety system and which could trigger off a shutdown of the gas operation or even a complete engine shutdown. The most complicated of these is the gas compressor system, while the sealing oil system, the ventilation system for the double wall piping, and the inert gas system are relatively simple. The engine systems also have to be integrated with, or at least connected to, the plant's overall control and safety system, so it is important that the interface between the systems is clearly defined in each case.

In the engine room of a ship or the engine hall of a power station, the presence of gas will be detected by a hydrocarbon (HC) analyser. The ventilation system of the intervening space of the double wall piping also incorporates an HC analyser, where an alarm is set off at a gas concentration of 30% of the lower explosion limit and shutdown occurs at 60%.

During operation in the dual fuel mode, malfunctioning of the pilot fuel injection system or gas injection system may involve a risk of uncontrolled combustion in the engine.

In the event of a damaged gas spindle seat or the sluggish operation of the gas valve, the exhaust gas temperature will increase considerably, and an exhaust temperature sensor will set off an alarm with subsequent shutdown of the engine. In the case of very sluggish operation or even seizure of the gas valve spindle in the open position, there is a risk that large quantities may be injected into the cylinder and when the exhaust valve opens, a hot mixture of combustion products and gas flows out into the exhaust pipe, where it will burn relatively slowly, when mixing with the scavenge air/exhaust gas there. The temperature sensor will then set off an alarm for high exhaust gas temperature for the cylinder in question, with subsequent engine shutdown. In the highly unlikely event of a more intensive combustion in the exhaust gas receiver, the pressure will increase considerably. For stoichiometric reasons, this pressure will not exceed 15 bar. The exhaust gas receiver is designed to maintain its integrity at this pressure.

However, the engine safety system will detect defective gas valves. The gas flow to each cylinder during one cycle is monitored by measuring the pressure drop in an accumulator in each gas supply line. If the pressure drop exceeds the value at full load, the engine will shut down immediately. With this system, any abnormal gas flow, whether due to seized gas injection valves or fractured gas pipes, will be detected immediately, the gas supply will be discontinued, and the gas lines will be purged with inert gas.

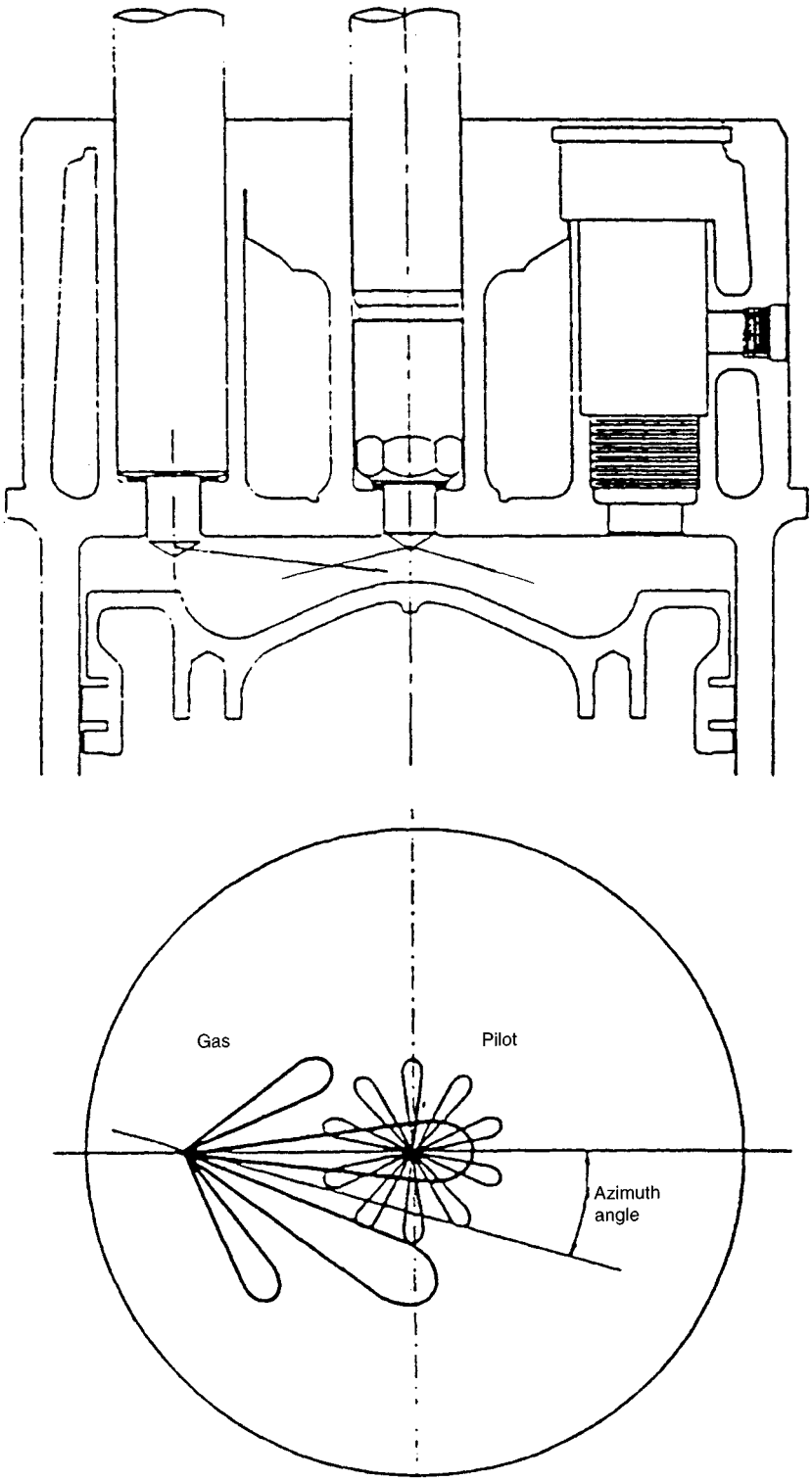


Figure 25.14 The 'H-process' combustion chamber layout (General Electric)

Ignition failure of the injected natural gas can have a number of different causes, most of which, however, are the result of failure to inject pilot oil into a cylinder. Examples are; leaky joints or fractured high-pressure pipes, a seized plunger in the pilot oil pump, a malfunctioning suction valve in the pilot oil pump, a failure of pilot oil supply to the engine, or other faults in the engine forcing the fuel pump to the zero fuelling position. If any of these faults are detected the gas injection is stopped immediately.

A safety valve ensures that gas injection can only take place after pilot oil injection. Between injections, the safety valve is in the open position and to enable gas injection, the safety valve must close. When the pressure on the pilot side rises to the opening pressure of the pilot valve, pilot oil is injected and the oil pressure acting on the safety valve spindle closes it, thus enabling the control oil pressure to build up and open the gas valve.

Despite this system, pilot fuel can be injected without being ignited if there is a sticking or severely burned exhaust valve, provided that this involves so much leakage that there is not enough compression pressure to ensure ignition of the pilot oil. However, the burning of an exhaust valve is a rather slow process extending over a long period, during which time the exhaust gas temperature rises and sets off an alarm well in advance of any risk of misfiring.

25.7 Applications

25.7.1 Automotive

Dual fuel engines have not been widely used in automotive applications. The relatively high cost of two separate fuel systems, and the problems of finding space for adequate quantities of

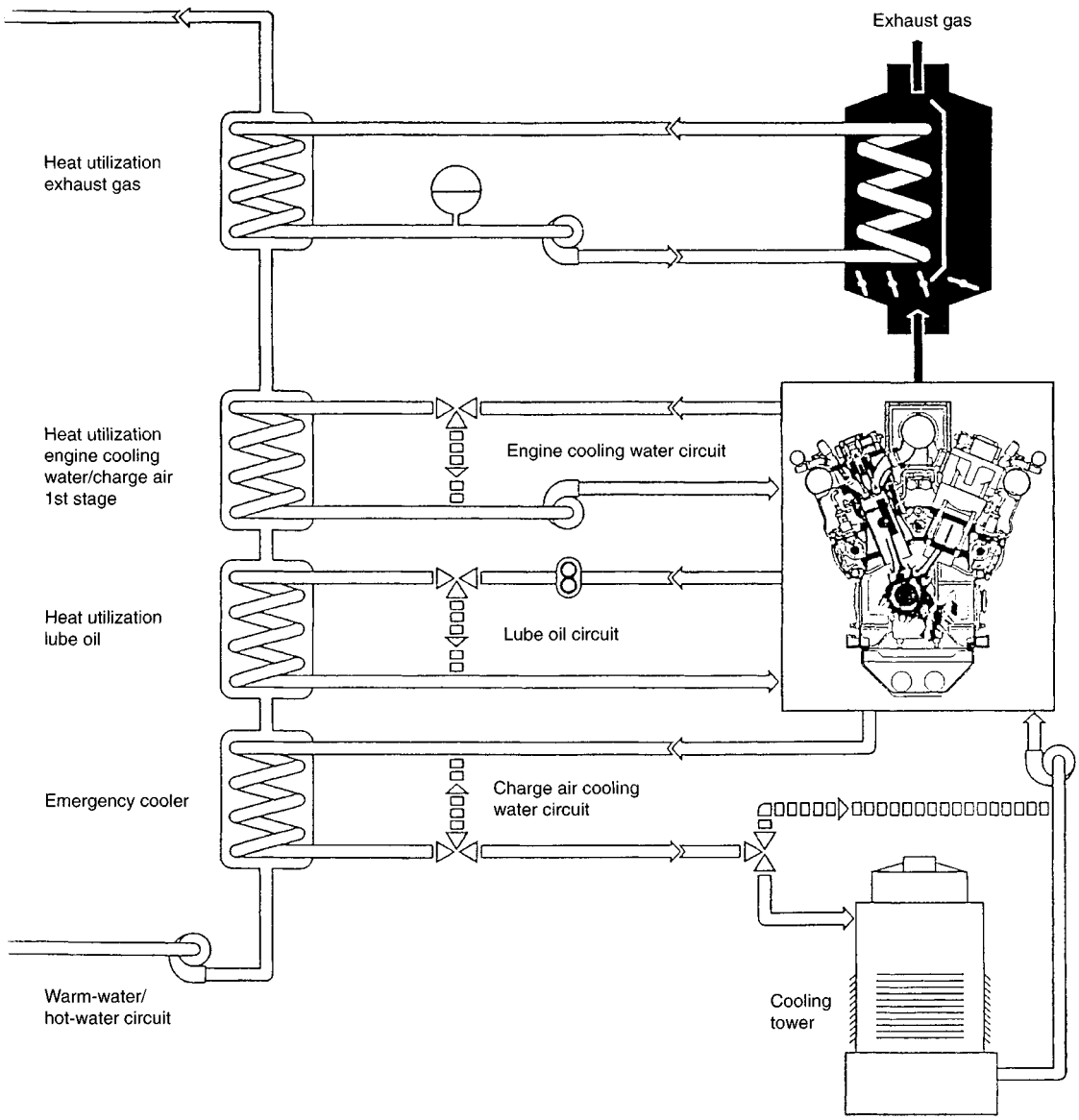


Figure 25.15 Typical waste heat utilization system based on a dual fuel engine (MAN B&W)

each fuel, have restricted their use to special applications such as city buses[26].

Conversely, where there are strong incentives for gas fuelled vehicles, such as emissions reduction and alternative fuel programmes, having a dual fuel engine does mean that the vehicle can continue to operate if its gas supply runs out. Target vehicles for alternative fuels programmes are normally those which return to a depot each day where refuelling facilities can be made available. Dual fuel vehicles give more flexibility.

Other advantages over spark ignited gas engines include higher thermal efficiency and lower conversion costs where a diesel engine is used as the starting point. However, the problems of part load operation have been a barrier; automotive engines spend much of their time operating at light load. Much research and development work has been done to try to resolve these problems, notably by BKM Inc.[27,28].

A significant step forward on the automotive front took place in 1996, when Caterpillar announced a new series of dual fuel engines in the power range 140 to 320 kW[29]. These engines are variants of existing electronically controlled diesel engines, with a gas fuel injection system and additional electronic control module. These engines are conventional dual fuel engines (without prechamber) and pilot fuel quantity is about 15% in most applications. Prechamber versions are being developed.

25.7.2 Locomotive

Many of the arguments stated above for automotive applications also apply to railway locomotives. However, there has been a strong focus on locomotive emissions in California, and this has led to a number of development programs aimed at utilizing CNG and LNG in locomotives[20,22,30,31]. A significant part of this effort has been devoted to the widely used General Motors EMD two stroke engines, as described earlier.

25.7.3 Stationary (power generation and mechanical drive)

The market for dual fuel engines for power generation purposes has increased as a result of more widespread introduction of emissions regulations for stationary plant, and the increasing availability of pipeline natural gas. This is the main area of application for the prechamber low NOx engine.

The dual fuel engine also finds favour in situations where gas is cheaper than diesel fuel, but the supply is unreliable, such as oil and gas pipelines and wellheads[24], offshore platforms[32], and biogas sources (landfill sites and sewage works)[10]. Also falling into this category are the increasing number of situations where gas is available at relatively low cost on an "interruptible supply tariff", whereby the gas supply may be cut off from time to time.

Dual fuel engines are well suited to cogeneration applications, providing both heat and electrical power from a low emissions prime mover. A typical waste heat utilization system to provide hot water for process or district heating is shown in *Figure 25.15*. One application where there is a true synergy between engine and fuel source is in sewage treatment works, where recovered heat from engine exhaust and jacket water can be used to maintain digester tanks at the required temperature for optimum gas production.

In addition to power generation, dual fuel engines are also used for driving pumps and compressors. Gas pipelines require compressor stations at intervals along their length. Because the gas is available at a relatively high pressure (typically 70 bar), this is an area where the gas diesel is sometimes preferred[33].

25.7.4 Marine and offshore

Marine applications of dual fuel engines are limited to those where gas is available from the cargo, such as LNG carriers. Both conventional dual fuel[34] and gas diesel[25] variants of low speed two-stroke propulsion engines have been developed to compete with the traditional steam turbine propulsion systems for this application.

The system developed by Statoil and MAN B&W for recovering and using the hydrocarbon gases (VOC's) given off by the cargo of crude oil shuttle tankers[19] has already been described.

Offshore oil platforms provide an ideal application for dual fuel engines. Gas is available at little or no cost, but the supply cannot be relied on. Likewise, crude oil may be available from the well or from storage tanks, and a certain amount of (expensive) distillate or heavy fuel is available for use when no other fuel is available. This is the application for which Wartsila developed their gas diesel engine. A modern alternative to the fixed oil platform is the floating oil production vessel, with multiple production, storage and processing functions[32], where a multi-fuel engine is even more beneficial. The ability of the gas diesel to deliver consistent performance on a range of different fuels, and to switch quickly from one to another, makes it very well suited to this application.

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