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Diesel engines in hazardous areas on offshore installations

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SYNOPSIS

A risk associated with the use of diesel engines in potentially flammable atmospheres is now well recognised. A diesel engine which is intended for operation in hazardous areas on offshore installations is considered to be a source of the possible ignition of a surrounding flammable atmosphere with subsequent fire or explosion on an unpredictable scale. The risk could be significantly reduced by the design and construction of special protective arrangements for diesel engines in order to minimise the effects of an immediate, possibly flammable, environment.

It is impossible to achieve absolute protection. However, reasonable or acceptable safety, which is a compromise between the cost of protection and the risk involved, can be achieved.

One of the main problems of protection of diesel engines lies in the interpretation of the concepts of 'acceptable safety' and 'reasonable protection'. Requirements for the protection of diesel engines, which are dependent upon a company's code of practice or other recommendations, differ considerably and in some cases this leads to over- or under-protection. This also indicates an absence of uniform criteria for an acceptable level of protection.

The author considers possible hazardous situations by applying principles of combustion theory and gas dynamics and by analysis of experimental data on gas combustion and, finally, identifies areas of risk and recommends methods of protection of diesel engines.

Thus is is hoped that this paper will contribute to a better understanding of the problems associated with the safe use of diesel engines on offshore installations.

INTRODUCTION

Diesel engines are widespread in the offshore industry as prime movers for various types of equipment. Their uses include the source of power for cranes and transportable power packs used for wire-lining, well servicing, etc. Operational conditions very often require diesel engines to be located in areas designated as 'hazardous' due to the possible presence of flammable gases in the surrounding atmosphere. A diesel engine could provide sources of ignition and initiate combustion of a flammable surrounding atmosphere. In this case the classical combustion triangle of fuel, source of ignition and atmospheric oxygen is present.

To break this combustion triangle, there are two options. The first is to separate sources of ignition from fuel (in this case flammable gas). This can be achieved by installation of the engine in a pressurised enclosure where the air intake for pressurisation is arranged from a safe area and exhaust gases from the engine are discharged to a safe area. Usually, permanently installed engines driving equipment such as generators, fire pumps, etc are protected by pressurised enclosures. Requirements for such engines and their pressurised enclosures are clearly outlined in the UK Department of Energy Guidance Notes on Design and Construction of Offshore Installations and will not be considered here.

The second option is to eliminate possible sources of ignition within the engine and prevent the spread of combustion to outside the engine if ignition does occur from a source which cannot be eliminated due to the nature of the engine. This method of protection is usually used for non-fixed and transportable engines such as crane engines and various diesel-driven power packs intended for use in zone 2 hazardous areas as defined in BS 5345 or IEC Publication No 79. This method of protection will be considered further in the text.

Non-fixed and transportable engines could, for illustration purposes, be placed at the bottom of an 'incident iceberg' as they operate in 'unsafe conditions' due to the possible presence of a flammable mixture (see Fig 1).

An accumulation of circumstances such as the occurrence of a flammable atmosphere, the presence of a source of heat and ignition, can lead to an explosion with disastrous effects (top of incident iceberg).

At the present time, despite some progress achieved in the construction of protection systems, there are still considerable problems involved with providing assured protection to diesel engines offshore. In the following text diesel engines are considered, operating in a hazardous, flammable environment; stage 3 of the incident iceberg.

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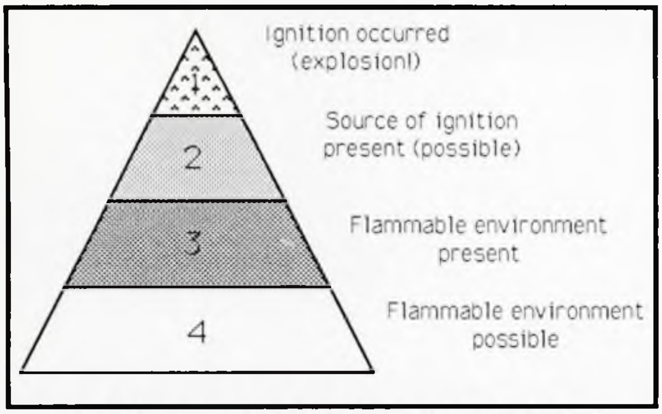


Fig 1: Incident iceberg of operating conditions

THE NATURE OF HAZARDS

Every conventional, unprotected diesel engine can be considered to be hazardous. The nature of the hazards are as follows.

Excessive surface temperature of the engine parts and exhaust gases

The exhaust gas temperature of diesel engines varies depending on the design, construction and load of the engine, and could be as high as 500°C. The surface temperature of the exhaust manifold and piping depends on the exhaust gas temperature and construction of the exhaust system. Under certain conditions, such as bearing failure or mechanical damage, surface temperatures of some other parts of the engine could also be excessive.

Excessive temperatures of exhaust gases and engine parts could exceed the auto-ignition temperature of the mixture of gases involved. The auto-ignition temperature of the mixture depends on the chemical composition, concentration and physical phase in the air, and the heat transfer from the hot surface to the flammable gas–air mixture. A knowledge of the auto-ignition temperatures of the various gases in the different areas of a platform and their most flammable mixtures with air are essential for determining the safe surface and exhaust gas temperatures of diesel engines.

Discharge of sparks and flames

In exhaust systems carbon deposits can build up under normal operation of the engine and especially when an engine develops a combustion fault.

When burning carbon deposits and rust are blown out by exhaust pressure, they discharge as sparks which may be hot enough to ignite a flammable surrounding atmosphere.

A direct flame path exists in the engine's cylinder head during the overlapping of inlet and outlet valves and this could be the cause of ignition of flammable gases in the induction manifold. The possible consequences of the ignition could be stabilised combustion, or combustion with fast moving flames, or even explosion or detonation with final blowout flames to the atmosphere. Ignition could occur in the exhaust system with the same effect. The discharge of flames and sparks can be through joints and penetrations in exhaust and induction systems and cylinder head joints.

Overloading of the engine could also be the cause of the discharge of flames and sparks. Thermal overloading due to the increase of mechanical load on the shaft, or inefficiency of the cooling system, can increase exhaust gas and surface temperatures, as well as damage the engine mechanically, causing discharge of sparks and flames. Speed overloading could occur when the engine ingests a flammable mixture and continues to run even after the fuel has been shut off. In this case, the speed of the engine becomes uncontrollable resulting in the engine being overheated and finally destroyed with the discharge of sparks and flames.

The engine crank case has always been considered as a source of hazard since the oil–vapour–air mixture becomes very explosive when heated to a certain temperature. In the case of a failure of the lubrication oil system or bearing, a rapid rise in bearing temperature could provide hot spots for ignition of the oil–air mixture and an explosion blowing out flame.

Frictional and incendive sparking could occur when rotating parts of the engine come into frictional contact with stationary parts, eg when the radiator steel fan comes into contact with the surrounding steel casing.

Diesel engines with light aluminium alloy parts could generate incendive sparking. Aluminium alloys contain aluminium and magnesium which are very good de-oxidisers. They reduce iron oxide (rust) to iron and combine with the released oxygen to form aluminium oxide. This is a strongly isothermic reaction which may produce a large volume of incendive sparks. The incendiary sparking could be generated by friction between aluminium and steel, for example when an aluminium alloy radiator fan contacts with steel casing.

Rotating parts of an engine with belt drives could generate and discharge static electricity in the form of sparking. Electrical equipment fitted to the engine can also produce electrical sparks with very high energy. Electrical starters, or generators of engines, can produce such sparks.

Mechanical vibration of the engine and associated piping systems can be the cause of frictional contact of engine parts and loose joints in piping. This could create additional hazards by producing sparks and magnifying the hazards described above.

Careful investigation of all possible hazards, understanding the processes of their possible development, assessment of the magnitude of hazards and the probability of their occurrence, are the initial factors to be considered in the design of effective protection for diesel engines on offshore platforms.

APPLICABLE THEORY FOR PROTECTION OF DIESEL ENGINES

Before designing effective protection for diesel engines, it is necessary to assess quantitatively the scale and probability of expected hazards. This can be done by using basic principles of the theory of combustion of gases and gas dynamics as well as the principles of probability theory. On this basis the theoretical study of ignition and combustion processes of flammable mixtures in connection with diesel engines are considered in the following section.

Ignition

A major hazard on offshore installations may be expected from the escape of production gases. Therefore further consideration of the combustion properties of the gas will mainly be related to natural gas with a chemical composition of Bacton gas (95% methane, 3.5% ethane, 1.5% nitrogen and a small percentage of butane and other gases). However, the possible presence of various vapours and other gases on offshore platforms during diesel engine operation should not be ignored when determining safety aspects of such operations.

It is well to mention that not every gas–air mixture can be ignited – only when there is a certain percentage of gas in the air. The range between the minimum and maximum percentage of gas in air that can be ignited is usually called the range of flammability, and the minimum and maximum percentages are known as the lower and upper flammability (or explosion) limits. For natural gas–air mixtures the range of flammability is between 3.8 and 17% of gas by volume in air at ambient temperature and atmospheric pressure. For methane it is 5.5–15%.[1] The most violent reaction occurs when there is enough oxygen for complete combustion, ie the correct combustion mixture or stoichiometric mixture. The concentration of gas that forms 0.9–1.0 of the stoichiometric mixture is considered to be the most dangerous as it needs minimum energy for ignition. For natural gas this is 9.5–10% of gas by volume in air. It was shown by experiment[1] that the lower limit of flammability reduces by 10% of that determined under ambient conditions each time the temperature of the explosive mixture is increased by 100°C. This is very important when considering diesel engines since explosive mixtures at certain parts of the engine could be at temperatures of 300 to 500°C and therefore the lower limit of flammability could be reduced by 40–50%.

It is well known that for initiation of ignition of an explosive mixture it is necessary that a certain minimum energy be released by the heat source. This minimum energy, called minimum ignition energy, is transferred to molecules of explosive mixture and activates them. Activated molecules collide with other molecules of explosive mixture and initiate the chemical reaction of combustion. The difference between the energy of activated and inert molecules is called the activation energy. The combustion process could be described on the basis of the kinetic theory of gases and the more modern chain reaction theory of combustion. It must be noted that here the author considers ignition conditions not as a process, but as a particular moment, 'frozen in time', when ignition takes place.

When a reaction of combustion has started it is self-propagating if the following condition is fulfilled:

$$q_p > q_w$$

where q_p = heat released by chemical reaction of combustion in unit time;
q_w = heat lost to the surroundings.

According to Lewis and Gaydon[2,3], the minimum ignition energy, E_{ign}, is a function of the temperature gradient of the gas ($T_e - T_0$), thermal conductivity (λ), minimum flame diameter (quenching diameter, d), and standard flame speed (υ_s). It is difficult to find a positive relationship for these parameters. However, Lewis derived such a relationship which takes the form:

$$E_{ign} = \pi d^2 \frac{\lambda_0 (T_e - T_0)}{\upsilon_s} \qquad (1)$$

and proved to be in approximate agreement with experiments and can be used for practical purposes for hydrocarbon flames.

Lewis, during experiments with electric spark ignition, found that the minimum ignition energy for methane–air mixtures is about 0.8×10^{-3} J at atmospheric pressure. Other sources, such as ref 3 (more recent and probably more accurate), quote that hydrocarbon–air mixtures have a minimum ignition energy of $0.2–0.3 \times 10^{-3}$ J at atmospheric pressure. In any case an electric spark of such energy is so small that one cannot see it with the naked eye.

Ignition depends very much on the nature of the heat source. As previously mentioned, diesel engines provide heat sources for ignition and the mechanism and parameters of ignition of flammable, explosive gas–air mixtures by these sources is different. However, common characteristics for all types of ignition are as follows.

1. Ignition depends upon the physical and chemical state of the explosive mixture.
2. Ignition is possible only within the range of the flammability limits.
3. The heat energy of the source of ignition (such as temperature and size of surface, size of hot sparks and their temperature, power of electric spark) is in inverse proportion to the 'activity' of the gas mixture. (The more active a mixture is, the less energy is required for its ignition.)

Therefore, when the concentration of oxygen changes in the gas–air mixture, the mixture is pre-heated, or the pressure is above atmospheric, ignition could occur with a lower ignition energy.

Notwithstanding the type of heat source, the time of contact of the hot ignition source with a relatively cold explosive mixture is a very important factor. In any case, the time of contact must be greater or equal to the time required for initiation of the chemical reaction of combustion which can be described by the empirical relationship:[1]

$$\sigma_p = \frac{20}{T_e - T_0} \times \frac{a}{\upsilon_s} \qquad (2)$$

where σ_p = reaction time in flame zone (seconds),
$T_e - T_0$ = temperature of combustion and initial temperature of gas,
υ_s = standard flame speed in the flammable mixture,
a = coefficient (temperature conductivity).

Ignition of a surrounding explosive atmosphere by the turbulent stream of exhaust gases discharged from an exhaust pipe or broken joint occurs as a result of intensive mixing of a cold explosive mixture without exhaust gases. A particular feature of turbulent stream ignition is the fact that at certain times and points within the turbulent stream the concentration and temperature of the mixture could be such that ignition could occur despite the average temperature and concentration of the mixture being much less than that required for ignition. Therefore, the temperature of exhaust gases should be kept well below the auto-ignition temperature of expected explosive mixtures.

The theory of ignition by a hot turbulent stream is not yet developed and, therefore, safe temperatures of exhaust gases can be determined only by experiment. Various codes of practice recommend that the temperature of exhaust gases should not exceed 0.6–0.8 of the auto-ignition temperature. Table I gives safe exhaust gas temperatures for a number of gases that are most likely to be present on offshore installations.

Ignition of flammable mixtures by hot surfaces has the same basic mechanism as auto-ignition, the only difference being that ignition by a hot surface is localised whereas auto-ignition occurs in the whole volume of gas. This is described by standard tests recommended by, for example, BS4056 or ASTM test D2155/66.

There is a functional relationship between ignition energy (proportional to activation energy) and the surface temperature necessary for ignition of a flammable gas which is in contact with a hot surface. One of the forms of such a relationship is

Table I: Ignition data of some gases

Gas	Ignition temperature (°C) T_{ign}	Safe exhaust gas temperature (°C) (taken as two-thirds of T_{ign})	T/Class (BS 4683 Pt 1 1972)
Natural gas	482–650	320	T1
Methane	595	396	T1
Propane	470	313	T1
Condensate vapours	275	183	T3
Average kerosene vapours	210	140	T3
Tri-ethylene glycol	370	246	T2

given below as an example of the relationships that exist between different properties of the gas, its ignition energy and the surface temperature necessary for ignition. The general relationship has been derived by Adomeit,[4] who studied ignition by hot wires and rods:

$$T_w = f\left(\frac{q_w}{\sqrt{\lambda\, Q\, (A_\varphi)}}\right) \times \frac{E^*}{R} \qquad (3)$$

where T_w = surface temperature,
R = molar gas constant,
Q = the molar reaction enthalpy,
q_w = the smallest local value of heat flux at ignition body in non-reactive flow,
A_φ = coefficient which represents the influence of the equivalence ratio upon the overall rate of heat generation,
E^* = activation energy of ignition reaction,
f = function.

As can be seen from the above relationship, E^*/R is a characteristic of the gas and the following:

$$f \; \frac{q_w}{\sqrt{\lambda\, Q\, (A_\varphi)}}$$

is a characteristic of the heat exchange between a hot surface and a gas in the surrounding atmosphere.

Ignition temperatures of flammable gases ignited by hot surfaces are significantly higher than corresponding auto-ignition temperatures. This can be explained by the fact that the condition of heat exchange and physical state of the gas in practice differs from that of standard test conditions. For comparison, auto-ignition temperatures and ignition temperatures of a hot surface acting as a source of ignition are given in Table II.

Conditions of free convection as in the experiments given in Table II, are not achievable in practice since the radiator fan of a running engine and wind will always create an air flow (explosive mixture flow) along hot surfaces. Therefore, the time of contact of the mixture with the surface will be shortened and the ignition temperature will be higher than that shown in Table II. In view of this, the surface temperature of the diesel engine parts could be safely taken as 0.9 or even as being equal to the auto-ignition temperature of the expected explosive mixture, which provides a safety factor 1.5 to 2.

The main feature of ignition by electric sparks is that relatively small spaces between electrodes produce very high energy. Therefore, between electrodes, due to the high level of ionisation of molecules, a combustion reaction starts practically instantaneously. However, propagation of the reaction from the zone of ignition to the bulk of the explosive mixture depends upon the state of the mixture and the power or energy of the spark.

In general, discharge of static electricity sparks could occur from pre-charged items isolated from the earth conductors, from dielectrics under the influence of high voltage, or from so-called 'sliding discharge' when two dielectric surfaces slide against each other (like some belt drives) or any other object. Ignition by impact, or contact sparks, occurs due to the generation of high temperatures at the points of contact, the surface of contact being small in relation to the force applied, especially if the surface is rough. The temperature at the point of contact could reach the melting point of the metal and, if in motion, small particles of melted metal may tear away and discharge as sparks. It was found that during the impact of two steel objects with a carbon content of up to 0.8%, sparks were produced with a temperature of not lower than 1350–1400°C. A particle of metal with such a temperature is undergoing the process of oxidisation with the generation of additional heat. The most dangerous impact sparks are those generated during the impact of aluminium alloys with a corroded steel surface and during the impact of two aluminium alloy parts with a surface oxidisation film of over 2–5 µm (0.002–0.005 mm) thickness. During frictional contact particles of iron and alu-

Table II: Comparison of auto-ignition and hot surface ignition temperatures

Gas–air mixture	Temperature °C (K)		Hot surface		
	Auto-ignition	Hot surface ignition	Material and shape	Diameter (mm)	Velocity of gas (m/s)
Hydrogen (H_2)	510(783)	880(1153)	Steel rod	6.3	60
Methane (CH_4)	610(883)	1470(1743)	Tungsten wire	0.025–0.5	Free convection
Methane (CH_4)	610(883)	1160(1433)	Steel net	–	Free convection
Natural gas (93.2% CH_4)	500(773)	1180(1453)	Steel rod	11.2	Free convection
Natural gas	500(773)	940(1213)	Steel rod	37	Free convection

minium tear away. Iron oxide and aluminium particles produce a thermal reaction with a large release of heat as previously mentioned.

Apart from hot exhaust gases and hot surface temperatures, ignition of flammable gases could occur in the induction system during the overlapping of valves or scavenge ports. This is more likely to happen when the engine is overheated. In this case the fuel–air mixture becomes preheated and ignition in the cylinder can occur earlier than the designed moment resulting in high explosive pressures (P_z) and temperatures than expected. Ignition timing will be disturbed and in this case the probability of blowback and failure of joints is increased. One would expect that in the induction system the velocity of the gas–air mixture would be much higher than the flame velocity and, if ignition occurs, flame could not propagate to the open end of the induction piping as it would be drawn in the direction of the cylinder head and cylinders. This could be true for a uniform laminar flow of the flammable mixture in an air intake. But in reality uniform laminar flow can never be achieved in a diesel engine induction system. Owing to the reciprocating motion of pistons and the presence of bends and obstacles in the piping, flow becomes turbulent. Different types of obstacles can serve as flame stabilisers where the velocity of the gas flow could be reduced to less than the flame velocity. In this case, flame stabilises near such obstacles and then propagates to the open end of the system or to the outside of the engine causing external ignition. Under certain conditions, ignition can occur in the exhaust system. However, the probability of ignition of a flammable mixture in the exhaust system is less than in the induction system. Flammable gas from the surrounding atmosphere cannot enter exhaust piping when the engine is running at normal speed since the pressure of the exhaust gases will always be higher than atmospheric. However, during starting and stopping of the engine, flammable gas from outside could be present in the exhaust system and be ignited.

Combustion and probable combustion régimes

Any combustion process is a chemical reaction between a flammable substance and oxygen. Generally the dynamic aspects of the reaction rather than its chemistry are considered here. For the sake of definition one form of combustion can be presented schematically, as in Fig 2.

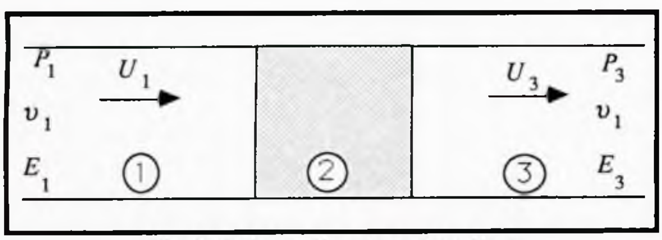


Fig 2: Schematic of combustion

Here, reaction zone 2 (flame) is considered to be standing still, and unburnt gas with initial parameters P_1, v_1, E_1 and velocity U_1 is streaming towards reaction zone 2, and burned gas or products of combustion with parameters P_3, v_3, E_3 and velocity U_3 is flowing away from the reaction zone. The symbols P; v; E are pressure, specific volume and specific energy respectively.

The laws of conservation of mass, momentum and energy can be applied for the initial and final states of the gas.

$$\rho_1 U_1 = \rho_3 U_3 \qquad \text{Conservation of mass}$$

$$U_1^2 \rho_1 + P_1 = U_3^2 \rho_3 + P_3 \qquad \text{Momentum}$$

$$H_1 + \frac{U_1^2}{2} + Q = H_3 + \frac{U_3^2}{2} \qquad \text{Energy} \qquad (4)$$

where ρ = density of the gas = $1/v$
$H = E + Pv$ (H = enthalpy of the gas)
Q = heat of reaction of a unit of gas

For any gas state, the equation of state can be written in the form:

$$P = \boldsymbol{R}T\rho$$

where $\boldsymbol{R}$ is the gas constant and T is the temperature.

Assuming that the combustion reaction always goes to completion and neglecting all forms of loss, the state of the gas in zone 3 ($P_3v_3E_3$) can be found. The solution of the above four equations is usually represented by the pressure–volume curve (Pv) for the burned gas. The curve is called the Hugoniot adiabatic or curve of combustion (Fig 3) and in the final form can be presented as:

$$\frac{\kappa}{\kappa - 1}(P_3v_3 - P_1v_1) - Q = \frac{1}{2}(P_3 - P_1)(v_1 - v_3) \qquad (5)$$

where κ is the ratio of specific heats.

The Hugoniot adiabatic (curve) was introduced as a general equation for a single compression of gas by shock waves. If

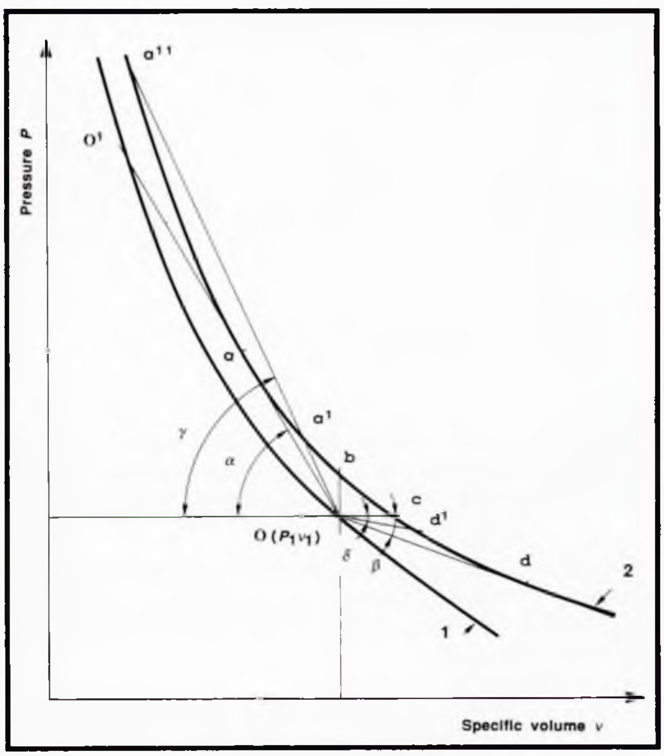


Fig 3: Hugoniot curves

Here, curve 1 = adiabatic for shock compression; curve 2 = combustion curve; O(P_1v_1) = initial parameters of the gas mixture; O' = final state of shock of compressed gas mixture. Symbols on combustion curve 2 are final states of the gas for different combustion régimes.

there is no chemical reaction of combustion the Hugoniot curve passes through the initial point with parameters of the gas P_1v_1. If there is combustion the curve lies above point O (curve 2). The curve represents the curves of final states P_3v_3 for any initial state P_1v_1 and the addition of heat.

The Hugoniot curve of combustion consists of three parts which in general represent three possible types of combustion régimes; detonation, deflagration and non-stationary régimes. The top part of the curve, above the point b, describes the state of detonation products and the branch below point c describes the final state of the products in deflagration régimes. Between points c and b lies imaginary states of non-stationary régimes. Two tangents Oa and Od can be drawn from the initial point O to the Hugoniot curve with points of tangency at a and d. Point a represents the final state of the gas with minimum possible detonation velocity $D = v\sqrt{tg\alpha}$ and point d represents the final state of gas with maximum velocity of deflagration. Hugoniot curves can be drawn for any given flammable gas using experimental data, and they are very convenient for analysis of the combustion properties of a particular gas. As an example, Hugoniot curves for methane are given in Fig 3(a).

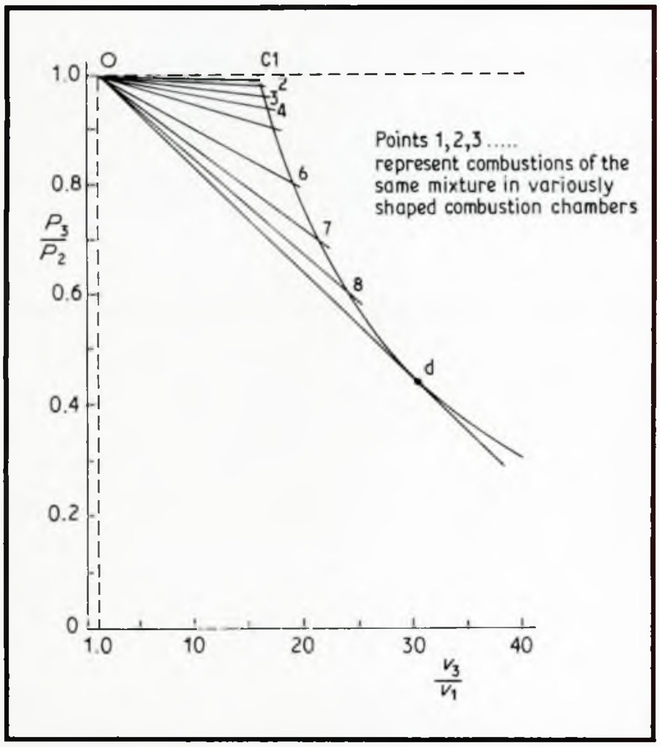


Fig 3a: Deflagration branch of the Hugoniot curve for a stoichiometric mixture of ethyl alcohol and oxygen

When considering a diesel engine as a possible source of ignition of flammable gases, it is convenient to consider the three above mentioned probable combustion régimes with subdivisions as follows.

1. Deflagration
   a. Combustion in an open space – ambient conditions
   b. Combustion in pipes and ducts
   c. Combustion in a closed volume
2. Non-stationary combustion régimes with accelerating flames
3. Detonation

Deflagration

Deflagration is a slow combustion described by the lower branch of the Hugoniot curve and usually takes place with a fall in pressure, $P < P_1$, in the reaction products as they expand. Each point on the lower branch of the curve represents the deflagration régime with a constant velocity defined by the slope of the straight line (Mikhelson–Rayleigh line) from the point O to the appropriate point on the Hugoniot curve. For example, the velocity of combustion products in state d^1 is defined by $v_{d1}\sqrt{tg\alpha}$ which is less that $v_d\sqrt{tg\beta}$ at state d (Fig 3). (The deflagration branch of the Hugoniot curve is shown in Fig 3a).

Slow combustion in open space at ambient conditions

This combustion régime takes place when there is no restriction on the expansion of exhaust products at atmospheric pressure and ambient temperature. Under this condition there is no significant change in pressure ($P \cong P_1$) and the flame velocity is very low. On the Hugoniot curve this mode of combustion is presented by the final state of the combustion products at point c. The slope of the secant Oc is very close to zero, therefore the speed of products is minimal and consequently flame speed is very low. Flame velocity achieved during this mode of combustion is very often called standard flame velocity. Some examples of flame velocities taken from various sources are given in Table III and for natural gas–air mixtures it does not exceed 40 cm/s.

The combustion propagates by conducting heat from burnt to unburnt gas and also by diffusion. Slow combustion in an open space with standard flame velocities relates to laminar flow of gas and has no practical significance as laminar flow in open space cannot be stable since the flow gradually becomes turbulent and the flame accelerates.

However, it should be noted that most types of combustion initially start as slow combustion with subsequent development into other combustion régimes. Therefore, it is useful to know the basic parameters of slow combustion when considering more violent processes in ducts and pipes.

Combustion in pipes and ducts

In ducts and pipes the combustion wave usually has a greater velocity than that given in Table III. The explosive mixture is confined to the pipes and flow induced by thermal expansion is restricted by the pipe walls. Due to heat losses through the walls and boundary layer friction a combustion wave has a divergent character of propagation. This is due to the fact that mass flow through the cross-section of the pipe is not uniform, as mass flow in the centre of the pipe is greater than at the boundary. Consequently the area of chemical reaction of combustion is not plain but curved and the flame front is stretched. Thus more unburned gas enters the reaction zone producing more heat and finally increasing the speed of the flame.

Final states of combustion during slow combustion in pipes are presented on the lower branch of the Hugoniot curve between points c and d with maximum speed of reaction and maximum velocity of the flame front as point d. Propagation of slow combustion depends on the thermal conductivity of the mixture (λ) and the rate of chemical reaction, which in turn depends upon the concentration of the reagents and the temperature. The equation for the Hugoniot curve of slow combustion can be expressed as:

Table III: Standard flame velocities of some gases

Gas–air mixture	Natural gas	Methane	Propane	Ethane	Ethylene
Stoichiometry of mixture (%)	10	9.8	4.02	4.7	6.51
Flame velocity (cm/s)	40	37	42	32	64

(see ref 5)

$$\frac{\kappa}{\kappa - 1} P v - Q = \frac{1}{2} v \,(P - P_0) \qquad (6)$$

where speed is determined by the slope of tangent at point d and is:

$$\frac{P_0 - P}{v} = \kappa \frac{P}{v} \qquad (7)$$

and then

$$P = \frac{P_0}{\kappa + 1} \qquad (8)$$

which is always less that P_0.

With assumptions of lossless combustion and the independence of λ (the thermal conductivity from temperature), Zeldovich[5] showed that if the flow in the pipe is laminar the flame velocity attains the maximum value and is dependent only upon the heat of reaction and maximum velocity of stationary slow combustion. This is proportional to $1/\sqrt{Q}$, where Q is the heat of reaction per g of mixture. In this case the flame velocity could be expressed by the final formula:

(ref 5)

$$V_c = \frac{C_0^2}{\sqrt{2Q\,(\kappa^2 - 1)}} \qquad (9)$$

where C_0 is speed of sound in a gaseous mixture and κ is the ratio of specific heats.

Calculations carried out by the author for a stoichiometric mixture of Bacton gas with air gives the value of V_c, maximum velocity of slow combustion, equal to 78.6 m/s. Initial parameters of the gas taken were: $T_0 = 298°$K, $P_0 = 1$ bar, $R_0 = 0.2996$ kJ/kg·°K (gas constant), and $\kappa_0 = 1.39$. Heat losses were not taken into account and actual speed would be less than calculated by this formula. The maximum expected flame temperature during slow combustion in pipes is 2200°K.[2,5]

As it was shown by Lewis and others, flame speed is increased with an increase of pipe diameter when all other conditions remain the same.

Combustion in a closed volume

This type of combustion can take place within the diesel engine in spaces such as induction and exhaust systems and also in crank cases when there are no provisions made for relieving the pressure of expanding combustion products. Combustion in a closed volume can be treated as slow combustion or deflagration and does not involve a dangerous combustion régime as such but, due to the restrictions on free expansion of combustion products, pressure in such confined spaces rapidly increases and explosion may finally take place, especially if the closed volume is large. This may happen if, eg, the shutdown device closes the air intake too late, when combustion of the explosive mixture in the induction system has already started. Ignition of flammable gas–air mixtures in the air intake is more likely to occur in the induction manifold from the closed end of the system. When there are no relief devices fitted, the system should be treated as a closed vessel and in which case maximum explosion pressure is expected.

Due to the fact that combustion in a closed volume starts as slow combustion, and closed volumes within the diesel engines are usually not large enough to allow significant acceleration of the flame front, the expected flame velocities are expected to be not higher than the maximum velocity of slow combustion. For a natural gas–air mixture it is 78.6 m/s, calculated without taking heat losses into consideration. Calculations of maximum explosion pressures and analysis of experimental data of the various references were carried out by the authors. The calculated maximum explosion pressure for a stoichiometric mixture of gas and air is $P_{exp} = 8.4$ bar and the maximum temperature is $T_{exp} = 2506.6°$K (assuming adiabatic explosion). Maximum pressures obtained from experiments with methane in steel bombs were 7.5–8.4 bar when the initial pressure was 1 bar.[3,6] From equations for explosive pressures:

$$P_{exp} = \frac{(K - 1)Q}{v_0} \qquad P_{exp} = P_1 \frac{T_e}{T_i} \frac{\sum M_e}{\sum M_i} \qquad (10)$$

it can be seen that the maximum explosive pressure (P_{exp}) very much depends upon the initial parameters of the gaseous mixture P_1, T_1, v_0, – pressure, temperature, density (specific volume). Here $\sum M_i$ and $\sum M_e$ are the number of moles present before and after reaction respectively. Q is the heat of reaction and κ the ratio of specific heats. It has been generally established that the explosive pressure is approximately half that of the detonation pressure for the same gas.

Non-stationary combustion régimes with accelerating flames

Slow combustion with constant speed could be possible as long as flow remains laminar, but if gas flow becomes turbulent the combustion process is prone to accelerate. Combustion produces stream turbulence which increases the surface of the combustion wave and therefore the amount of a gas burnt per unit time. The more gas burnt per unit time the more rapid the combustion will be, resulting in more intensive turbulence and so on, so that the combustion process becomes unsteady and self-accelerating. Turbulence originates at the walls in tubes and ducts and is carried into the flow from turbulating devices and is generated by the flame itself. Rough pipes, deposits in pipes, bends, obstacles and long pipes all increase turbulence and contribute to flame acceleration.

Generally, gas dynamics considers a gas flow in pipes with a Reynolds number of > 2300 as turbulent, and self-acceleration of flames could therefore occur in such pipes.

It was shown by Schelkin[6] that turbulence generated by the flame itself could increase laminar velocity by 10 to 100 times. This is due to the fact that turbulent velocity is equal to laminar velocity increased by the ratio of the turbulence-enlarged surface of the flame front to the area of the undisturbed surface. For example, a 90 deg bend with a bending radius-to-diameter ratio equal to 1, increases the surface-to-reaction ratio by a factor of 2. Therefore an increase in flame velocity could be expected of at least twice that of the initial laminar flow. In turbulent flow it could be even higher. Experiments carried out by Pyroban Limited[7] with propane–air mixtures in a pipe of 50 mm in diameter, 1.5 m long and three bends, showed flame speeds as high as 277 m/s.

It should be noted that flame-generated turbulence finally reaches a constant value in a pipe of given diameter and any additional increase of turbulence is normally due to the construction of the piping and the number and location of turbulence-generating items such as bends, valves, venturi, etc.

Non-stationary combustion régimes in the Hugoniot diagram (Fig 3) correspond to the line c–b. Flame speed in this region is above the maximum velocity of slow combustion but smaller than detonation velocity. However, if a flame is progressively accelerating it could reach detonation velocities or high velocities of non-detonative combustion which possess high destructive power just as detonation does.

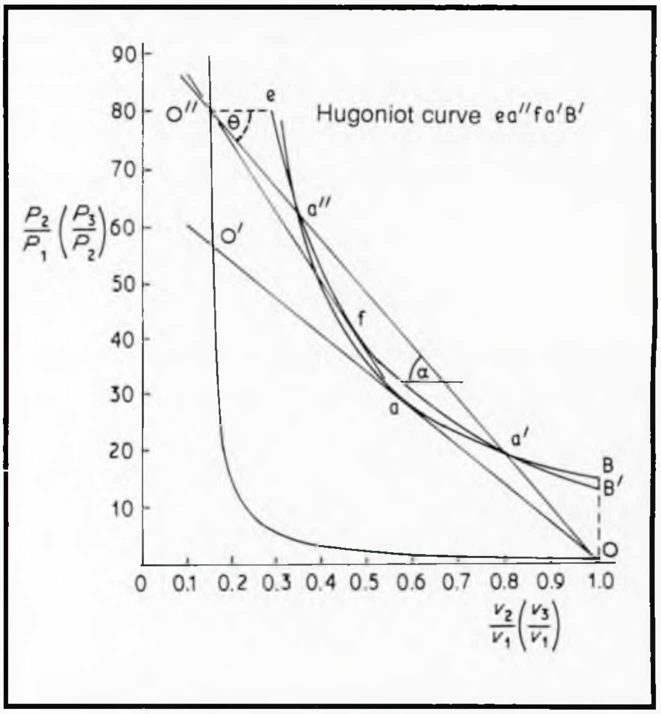


Fig 3b: The Hugoniot curve for the mixture $CH_4 + 2O_2$

Detonation

Detonation is a practically instantaneous process of combustion of flammable gases which is accompanied by very high flame velocities (2000–3000 m/s) and pressures and therefore has a high destructive power. Detonation usually develops in confined spaces such as ducts and pipes. In diesel engines operating in flammable atmospheres detonation could develop in the air intake, in exhaust systems, and in crank cases.

Detonation is initiated by shock waves or sometimes develops by transition from normal combustion with progressively accelerating flames. This is more likely to occur in long pipes. The phenomenon of combustion was studied by Zeldovich & Newman and according to their model fully developed detonation is the instantaneous combustion of flammable gas, compressed and heated by shock waves which are created by combustion itself or introduced by other sources.

Régimes of detonation on the Hugoniot curve are presented by their final state of detonation products as points above point b in Fig 3. The point of tangency a, is the final state of detonation with minimum detonation velocity; very often called Chapman–Jouquet detonation. Final detonation states which lie above point a represent so-called over-compressed detonation and below point a, a state of under-compressed detonation. During the compression by shock waves, parameters of the flammable gas change along the secant OO^1, with the final state of compression at point O^1 which lies on the adiabatic of compression without combustion (curve 1). Gas ignited by the high temperature of shock compression with following combustion and expansion of gases is along line O^1a. Final parameters of this detonation are attained at point a on combustion curve 2. Slopes of secants drawn from the point $O(P_1 v_1)$ to any point on the upper branch of the Hugoniot combustion curve determine the velocity of detonation. Above point a the velocity of over-compressed detonation is greater than the minimum velocity at point a and theoretically could reach infinity. So points a^1 and a^{11} represent under-compressed and over-compressed detonation with equal velocities since the slope of secant Oa^1a^{11} is the same. An example is given of the detonation branch of the Hugoniot curve for a methane–oxygen mixture in Fig 3(b).

In the case of diesel engines the initial shock wave could develop from pressure impulses in the induction system as the gas flow is not steady and always turbulent due to reciprocating movement of pistons and valves and the design of the system. During the overlapping of valves or scavenge ports, the blow-out flames could occur from the cylinder of the engine and initiate a shock wave. In the case of an ignition this shock wave, in a sufficiently long pipe, could develop into a strong shock wave with velocities of Mach 3 to 6 and detonation. Fully developed detonation occurs when the detonation velocity and the shock wave velocity are equal. According to Zeldovich[5], for full development of detonation, the pipe should be at least 15 diameters in length with developed turbulent flow.

Not every combustion process occurring in pipes develops into a detonation. If a flammable gas–air mixture is slow burning, detonation may not occur at all. There is no data readily available on the detonation of methane or natural gas in air. But Lewis[2] obtained detonation of a methane–oxygen–nitrogen mixture with the stoichiometry $CH_4 + 1.5O_2 + 2.5N_2$. A velocity of 1880 m/s was recorded.

Investigations carried out by Zeldovich[5], Schelkin[6] and others showed that sometimes slow burning explosive mixtures which cannot detonate under normal conditions can detonate if a small percentage (1.5–3%) of other gases is added to the mixture. The addition of other gases to flammable mixtures changes the kinetics of the chemical reaction and a normally undetonative mixture could detonate.

In an induction system, when a natural gas–air mixture is ingested, there may be several gases mixed together such as compounds of natural gas, oil and diesel fuel vapours, carbon monoxide, water and others. Most of these gases can detonate individually. The composition of the flammable mixture in the air intake can be even worse when a crank case breather is connected to the air intake, or a permanently installed easy-start system leaks. The chemical composition of such a mixture is unpredictable and could vary considerably depending upon its condition, time in service and design.

In view of this, the possibility of detonation in induction and exhaust systems cannot be ignored. In order to assess the magnitude of expected detonation pressure and flame velocities, calculations of possible theoretical detonation for a stoichiometric mixture of Bacton natural gas with air were carried out by the author. The calculations were based on thermodynamic and gas dynamic properties of the gas and the Zeldovich–Newman model of detonation. The results of the calculations are summarised in Table IV from which it can be seen that the mixture of natural gas and air chosen by the author cannot detonate due to the following reasons.

Table IV: Summary of calculated detonations

Calculated detonation parameters for a stoichiometric mixture of natural gas and air. [1 kmol of mixture taken as 0.0944 CH_4 + 0.0032 C_2H_6 + 0.0009 C_3H_8 + 0.0015 N_2 + 0.19 (O_2 + 3.76 N_2).]

Standard heat of reaction of 1 kmol of mixture	ΔH_R^o	82146.072 kJ
Heat of reaction at combustion temperature	ΔH_R^T	84822.122 kJ
Heat lost on dissociation	E_d	2396 kJ
Heat lost on heating of nitrogen	Q_{N_2}	33991.3 kJ
Effective heat	Q_{ef}	48434.3 kJ
Ratio of specific heats at combustion temperature	κ	1.29
Ratio of specific heats of unburnt gas	γ	1.39
Lossless detonation velocity	D_0	1522.2 m/s
Detonation heat and mechanical losses	ΔD	60.88 m/s
Maximum pressure behind the shockwave in front of the detonation:		
with loss	P_2^1	20.59 bar
without loss	P_2	22.35 bar
Temperature behind the shockwave:		
with loss	T_2^1	1275°K
without loss	T_2	1376.6°K
Reaction time in pipe:		
id = 100 mm	t	8.2113 x 10^{-5} s
id = 150 mm	t	1.23169 x 10^{-4} s
Minimum length (and diameter) of pipe needed for fully developed detonation in pre-turbulated flow:		
d = 50 mm	L	0.75 m
d = 100 mm	L	1.5 m

To initiate and propagate detonation, the speed of the shock wave should be high enough to get the temperature T_2 behind the shock wave higher than the shock ignition temperature of the mixture $T_{ign\ sh}$. If the shock wave velocity is small, the temperature behind the shock wave cannot reach the ignition temperature of the gas mixture. Therefore the combustion cannot be instantaneous like it is in detonation, but will continue by conducting the heat liberated in the reaction to the unburnt gas and by diffusion, as occurs in deflagration. It should be noted, however, that shock ignition temperatures are usually higher than spontaneous ignition temperatures (T_{ign}) as the induction period for initiation of the chemical reaction in shock waves is very short. For most mixtures the difference between $T_{ign\ sh}$ and T_{ign} is not very high but for methane–oxygen mixtures the shock ignition temperature was found to be unusually high (1087°C)[3] (shock tube experiments). It is obvious that this temperature is much higher than the spontaneous ignition temperature of the same mixture (556°C). If it is assumed that shock ignition temperatures of natural gas–air mixtures are about the same order of magnitude as for methane–oxygen mixtures, then the calculated temperature (Table IV) behind the shock wave T_2^1 is not high enough for ignition of the mixture.

But in reality, detonation is more complicated. A shock wave could reflect from walls or obstacles thus forming rarefaction and oblique shock waves which, under certain conditions, could increase the local compression ratio of the gas and ignite it. However, even under conditions of successful ignition, detonation does not necessarily occur as the transition to detonation depends upon the pipe diameter and the size of the detonation cell (size of transverse shock wave spacing). For comparison the detonation cell sizes 'Z' of some gases are as follows:

$$
\begin{aligned}
CH_4\text{–}O_2 &= 4.3\ \text{mm}\\
CH_4\text{–Air} &= 300 \pm 20\ \text{mm}\\
H_2\text{–}O_2 &= 1.5\ \text{mm}\\
H_2\text{–Air} &= 15\ \text{mm}\\
C_2H_4\text{–}O_2 &= 0.2\ \text{mm}\\
C_2H_4\text{–Air} &= 5.9\text{–}9\ \text{mm}
\end{aligned}
$$

There exists a critical pipe diameter for the failure of detonation and it was found[8] that this is half the size of the cell. Propagation of the detonation and its existence is possible if the pipe diameter is larger than the critical diameter equal to 0.5Z.

It can be expected that a real gas–air mixture in the air intake will be different from that given in Table III and this could increase the probability of detonation. Even a small increase in initial pressure and temperature, as it is in supercharged engines, significantly increases pressure and temperature of the shock wave and detonation could be possible. This should be taken into consideration for the design of protective systems for diesel engines. This is especially important when it is expected that the diesel engine will be located in an area where several types of gases may be present.

Mechanism of flame quenching

Flame fronts in tubes are not plain, but curved and stretched. The extent of curvature is determined by the Karlovits number or stretch factor K. There exists a critical divergence (characterised by K) of a flame front at which the balance between production of heat and loss of heat is not established, ie more heat is being lost than produced and therefore the flame is quenched by the unburnt gas only, even without external heat sinks.

Critical divergence can be obtained if the flame is confined between two parallel plates, the distance between being h and called the quenching distance, or in a pipe of diameter d, called the quenching diameter. Quenching distance and quenching diameter are minimum dimensions in which flame can still propagate, but are quenched if these dimensions decrease. These critical parameters of flames are used in the construction of flame arresters. Every gaseous mixture has its own critical parameters which means that flames of the mixture can be effectively quenched only by a flame arrester which is especially designed to quench flames, the critical parameters of which are the same as the mixture. Quenching diameter is very often called the minimum flame diameter and it is considered that gas cannot be ignited if it is confined to a volume less than that of the quenching diameter at ambient conditions. This principle is used in the design of flame arresters. The main purpose of flame arresters is to divide the flame front into a number of elementary flames of a size less than the quenching diameter (distance). These elementary flames are extinguished by cool, unburnt gas and additionally, the flame arrester aperture provides an external heat sink.

Quenching distance/diameter depends on the speed and pressure of the flame front. It was found that quenching distance is approximately inversely proportional to flame velocity and pressure.

With an increase of flame velocity and pressure, as in the case of explosion or detonation, quenching distance and diameter decreases. From experiments it was found that quenching distance and diameter should be taken as 50% of that determined at ambient conditions for the design of flame arresters to quench explosion flames.[9]

Quenching distances and the diameters for some gaseous mixtures at ambient conditions are given in Table V using experimental data from various sources, which are in good agreement with each other.

Attempts to compute quenching distances and diameters theoretically lead to very lengthy and unreliable calculations due to a number of simplifications and assumptions accepted in combustion formulae. It is more practical, simple and reliable to determine a quantity, such as quenching distance, by experiment. For approximate calculations concerning flame arresters, empirical formulae[9] can be used in addition to a subsequent prototype of arrester.

Quenching distance and diameter should not be confused with the concept of minimum experimental safe gap (MESG) (Table V) which is normally used for the design of explosion-proof enclosures for electrical apparatus. MESG is approximately equal to half the quenching distance.

Probability of ignition and explosion caused by diesel engines

The probability of ignition and explosion of a gaseous mixture surrounding a diesel engine is a combination of two probabilities. First, the probability of the appearance and existence of an explosive mixture in close proximity to the diesel engine and secondly, the probability that the diesel engine at this time will provide a source of ignition.

The probability of occurrence of a flammable atmosphere obeys theoretical laws. Such probable events as gas or vapours leaking and mixing with the surrounding diesel engine atmosphere in explosive concentrations from process vessels, relief valves, various joints in pipelines, wireline lubricators or bottom hole plugs etc, are events of an occasional nature which may be assumed to be stationary and ordinary events falling within the Poisson's law of distribution. The probability of such occasional events, ie the probability of occurrence of an explosive atmosphere, can be found by:

$$P = 1 - e^{-\lambda\tau} \tag{11}$$

where: P = probability;
λ = average number of events per unit time;
τ = time between two similar operations (for example, period between inspection of process vessel, maintenance of relief valve or workover/wireline operations). This can be taken as 1 year.

Research carried out by Shevchenko *et al*[1], and the author's offshore experience, have shown that the probability of a gas/vapour leak and creation of an explosive mixture lies between 1 and 10^{-5} and in relation to hazardous areas is as follows:

Zone 0 $P = 1$
Zone 1 $P = 10^{-2}$
Zone 2 $P = 10^{-4}$ to 10^{-6}

Areas can be considered safe when P is less than 10^{-6}.

There are no data on calculated probabilities of ignition by diesel engines and the author's attempts to calculate this proved to be unsuccessful due to the very limited number of known cases of ignition and explosion caused by diesel engines. However, a few of the known cases ended with devastating effects which cannot be ignored. It should be assumed that if the engine is not protected at all and produces all sources of ignition as described in The Nature of the Hazards section, the probability that such an engine would ignite an explosive mixture is 1. Moreover, even if one of the sources continually produces a hazard, as for example when the engine has an exhaust gas temperature above the auto-ignition temperature of the gas involved, the probability of ignition should also be taken as 1.

The overall probability of ignition of an explosive mixture by an unprotected diesel engine at the present stage of research can be taken as a probability of the presence of an explosive mixture as given above assuming that the unprotected diesel engine is a positive source of ignition.

One could say that probabilities of such magnitude are insignificant, but when compared with the cost of protection of a diesel engine, and the cost of a gas-producing platform, it becomes clear that even with such small calculated probabilities, the risk of loss of life and the loss of a multi-million pound platform is too great.

SOME DESIGN PROPOSALS FOR PROTECTION SYSTEMS

General

On the basis of the above theoretical consideration and analysis of experimental data, combined with practical experience gained during surveying and testing of diesel engines for use offshore, it is possible to formulate the main principles for design of protected engines and their protective systems.

To minimise the risk associated with the use of diesel engines in potentially flammable atmospheres, the engines and any part of their protective equipment should be designed for the most unfavourable condition. This condition should be based on the following assumptions.

1. A flammable gas–air mixture in the most dangerous concentration can be present in an area of engine operation.

Table V: Critical parameters of some flames at ambient conditions

Gas mixtures		Standard flame velocity (cm/s)	Karlovits number, K	Quench distance, h (cm)	Quench diameter, d (cm)
9.5% CH_4–air	Methane	35–37	0.6	0.216	0.31
40% $CH_4 + O_2$	Methane	305	0.4	0.05	0.07
5% C_3H_8 + air	Propane	40	–	0.183	0.260
4% C_3H_8 + air	Propane	45	–	0.128	0.179

2. The engine can provide a source of ignition as described in The Nature of Hazards section.
3. Ignition, with development to the most dangerous combustion processes, such as detonation and explosion, could occur within the engine.

As previously mentioned it is not possible to achieve absolute protection. However, an acceptable level of protection must be achieved. Acceptability of the level of protection could be determined by accurate data on the physical and chemical state of the expected gas and knowledge of the expected condition of operation of the diesel engine. For example, if the engine is located in a region where no gases other than natural gas are present, the surface temperature can be taken to be as high as 450–500°C. But if condensate vapours are expected in the area then the surface temperature should not exceed 183°C (see Table I). The desired level of protection will determine the size and number of protective devices.

Induction and exhaust systems

Induction and exhaust systems should be designed to withstand the maximum expected pressure and temperature. Maximum pressures are detonation pressures (20.5 bar), but they need not be taken as design pressures because detonation itself can be prevented by correct construction of the systems. By making the air intake as short as possible, with a minimum number of bends and unnecessary obstacles, it is possible to avoid significant flame acceleration, development of strong shock waves and detonation. By avoiding the practice of leading crank case breathers and cold piping into the air intake, the possibilities of detonation are reduced.

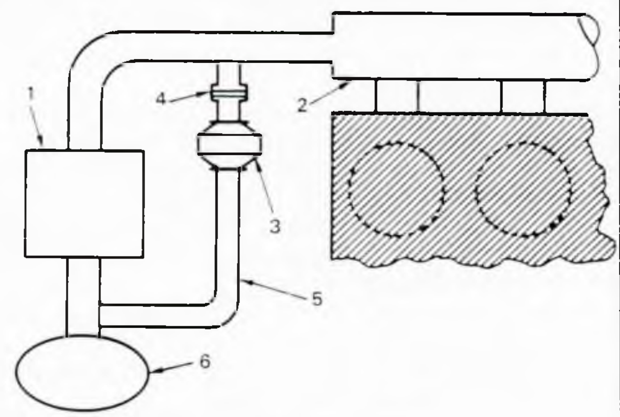
Detonation can thus be avoided, but explosion could occur in any system. Therefore the design pressure for the construction of induction and exhaust systems should be taken as the maximum possible explosion pressure (8.4 bar). The induction system from the cylinder head to the first shutdown device, including the flame trap, should be designed to withstand this pressure. All other piping should be designed for at least 2 bar, in accordance with a recognised code (eg ANSI B31.3).

This was confirmed during experiments carried out by Pyroban Limited and others when it was found that in short (1.5 m) open systems pressures even for fast burning mixtures did not exceed 1.84 bar. A design pressure of 8.4 bar for the inlet and exhaust manifolds and parts of the piping leading to the first shutdown device, corresponds well with many national regulations for diesel engine operation in underground coal mines as shown in Table VI.

In order to provide a safety factor on assumptions during calculations it is reasonable to take a design pressure of 10 bar. As mentioned before, in supercharged engines explosion pressures will be higher than 10 bar. Moreover, an induction system with a supercharger should be treated as a closed vessel between the engine and the supercharger. To predict pressure in such a case is difficult and therefore provision should be made to release explosion pressure by bypassing the supercharger as shown in Fig 4.

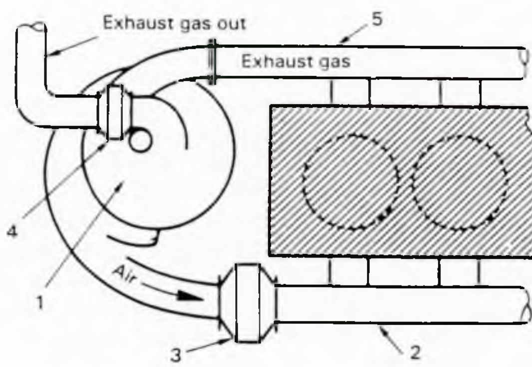
All connections in induction and exhaust systems should be flanged or screwed. Existing regulations require a minimum flame path in any connection of not less than 13 mm with a probability of having a gap between the flanges. This requirement has an obvious connection with requirements for flameproof electrical apparatus where flange connections are very often used without any gaskets at all. Such requirements cannot be automatically applied to diesel engine air intakes or exhausts (as it is at present) for the following reasons.

1. Diesel engine induction and exhaust system connections are normally designed with gaskets.
2. In the case of gasket failure the recommended 13 mm flame path could be efficient only in the case when the height of the gap would not exceed the quenching distance (1.28 mm for ambient conditions and at least 0.64 mm in case of explosion). The height of the gap is impossible to predict universally, as this depends upon the initial thickness of the gasket, number and tightness of fixing bolts and regularity of maintenance. However, if the



1. Rotor blade blower 2. Air intake manifold
3. Flame arrester 4. Relief valve with relief in bypass pipe
5. Bypass pipe 6. Air filter

Flame arrester fitted in bypass pipe



1. Turbocompressor 2. Air intake manifold
3. and 4. Flame arresters 5. Exhaust gas manifold

Flame arresters in turbocharged engines

Fig 4: Location of flame arresters in supercharged engines

Table VI: Minimum design pressure for engine induction and exhaust systems

Country . . .	*UK*	*USA*	*USSR*	*W Germany*	*Australia*	*Canada*
Design pressure (bar)	10	8.6	8	8/10	8.75	7.3

system is designed to withstand 10 bar and regularly serviced and maintained, the probability of flame being blown out through a flange connection could be significantly reduced.

Any penetration through the pressure wall of the system, such as spindles and stocks, should satisfy the same requirements as for flameproof enclosures as recommended by IEC Publication No 79 or BS5375. Gaskets used in the flange connection should be metal clad or of other suitable material in order to withstand high temperatures and pressures during the combustion of flammable mixtures.

No insulating material should be used on the outside surface of the system as, being permeable, it would allow flammable mixtures to make contact with the surface of the piping which, under insulation, would be even hotter.

Flame arresters

As ignition and flames are expected in induction and exhaust systems, provision should be made to quench these flames. Flame arresters serve this purpose by employing the mechanism of quenching described above. There are many different types of flame arresters, but for the protection of diesel engine induction and exhaust systems, the most suitable are crimped metal and parallel plate arresters. These types of flame arrester have low resistance to gas flow and are able to withstand explosion pressures. They are compact and their cost is relatively low. A flame arrester should be designed for a pressure of 10 bar, and a temperature of 2508°K, with an aperture size of 50% of the quenching diameter (distance)[9] of the expected flammable gas (or of the fastest burning gas in a mixture of different gases). In the case of natural gas it should be the quenching distance of propane. Subsequently the prototype flame arrester should be subjected to explosion pressure tests as being part of the pressure envelope. The induction manifold should then be subjected to explosion pressure tests at 1.5 times that of the expected or determined explosive pressure (taken as design pressure), as it is recommended in the IEC Publication No 79-1.

The anticipated flame speed determines the design width of the flame arrester. The effectiveness of the arrester increases with the increase in the width. Flame velocity during explosion depends upon the size of piping and should be determined experimentally by prototype tests. Calculations carried out by the author showed that a crimped metal flame arrester with a crimp height, h, = 0.6 mm, a diameter of 120 mm and a thickness of 38 mm, fitted in a pipe of 100 mm o/d and 1.5 m long, with two 90 deg bends (expected V_e = 256.8 m/s) is capable of quenching explosion flames of natural gas–air mixtures with a velocity of 513 m/s.

Flame arresters are usually designed with minimum resistance to gas flow and therefore relatively high pressure can be expected after a flame arrester in the event of an explosion. This is the reason for having air intakes and exhausts between flame arresters and the open ends of the systems made of robust construction, capable of withstanding 2 bar pressure as mentioned previously. An efficient flame arrester is one which is designed not just for a particular gas, but also for a particular air intake or exhaust system bearing in mind considerations of turbulence, possible flame acceleration, and the arrester location. The flame arrester should be located as close as possible to the expected source of ignition. Finally, each type of flame arrester should be explosion tested by a method as, for example, that recommended by Canadian Electrical Code and supplied with a certificate of testing and instructions for maintenance.

Spark arresters

The main purpose of a spark arrester is to prevent discharge of any sparks from engine exhausts to the atmosphere. It should be designed to satisfy engine manufacturers' recommendations for maximum back pressure in engine exhausts, it should be of robust construction and capable of withstanding pressures of at least 2 bar. Materials for spark arresters should be thermally suitable, that is, should not deteriorate or corrode at high temperatures. The effectiveness of spark arresters should be tested in simulated conditions and they should be supplied with a test certificate and particulars of construction and maintenance. The cyclone-type spark arrester is considered to be the most suitable for diesel engines offshore. Spark arresters should have clear identification and must not be confused with silencers. However, a combination of flame arrester and silencer is possible.

Shutdown devices and alarms

Protected diesel engines are expected to be equipped with shutdown devices and alarms activated by an overspeed sensor, a lubricating oil pressure failure sensor, high cooling water temperature sensors or high surface temperature sensors. When a number of shutdown devices are fitted in the air intake, they must be able to give instant relief of pressures, followed by non-return action in the direction of the open end, and gradual gas path deflection. The number of shutdown devices or alarms could be different for unattended and usually attended engines.

Overspeed shutdown devices

Any overspeed shutdown sensor should activate a shutdown device in the air intake (not shut down the fuel system). There are two main types of overspeed shut down devices activated by differential pressure in the air intake, or by a centrifugal speed sensor. From the point of view of reliability, the second type is preferable, as once it is set to a certain speed it does not need readjustment with time. The differential pressure valve is very reliable when new and well-maintained but in the author's experience it fails to operate properly in nine cases out of ten when tested on engines which have been in service for some time offshore. A differential pressure valve needs readjustment with time and the construction of the valve did not, in the past, make this easy. At present, there are differential pressure valves of an improved design on the industrial market which can be adjusted without dismantling the induction system. The main advantage of differential pressure valves is their mechanical independence from the engine.

Other shutdown devices

High water temperature, and oil pressure failure sensors, could activate the fuel rack of the engine in the ordinary way. This is considered to be sufficient and there is no need to fit the shutdown device for these conditions into the air intake. If the engine is normally attended a visual or audible alarm on loss of oil pressure or high water temperature is sufficient. For some particular applications of diesel engines it is useful to have a remote, manually operated, emergency shutdown device in the air intake, such as the 'Barber' Rig Saver. This is especially applicable for drilling engines. Remote and local controls of the engine should be installed so that the driller or engine operator can immediately shut down the engine during a blowout or drill stem test. It is considered good engineering practice when temporary duty diesel engines have provision for the shutdown device to be connected to the emergency shutdown system of the platform (ESD). In this case diesel engines can be stopped together with the rest of the firing equipment on the platform, by pressing the ESD button in the control room or other designated location. Additionally, diesel engines should have a shutdown device activated by a high exhaust surface temperature sensor or by high exhaust gas temperatures. Sometimes inert gas injection is used as an alternative to overspeed shutdown valves. This should operate automatically, activated by, for instance, a centrifugal overspeed sensor. When several shutdown devices are fitted in the air intake, the hydraulic characteristics of the induction system should be taken into consideration in order to avoid additional turbulence.

Cooling of exhaust gases

There are many proposals for the cooling of exhaust gases, but it seems that exhaust gas–water heat exchanges are the most effective and reliable as they reduce exhaust gas and exhaust manifold surface temperature at the same time and are quite simple to construct. The heat exchanger should be designed to be capable of reducing the exhaust gas temperatures to two-thirds of the auto-ignition temperature of gases expected to be involved. An example of a simple heat exchanger is the water-cooled manifold. An exhaust gas heat exchanger is part of an exhaust system at a point where the possibility of an explosion exists. Therefore it should be designed to withstand a pressure of 10 bar. When an engine is modified for use offshore, the installation of a heat exchanger should satisfy engine manufacturers' recommendations for back pressure, and usually, heavy duty radiators are necessary. It is an advantage to have the engine protection designed for a known expected flammable mixture as in this case the heat exchanger may be smaller and cheaper.

Interesting proposals have been put forward for the use of air cooled engines in hazardous areas by Weatherford Limited. Experiments carried out by this company and the author on a small (40 kW) air-cooled engine showed that cooling of the engine and exhaust gases to well below two-thirds of auto-ignition temperature can be achieved. Thermographs (Plate 3, Appendix A), show that the hottest surface temperature of the engine parts did not exceed 200°C, and the maximum exhaust gas temperature was 250°C at full load driving a variable displacement hydraulic pump. However, this method of cooling exhaust gases and engine parts requires additional research with regard to the reliability of maintaining steady and sufficient air flow from the cooling fan as it depends on the speed of the engine, condition of the drive belt, and other factors. Speed can be held constant with variable displacement pumps. However, even on the basis of data already obtained from the above experiments, it is possible to conclude that for small engines (up to 50 kW), air cooling can be accepted provided that the air flow reliability can be proved or the engine will be adequately protected from overheating if sudden air flow failure occurs.

In the case of the Weatherford engine a simulated fan belt failure test was carried out (the belt was removed and the engine was run for 20 min at full load on pump), and the temperature of exhaust gases was only 290°C maximum. This is mainly due to the overall derating of the engine by 50% which was possible in this case when the engine drove a variable displacement hydraulic pump.

Considerations for crank case design

Crank cases, with their vaporised lubricating oil–air mixtures, have always been considered as a potential cause of hazard – such a mixture in a particular range of concentration can explode and detonate. All that is needed to initiate such an explosion is a hot spot such as that which occurs in the event of bearing failure. Crank cases should be designed to withstand the maximum pressure during an explosion and be treated as flameproof enclosures. Small engines normally have crank case breathers which could act as pressure-relieving devices as well. Engines with a crank case volume of 0.5 m^3 and above should have suitable relief valves. Crank case breathers and relief valves should be fitted with individual flame arresters designed to quench oil mist–air flames.

Flame arresters in engines for use in a hazardous zone 2 area should not only quench explosive flames of oil mist–air mixtures, but also cool the gases after the flame arrester to a temperature below the auto-ignition temperature of expected explosive gas–air mixtures outside the crank case.

When an engine is equipped with an oil filler pipe and dipstick, their covers should satisfy the requirements for explosion-proof enclosures, ie they must be of screw type with a minimum of five threads engaged when fully closed.

Crank case breathers should not be connected to air intakes for reasons described above. The combination of flame arrester and non-return relief valve could be very effective as together they would constantly relieve excessive pressure, provide cooling of gases discharging from the engines and prevent air or gas–air mixtures from entering the crank case.

Some remarks on the protection of supercharged engines

The induction system of a supercharged engine has an air blower which could be a mechanical rotor blade type or a radial air compressor driven by an exhaust gas turbine. There are a number of problems with the protection of the induction system of such engines and in the positioning of protective devices. Due to the existence of high velocity turbulent gas–air flow with pressures above atmospheric, the expected flame velocity and final explosive pressure could be much higher than in an ordinary engine. The risk of explosion in a supercharged engine is greater, as a supercharger has very small clearances between rotors and casing, and a high flow resistance in the direction from the engine to the open end, thus providing practically a closed system between engine and blower. It is possible to reduce the risk of explosion in this part of the system by bypassing the blower and releasing high explosive pressures through a relief valve and relief system connected to the upstream side of the blower. The relief valve should be set to 1.5 times the maximum operational pressure of the blower and should be combined with a flame arrester. A rotor blade blower with radial clearances of less than half the quenching distance could be treated as a flame trap. A radial turbocompressor cannot be treated in the same way, because flames could propagate through the compressor due to the existence of passages of larger dimensions than the quenching distance. Moreover between the exhaust gas turbine and compressor a direct flame path can exist in the case of inefficient or worn seals. This could increase the probability of ignition in the air intake. In view of the above, it seems reasonable to locate a flame arrester as described in Table VII.

The location of flame arresters in supercharged engines, as shown in Table VII, are based on the assumption of the more probable cases of ignition. In all other respects, protection of supercharged engines is the same as for engines without supercharging, but it should be noted that the risk of using a turbocharged engine in hazardous areas is, in general, greater than for an engine without turbocharging.

Table VII: Location of flame arresters in a supercharged engine

Induction system	Rotor blade type scavenge blower	In bypass pipe together with relief valve
	Turbocompressor	Between the engine and blower
Exhaust system	Exhaust gas turbine	Between gas turbine and open end of the exhaust pipe, as close as possible to the turbine

Protection from other sources of hazard

Electrical equipment, being potentially hazardous in itself, should not be used with diesel engines. Starting arrangements and controls of the engine should not be electrical. Pneumatic, hydraulic or manual arrangements are safer for these purposes.

Reduction of the risk of ignition from static electricity is usually possible by using anti-static driving belts and special plastics for radiator fans incorporating some degree of conductivity. The risk of incendive sparking could be eliminated by avoiding the use of aluminium alloy parts within the engine. However, parts which are not in contact with any moving parts, and are protected from mechanical damage (eg oil filters), could be made from light metal alloy provided they are coated with a suitable epoxy paint.

To avoid any hazards from cold starting arrangements (such as an ethyl ether cold start system) they should be fitted so as to prevent any leakage into the engine air intake during normal operation of the engine. It is preferable to have metal piping with a self-closing valve connected to the air intake, but the liquid reservoir should not be permanently connected to the system as cold start is not used very frequently. The reservoir should be kept in a safe place and connected to the cold-starting system only when it is necessary to use it.

The engine and associated piping must, as far as practicable, be free from vibration, as vibration may be the cause of loose joints in piping and frictional contact between engine parts. To reduce vibration to an acceptable level the engine should be well balanced and installed on resilient mounts. The piping system supports should also be designed to absorb vibration.

Gas detection systems as a means of protection

A dedicated gas detection system can be fitted to a diesel engine with the purpose of initiating an alarm or shutdown of the engine depending upon the attended or normally not attended mode of operation. However, gas detection systems should be considered only as a secondary additional protection and should never be treated as a substitute for the protective arrangements as described above. The recent accident on Piper Alpha platform showed that explosion followed practically immediately after gas alarm indication in the control room. Time between sensing the gas (the lower explosion level; LEL) and the development of explosive concentrations can be very short and in the presence of an ignition source this inevitably will lead to explosion. The reliability and effectiveness of gas detection systems depends upon the number and correct position of gas sensors in the engine proximity or on the unit itself, the condition of air circulation in the engine proximity, the correct setting for a particular gas, and other factors. In the case of transportable engines the location of gas sensors for the purpose of protection of diesel engines could be very difficult. The following Figures show some possible types of protection.

Case A. Protection with the gas detection systems only (Fig 5).

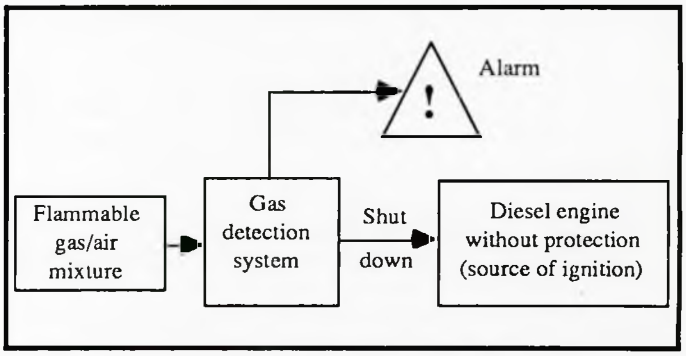


Fig 5: Protection by gas detection system only

In this case the logic system consists of three elements. Failure of the middle link, the gas detection system, immediately provides conditions for explosion since a flammable gas–air mixture can come into contact with an unprotected diesel engine. Even with an efficient gas detection system a flammable mixture could come into contact with a hot surface or engine sparks at the same time as the gas sensor detects gas. Also it should be noted that if the engine has a high surface temperature then this surface will remain hot for some time even when the engine has been stopped on LEL gas alarm. This will be the source of ignition if gas concentrations reach explosive limits. In any case, the probability of explosion for Scheme A (Fig 5) is very high.

Case B. Diesel engine fully protected and providing no source of ignition (Fig 6).

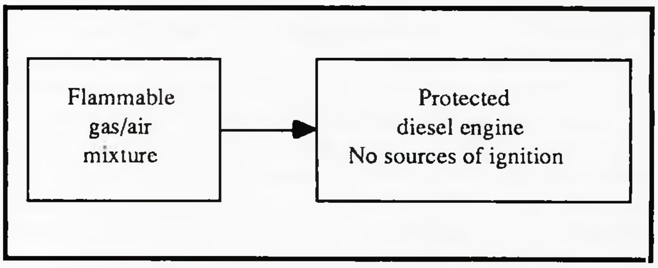


Fig 6: Fully protected diesel engine

In case B the probability of explosion is very small since even if a flammable mixture reached the protected engine, the engine will not provide a source of ignition.

Case C. Protected diesel engine with a gas detection system as a secondary means of protection (see Fig 7).

An explosive mixture can reach the protected engine and gas detector at the same time. Due to the engine being protected there is no immediate danger of explosion, but in order to eliminate prolonged contact the gas detection system initiates an alarm and the engine is shut down via the platform ESD.

Case A is considered the most unreliable of the three and this arrangement should be avoided. Case B is acceptable and Case C is the most reliable of the three. It should be noted that diesel-driven units, such as well kill pumps, drilling engines and

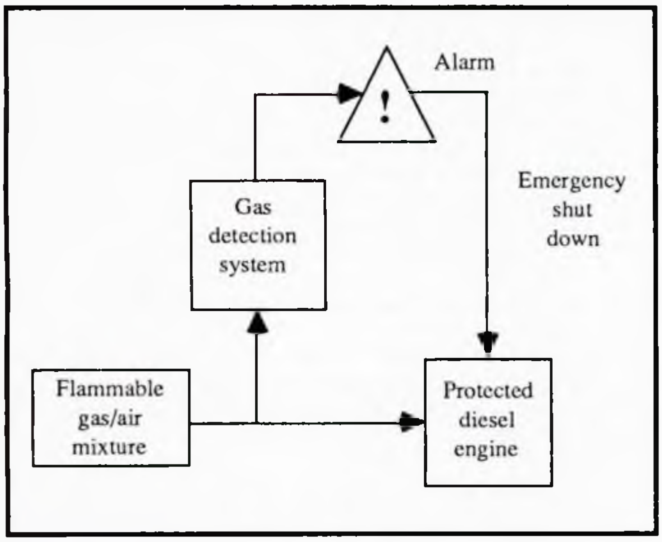


Fig 7: Fully protected diesel engine with secondary gas detection system

others, which could operate in a condition of well control, should be protected as in Case C (Fig 7).

PROPOSED INSPECTION, TESTING AND MARKING

Every diesel engine intended for use in hazardous areas must be subjected to a thorough initial examination and tests, and then regular inspections should be carried out at established intervals. Prototype tests of the engine and its protective devices should be carried out during manufacture, which should include the following tests.

1. Explosion test of exhaust and induction systems (fully assembled).
2. Vibration characteristics measurement.
3. Measurement of surface and exhaust gas temperatures.
4. Functional test of all protective devices.

Regular inspection and testing of engine safety arrangements, together with good regular maintenance, are essential factors for assurance of safe operation of diesel engines in potentially flammable atmospheres. The regularity of inspection and testing of safety arrangements is determined not just by running hours, but also by the effect of the marine environment, where safety devices may corrode and deteriorate even when the engine is not in use. Therefore inspection should be on a yearly or half-yearly basis and should be determined for each engine individually in view of its design or modification and use. The regularity of inspection of transportable temporary units should be monitored by an independent inspection authority, such as the Certifying Authority or relevant national authority.

Every new and modified engine should be examined and tested initially and an initial certificate of inspection should be provided. Details of safety devices associated with the equipment must be recorded on the certificate, together with a statement regarding the suitability for use in a particular flammable atmosphere. An applicant for such a certificate should provide all relevant certification for equipment and devices used with the engine.

Periodic inspection should be carried out by an independent authority and include a thorough examination of the engine, piping systems and all safety arrangements, which should be in good order and correctly fitted. During this inspection the engine should be subjected to the following tests.

1. Overspeed shutdown device test.
2. Measurement of surface temperatures under full load. (Normally the engine needs to be running for about 15 min on full load before measurements are taken.)
3. Tests of oil pressure failure, high water temperature, and other shutdown devices or alarm systems.

The inspector must check initial arrangements, test safety devices and determine the condition of the safety equipment such as spark and flame arresters. With satisfactory results of this inspection an endorsement on the initial certificate could be made for an extension of its validity for another established period. If a certified engine was subjected to further modification it should be re-tested according to the relevant sections of the initial test programme. An additional inspection and test by the owner or operator of the engine should be carried out after any transportation of the engine to ensure that no damage has occurred during transport. Suitable records of such inspections and maintenance must be kept.

Marking of a protected engine is very important as it gives an indication to any person of the protective arrangements and approved conditions of operation. A marking plate attached to the engine should include for example the following information.

1. Engine serial number
2. Flame arrester serial number and size of aperture
3. Spark arrester serial number and type
4. Suitability for temperature class (T1 – natural gas)
5. Date of initial test
6. Prototype test certificate number and authority stamp

With acceptance of the proposed system, it would be possible to establish continuous control of safety for diesel units used offshore and to withdraw engines which are not safe and not suitable for the intended purpose of operation in hazardous areas.

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APPENDIX A
Some examples of protection

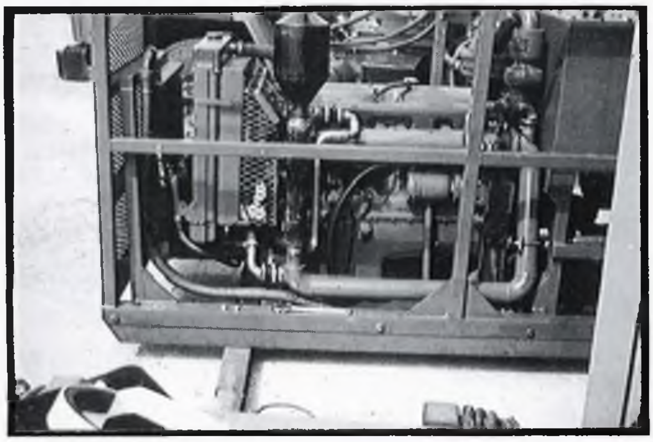

Plate 1. This hydraulic power pack is equipped with a Chalvin overspeed valve and flame arrester in the air intake. There is a water-cooled manifold and exhaust with all welded connections and spark arrester. The cleaning or inspection of such a spark arrester is impossible as no provision is made for any maintenance. (All welded piping.)

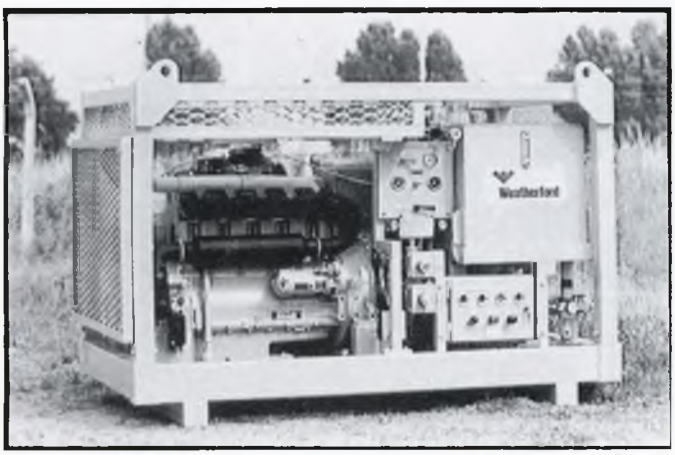

Plate 2. Hydraulic power unit driven by Deutz, 5 cylinder, 40 kW air-cooled diesel engine.
Standard Deutz exhaust system additionally equipped with spark arrester. Efficiency of standard joints on air intake and exhaust manifolds were established by a pressure test to 10 bar. Small flame arrester can be seen on the end of the crank case breather pipe; however, its location in the proximity of exhaust piping should be avoided. Red ESD button on control panel activates shutdown device in air intake.

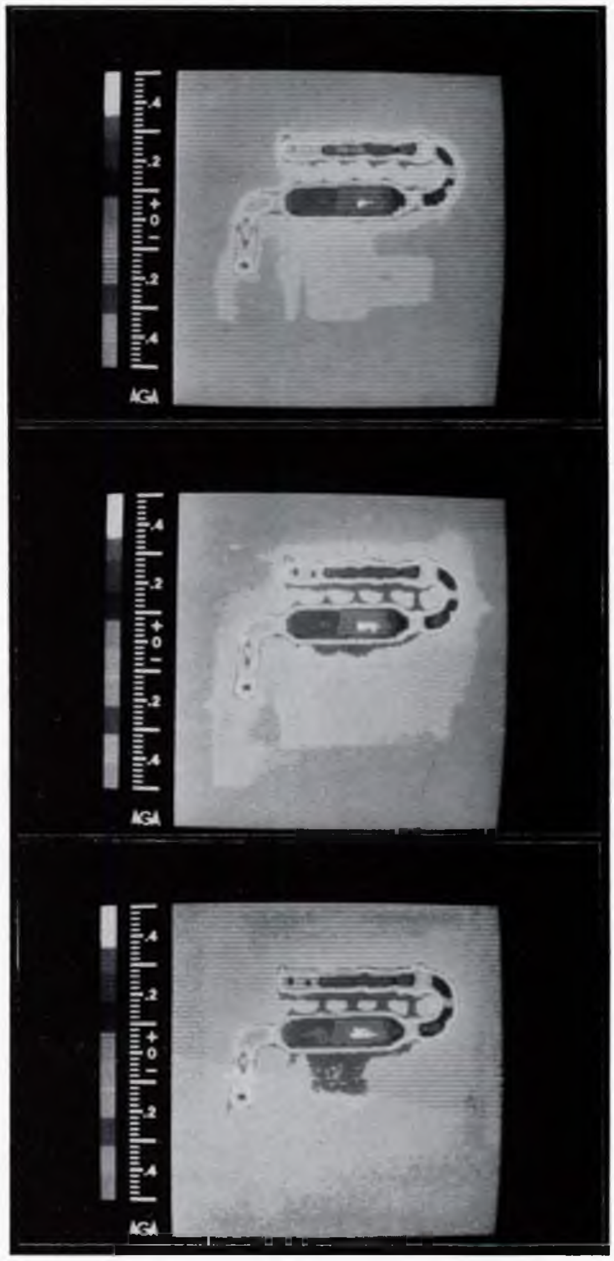

Plate 3. Thermographic survey is carried out for determining the hot spots within the engine.
Thermographs of 40 kW air-cooled engine on test. Various shades correspond to different temperatures of the engine parts. The hottest point on the surface of the unit is the spark arrester casing. Temperature was measured with thermocouples installed in accordance with the thermograph indicators. From top to bottom: max temp 185°C, 2900 psi; max temp 185°C, 3000 psi; max temp 183°C, 3100 psi.

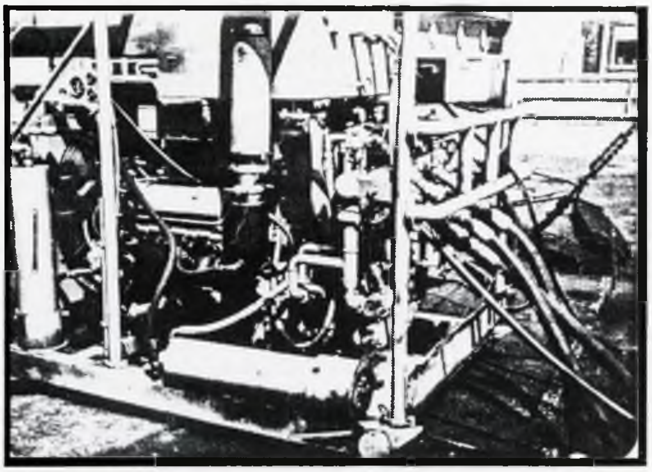

Plate 4. Hydraulic power pack after modification. Spark arrester connections are unsatisfactory, hydraulic characteristics of the spark arrester were not considered and as a result the surface temperature on the bend became more than 350°C. Without a spark arrester it is 180–200°C.

Plate 5. Example of combined differential pressure shutdown valve and flame arrester.

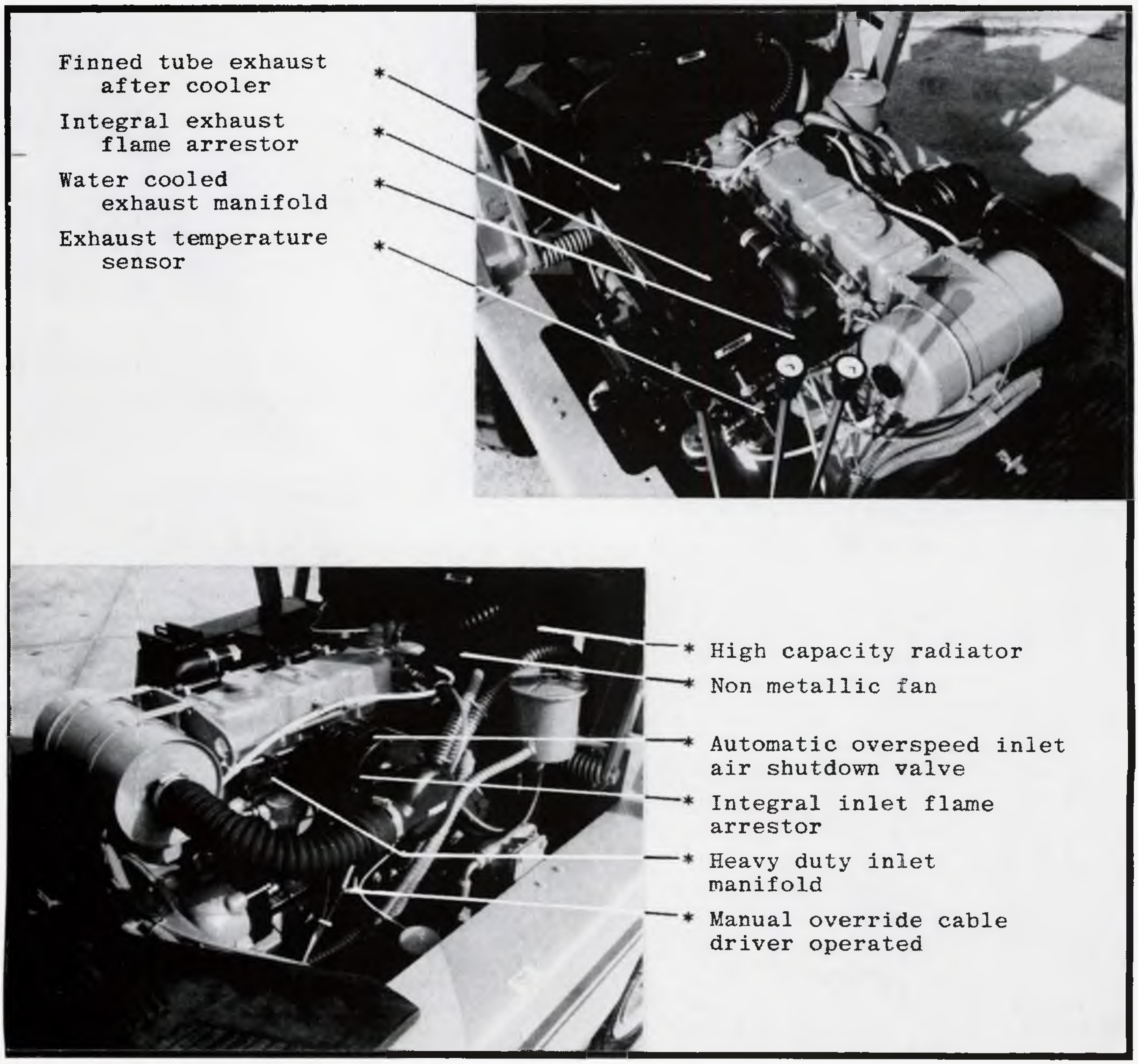

Plate 6. Example of protection system used

APPENDIX B

Symbols used in tests and calculations – Applicable theory for protection of diesel engines

1. Ignition

E_{ign}	=	minimum ignition energy
λ	=	thermal conductivity
d	=	quenching diameter
v_0	=	specific volume of original mixture
T_0	=	initial temperature of the gas
T_e	=	final combustion temperature
T_w	=	surface temperature
q_w	=	local value of heat flux
Q	=	molar reaction enthalpy (heat of reaction)
A_φ	=	coefficient of influence of equivalence ratio on the overall rate of heat generation
φ	=	equivalence ratio
R	=	molar gas constant
f	=	function

2. Combustion and probable combustion régimes

P	=	pressure	of the gas entering combustion zone (subscript 1) and of combustion products (subscript 3). For initial parameters of original gas, subscript o is used.
v	=	specific volume	
E	=	specific energy	
U	=	velocity	
ρ	=	density	
H	=	enthalpy	
T	=	temperature	
κ	=	ratio of specific heats	
Q	=	heat of reaction of unit of gas	

C_1, C_0	=	speed of sound in original gas
D	=	detonation velocity
V_c	=	flame velocity in deflagration
P_{exp}	=	max explosion pressure
T_{exp}	=	max explosion temperature
T_e	=	final temperature of combustion
ΣM_e	=	number of moles after reaction
ΣM_1	=	number of moles before reaction
Re	=	Reynold's number
d	=	pipe diameter
L_{min}	=	minimum length of pipe required for full development of turbulence or laminar flow
M	=	Mach number
D_0	=	lossless detonation velocity
D^1	=	detonation velocity with losses
ΔD	=	lost velocity
P_2, T_2	=	pressure and temperature behind the shock-wave in lossless detonation
P_2^1, T_2^1	=	pressure and temperature behind the shock-wave in detonation with losses
$T_{ign\,sh}$	=	shock ignition temperature
ΔH_R^0	=	standard heat of reaction
ΔH_R^t	=	heat of reaction at combustion temperature
c_p	=	specific heat at constant pressure
c_v	=	specific heat at constant volume
Q_{N_2}	=	heat lost on heating of nitrogen
E_d	=	dissociation energy
K_p	=	equilibrium constant
M_g	=	mass of 1 kmol of gas/air mixture
σ	=	friction force
f	=	coefficient = $0.0083\sqrt{Re}$
w	=	velocity of products of reaction
c	=	local speed of sound
q	=	thermal losses
γ	=	ratio of specific heat at ambient conditions for shockwave relations

Discussion

M Camilleri (American Bureau of Shipping) I congratulate the author for giving us an excellent paper, in particular with respect to the identification of sources of ignition in diesel engines. With regard to the use of protected diesel engines in hazardous areas offshore, it is felt that the 'hazardous area' is dealt with very briefly and only in general terms. The resulting impression is that a protected diesel engine may be installed in any hazardous zone. Accordingly my question is: would all the hazardous areas be defined as zone 2 where a protected diesel engine will be acceptable, or as zone 1 where the protected diesel engine requires additional protection?

A Sokolov (LR) The hazardous areas classification is based on the assumption of the possibility of a gas presence in the given area, but not on the assumption of potentially hazardous equipment being present in the area. The fact that protected or unprotected engines are installed in zone 2 is not changing the hazardous environment as such (zone 2 will still be zone 2 as the probability of gas escape remains the same), but the protected engine makes the operation in such a hazardous zone safer.

G W R C Easton (Chevron UK Ltd) I would like to thank Mr Sokolov for this very interesting paper on the ignition sources presented by diesel engines, particularly with reference to offshore installations. However, what concerns me more than anything else, is that given a diesel engine, and indeed any combustion engine, with an ignition source in an environment where the fuel for combustion is inherently present, what, or how, do we define in absolute terms a 'safe area'? If we accept the ideas postulated in Mr Sokolov's Figs 5–7, it would seem that not only should we be looking at the protection of diesel engines, but also at the likelihood of the presence of a 'flammable gas–air mixture' (especially in the light of recent events in the North Sea) that could possibly be ignited by any heat engine either internally or externally.

A Sokolov (LR) I agree with Mr Easton that the likelihood of the presence of a flammable gas–air mixture determines the necessity of the protection of diesel engines or in fact any heat emitting apparatus (gas turbines, compressors, glycol reboilers, etc). I also agree that the present definition of hazardous areas, especially zone 2 areas, and the safe areas requires clarification if not complete revision.

D L Morgan (KCA Drilling) You suggest that a fully protected zone 2 engine should be shut down in a high gas situation.

In a well control situation this could shut down the means controlling the problem. Your comments please.

A Sokolov (LR) In a well control situation the engines involved are normally manned and the decision to shut down the engine lies with the operator. If the engine is protected as recommended in the paper (Case C), then protective devices of the engine will prevent immediate ignition of the gas and the gas detection system will give the operator an indication of a gas presence. The operator then should make the decision to shut down the engine or not on the basis of the overall situation.

A D Chaplin (Department of Energy) From experience, a question that is constantly raised is: what is the maximum allowable surface temperature of a diesel engine operating in a hazardous area? Does the author know of either a method by which the maximum surface temperature can be determined from a given set of operating conditions, or of a maximum surface temperature which can be used that has been determined either by experiment or experience? This same question also applies to the determination of the maximum allowable exhaust gas temperature at the point where first direct contact is made by the exhaust gas stream with the outside atmosphere.

It is agreed with the author that this subject is complex and depends upon variables such as ventilation flow rates, flammable atmosphere 'residence time' at a hot surface, gas composition and auto-ignition temperature to name but a few. The subject is further complicated by equipment operating parameters such as maximum flame arrester operating temperatures, exhaust gas cooler efficiencies, the effects of fouling and the engine temperature when stopped.

To date, the arbitrary safe maximum temperature appears to be set at or about the lowest auto-ignition temperature of any flammable hydrocarbon product that can be found on an offshore installation. This limitation is restrictive, overly conservative and inefficient, but in the absence of anything better, must be adhered to. Any help the author can provide in this area would be welcomed by the Industry.

A Sokolov (LR) Mr Chaplin quite rightly noted that a great number of variables exist which influence surface ignition temperature. In view of this, and the fact that the theory of surface gas ignition is not yet in a form which can be used for solving practical problems, it is considered that the best way of determining surface ignition temperatures is by experiment. Experiments can be conducted, for example, as follows.

In the case of exhaust manifolds and piping, they should be grouped on the basis of engine power (size). For example, engines with exhausts from 50 mm to 100 mm can go in one group and from 100 mm to 200 mm in a second group. Then experiments for determining surface ignition temperatures could be carried out for a chosen gas, perhaps the one which is in highest proportion in the group. This would provide a safety margin for all exhaust gases that are present in smaller amounts in the group, as their surface temperatures will be higher.

The condition of heat exchange between the hot surface and the gas for every case is impossible to predict. Therefore it would be reasonable to carry out experiments in the most unfavourable conditions, ie conditions of free convection (no air–gas mixture movement) so that when these results are applied to the real situation it will provide an additional safety factor.

From the knowledge already available it can be predicted that in any case the surface ignition temperature will be at least 40–50% higher than the auto-ignition temperature of the gas. Diesel engines, and in fact any heat engines, should then be assigned a notation which indicates clearly the gas atmospheres in which the engine can operate. The platform operator should then choose a diesel engine for a particular environment (for example CH_4 engines cannot be placed in areas where gas condensate vapours may be present).

As far as the ignition of gas–air mixtures by hot exhaust gases is concerned, extreme caution should be exercised in accepting the results of experiments, as the exhaust gas stream turbulence will be different in every experiment, and therefore a considerable number of experiments should be carried out before a reliable mean temperature of turbulent flow is established.

G Victory (Retired) The author rightly says that 'Every conventional unprotected diesel engine can be considered to be hazardous' when used on offshore installations. This also applies to any electrical equipment in the same environment, and similarly to either diesel engines or electrical equipment fitted on oil tankers and gas carriers! Rules and regulations are applied in an effort to obtain a suitable level of safety, but a number of very catastrophic incidents in recent years on both ships and installations leads one to query whether in fact an 'acceptable level of protection' is actually being achieved.

In the Introduction to the paper we are informed that one option is to 'install the engine in a pressurised enclosure where the intake for pressurisation is arranged from a 'safe' area and exhaust gases are discharged to a 'safe' area.' But how safe is 'safe'? Let us examine the origin of the use of the word 'safe' in this context. It does not mean 'safe' in the accepted sense of the word but 'safe' by definition – the definition being taken from the IEC Publication 79-10 – which in my opinion does not lay down adequate safeguards for ships or installations where flammable vapours can be released in very close proximity to accommodation, personnel spaces, and what in effect are the equivalent of small power stations!

Let us examine the source and intent of IEC 79-10. It goes back many years and was produced to cover shore storage and refinery installations where different services such as storage spaces, refining processes, generator stations and personnel cafeteria and resting areas could be comparatively widely dispersed.

Shore authorities would not permit a hotel, a power station and potentially dangerous oil storage and processing installations to be built contiguous to each other, much less if they were separated only by a steel bulkhead. Yet this is the position we find on ships and offshore installations. Perhaps we need a more rigorous statement of fundamentals than is provided by IEC 79-10. These cover only normal operations and are not adequate for unusual events. I have spoken to this effect at both the UK IEE in the production of 'The electrical installations in ships', and the IEC meeting at Washington which covered the electrical safety of tankers, but to no avail!

The definitions of zones 0 and 1 are not unreasonable as the presence of flammable vapour at all times is contemplated, but the definition of zone 2 considers only that such conditions are unlikely to occur in 'normal operating' conditions, and only for short periods. Yet in 'unusual' conditions an entire ship or installation can be enveloped in a gas cloud and the time for which explosive conditions exist only affects the 'probability' that fire or explosion will take place and not the 'possibility' that it will do so! But then the code goes on the say 'all other areas are classified as safe'. This is most misleading, for not only are unusual circumstances ignored in zone 2 classification, but also there is the assumption that all spaces outside zone 2 are safe, and can therefore be used as source points for ventilation or pressurisation of areas, which are not otherwise protected, and it therefore produces a false sense of safety. There were no safe areas on the *Royston Grange* and *Betelgeuse*, to mention just two.

In the event of a massive release of gas or in certain weather conditions, these ships and installations are so compact that there are really no spaces completely safe from which pressurised air may be taken, within the overall blanket envelope of the unit. Yet the author accepts this very suspect standard as covering his first option – installation in pressurised enclosures – for diesel engines, and then goes to great lengths to identify the great hazards of such machines in potentially hazardous surroundings.

Would the author agree that for ships and offshore installations the IEC definitions are, perhaps, not stringent enough to produce an adequate level of safety? Perhaps they should start with a better definition of a 'safe space' such as 'a safe space is one in which an explosive concentration of gas cannot occur in 'normal' or 'abnormal' operating conditions! All other areas are hazardous to a greater or lesser extent'. They could then go on to identify other zones by reference to the specific hazards of the equipment or operations carried out therein, but such considerations must include both normal and abnormal operating conditions.

The author is much more realistic in his second option in which he assumes that the diesel engines will operate in 'unsafe' conditions – stage 3 of the 'incident iceberg' – and I congratulate him on the wealth of detail and the extent into which he has gone in establishing the fundamentals involved in the ignition of explosive vapours! Especially important are the determination of quenching distances and quenching diameters; essential in the design of flame arresters. Would the author agree that, having regard to the very small values of these criteria, the usual heavy gauge coarse mesh so-called 'flame arresters' fitted to tanker outlets would not prevent the passage of flame, and are only really effective in stopping people dropping tools or nuts and bolts into the oil tanks?

Incidentally some years ago I witnessed the testing by Shell (or possibly BP) of an ICI product called, I believe, Recumet. This was a slab of honeycomb copper filaments, manufactured by electrolytic deposit of copper onto a polythene base, after which the polythene was burned out leaving a matrix of very fine copper filaments which had low resistance to the flow of air or gas through it. In the tests at Borehamwood on a pipe about 25 ft long and of 2 ft diameter, filled with a propane–air mixture, which was remotely ignited, only a dull thump followed by a puff of smoke resulted from the Recumet, whereas flames some 10 ft long emerged when three normal flame arresters were placed in series at the end of the pipe. Unfortunately it was found to pick up liquid droplets when used on crude oil outlets, but it would be perfect for 'clean gas' outlets!

The author gives the flammable limits for methane and natural gas, and although I realise that he is dealing with offshore operations, I would have thought that those for LPG and crude oil vapours, which I believe are lower than either of the above, and are more likely to be met by marine engineers, would be worth quoting.

I am pleased to see that the author has drawn attention to the danger of a diesel engine running away, even with the fuel shut off, if it is drawing in flammable vapour. This danger is not always appreciated but there have been a number of cases of generators running away to destruction in a ship's main engine room – incidentally, a 'safe' space! However I see no reference to two likely sources of fire in diesel engine spaces: burst high pressure and other fuel pipes and scavenge fires. I wonder how the author would deal with those possibilities.

The author also draws attention to the fact that the minimum ignition energy required to ignite a methane–air mixture 'is so small that the spark cannot be seen with the naked eye'. He does not mention that an incendive spark can result from a person wearing nylon or man-made underwear touching an earthed object. This possibility had to be considered in the formal investigation into the *Mactra* explosion but it was rejected, not because it was not possible, but because the man in question had left the foredeck when the explosion occurred.

Some minor points in the paper for comment are:

1. In the section 'Gas detection systems as a means of protection' the author says, 'Even with an efficient gas detection system a flammable mixture could come into contact with a hot surface or engine sparks at the same

time as the sensor detects gas'! This is also applicable to any other source of ignition. Yet manual alarms from detectors in the air ducts are often used to protect pressurised 'safe' spaces. As the author points out, by the time action is taken it could well be too late! So I would suggest that the only realistic system is that shown in Fig 7 with the emergency shut down being automatically operated. Even so all gas detection systems are fallible and can give a false sense of safety.

2. The author considers that, because of the greater hazards, a supercharged engine should be derated by 50%! Apart from being uneconomical, why use a supercharged engine in the first place? Assuming a 50% uprating by supercharging, the consequent 50% derating would leave us with 75% of the unsupercharged engine's power.
3. Finally, the author draws attention to the fact that 'Regular inspection and testing of . . . safety arrangements, together with good/regular maintenance are essential factors for the safe operation of diesel engines in potentially flammable atmospheres.' This is a *sine qua non* for all marine activities, yet could well be the Achilles heel of those who try to improve safety at sea. For those to whom it should be most important are often those who neglected the basic precautions but console themselves with 'it cannot happen to me.' How wrong they can be! This is one aspect of safety for which I cannot suggest an answer except that we just hammer away the problem and use any casualty to enforce the point!

Perhaps I might close by suggesting that, with all the disadvantages the author has identified in using numbers of diesel engines (engines below 50 kW are mentioned) I am surprised that anyone should persist in using a number of individual diesel engines for a number of applications on the same installation. I would have thought that a central diesel power station with a large hydraulic pump to supply power units in various locations would be easier to locate safely and protect than a number of smaller units spread around the installation. Safety is difficult to achieve – why make it more difficult?

A Sokolov (LR) 'How safe is safe?' Well, in absolute terms the meaning of the word depends upon the subjective attitude of the user and the environment where the words 'safe' and 'hazardous' are used. Probably there are no safe areas on the whole planet. People are killed in their homes, on mountains, in jungles, not to mention the modern means of transport and industry. So, as there is no absolute safety, we have to accept the definition of 'safe' as given by international or national standards in relative terms, ie environment or equipment will be considered safe and remain safe if the people involved exercise reasonable caution. For example, the domestic kitchen is a safe area until somebody leaves the gas on and an explosion results. I think the standard definition of safe should be accepted. However, I can agree with you that in some applications the present definition of safe should be revised.

You gave good examples of flame arresters used in tankers. Of course due consideration should be given at the design stage in order to produce an efficient flame arrester suitable for particular flames and flame speeds.

Regarding your example of fire initiation by burst fuel pipes and fire in scavenge spaces, I would like to mention that this is a 'feature' of most conventional diesel engines. However, to prevent fires from bursting fuel pipes, sheathed fuel pipes can be used. There are a number of designs of such fuel pipes on the industrial market. Scavenge fires could only occur if there is something to burn in the scavenge space (oily deposits). Obvious treatment is to increase frequency of the cleaning of such spaces. I am sure most marine engineers who used to work on big low speed engines have experience in such work.

Regarding the supercharged engines, the paper was intended to draw attention to the increased danger of using supercharged engines in hazardous areas. Due to the higher than atmospheric pressures in the air intake, more violent combustion and explosions could take place than in the air intake of an engine without supercharging. Therefore extra care and consideration should be taken when designing and installing protection devices of supercharged engines. It was not my intention to suggest that a supercharged engine should be derated by 50%, as the subject of derating of supercharged engines was not considered in the paper.

Your proposal regarding a 'central diesel power station' should be addressed to oil companies for consideration when designing platforms of the future. But at present various small (and large) power packs are brought onto offshore installations during shutdown and maintenance programmes on a temporary basis. These need to be located at various points on the platform and I think that a centralised hydraulic power supply would be very difficult to implement.

How can you deliver hydraulic power to a crane for example or a fork lift truck? I agree with you that 'Safety is difficult to achieve', but safety is a compromise between the risk involved and the cost of eliminating the risk. 'Absolute safety' is not achievable.

D Lineham Please comment on whether there exists a hazard through exhaust from vessels attending platforms, ie supply boats, etc.

A Sokolov (LR) Under the present conditions of operation supply boats cannot approach a platform without the permission of the Offshore Installation Manager. When they approach and work in close proximity to the platform their engines never operate at full power so the exhaust gas temperature of the supply boat engines will be low. Also, the presence of a gas–air mixture in the area of supply boat operation is very unlikely with gases that are lighter than air. However, if gases that are heavier than air are involved (eg propane, butane), then of course a hazard exists, especially bearing in mind the presence of not just diesel engines but also electrical equipment on the supply boat.

W S Rogers (LR) Mr Sokolov is to be congratulated on a very comprehensive technical paper. With respect to the more practical side the author's comments are sought on the following.

The Department of Energy Guidance Notes Part IV, Section 7, 'Mobile Engines', state that maximum surface temperatures on portable units such as wireline units or welding sets should not exceed 250°C, and for forklift trucks should not exceed 350°C. Classification Society rules restrict hot surface temperatures on ships and mobile units to 220°C, and BS 6680, 'Diesel engines for use in coal mines and other mines susceptible to firedamp', restrict the external surface temperature to 200°C except in coal mines where the maximum surface temperature allowed is 150°C.

Classification Societies are required to assess offshore installations on a global basis, and whereas it is agreed that ventilation conditions are different in an offshore installation as compared with a mine, with respect to hot surfaces of large mechanical equipment in zone 2 areas, many variables exist that may favour an explosion after a major gas leak as follows.

1. The unpredictable severity of a major gas leak.
2. Ventilation rates changing with wind speed and direction.
3. Forced ventilation may have ceased.
4. Presence of diesel oil and other flammable fluids.
5. Gas to air mix ratios.
6. Changing reservoir characteristics.
7. Possible hot spots in large exhaust ducts and on machinery.
8. Malfunctioning mechanical and electrical equipment.
9. Unpredictable conditions following a major gas leak.

In view of the above, and considering that ignition temperatures are scientifically established under laboratory conditions using pure gases, is it reasonable to allow surface temperatures of mechanical equipment in zone 2 areas to exceed 200°C?

With reference to the length of flame path across closed joints, OCMA Mec 1 asks for 13 mm minimum through or across any joint, whereas BS 6680 has allowed 9 mm minimum flame path length for a flame path gap of 0.2 mm. Due consideration should be given to this when designing the cylinder head joints and manifold joints. We are aware that some manufacturers have criticised this requirement as excessive and the authors comments are sought on this point.

Under the 'Overspeed shutdown devices' section, it is noted that the opening paragraph states 'Any overspeed shutdown sensor should activate a shutdown device in the air intake (not shut down the fuel system)'. This is not fully understood as the Department of Energy Guidance Notes Part IV, 7.1.2, state that an inlet valve on the fuel supply should be automatically operated on engine overspeed. Could the author clarify as it is our understanding that the basic protection devices on any diesel engine would shut off the fuel supply on overspeed and for a protected engine operating in a hazardous area the aspirating air inlet shutdown valve is an additional independent protecting device?

A Sokolov (LR) Mr Rogers has obviously read many regulations and guidance notes on allowable surface temperatures, but may I draw your attention to the following. The latest addition to requirements suggests that 200°C for surface temperatures of diesel exhausts is safe for operation in a gaseous environment. What particular gas do they mean? Methane with an auto-ignition temperature of 550°C, or gas condensate vapours with an auto-ignition temperature of 275°C? I think that an engine surface temperature should be considered for a given gas, as surface ignition temperatures of various gases are different under the same conditions of heat exchange. An engine surface temperature of 200°C is unjustifiably low for methane and natural gas. This gives an expected surface temperature in conditions for free convection of 30-50% of their auto-ignition temperatures.

Regarding the cylinder head joint, I can agree with Mr Rogers only in the case of metal or metal joints. However even in this case the size of engine should be taken into consideration, and on small engines, explosion tests should be carried out to prove the efficiency of for example a 6 mm joint.

Every diesel engine is equipped with a governor which is also set to react or overspeed and decrease or completely shut down fuel if the engine exceeds a certain percentage of the maximum rev/min. An independent protective device in the air intake is, as Mr Rogers noted, an additional line of defence. However, in the past, attempts were made to fix pressure operating devices in air intakes connected to the fuel rack. Of course such arrangements were defeating the object of the exercise, and they were meant, in the part of the paper on overspeed, to emphasise the necessity of air intake shutdown.