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CONVERSION OF LIQUID TO GASEOUS FUEL FOR PREVAPORISED PREMIXED COMBUSTION IN GAS TURBINES

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ABSTRACT

Stationary gas turbines for power generation are increasingly being equipped with low emission burners. By applying lean premixed combustion techniques for gaseous fuels both NO_x and CO emissions can be reduced to extremely low levels (NO_x emissions <25vppm, CO emissions <10vppm). Likewise, if analogous premix techniques can be applied to liquid fuels (diesel oil, Oil No.2, etc.) in gas-fired burners, similar low level emissions when burning oils are possible. For gas turbines which operate with liquid fuel or in dual fuel operation, VPL (Vaporised Premixed Lean)-combustion is essential for obtaining minimal NO_x-emissions. An option is to vaporise the liquid fuel in a separate fuel vaporiser and subsequently supply the fuel vapour to the natural gas fuel injection system; this has not been investigated for gas turbine combustion in the past.

This paper presents experimental results of atmospheric and high-pressure combustion tests using research premix burners running on vaporised liquid fuel. The following processes were investigated:

- evaporation and partial decomposition of the liquid fuel (Oil No.2);
- utilisation of low pressure exhaust gases to externally heat the high pressure fuel vaporiser;
- operation of ABB premix-burners (EV burners) with vaporised Oil No.2;

- combustion characteristics at pressures up to 25bar.
- Atmospheric VPL-combustion tests using Oil No.2 in ABB EV-burners under simulated gas turbine conditions have successfully produced emissions of NO_x below 20vppm and of CO below 10vppm (corrected to 15% O₂). 5vppm of these NO_x values result from fuel bound nitrogen. Little dependence of these emissions on combustion pressure has been observed. The techniques employed also ensured combustion with a stable non luminous (blue) flame during transition from gaseous to vaporised fuel. Additionally, no soot accumulation was detectable during combustion.

NOMENCLATURE

| | | |
|-----------------|---------|--|
| CO | vppm | CO in dry exhaust gas at 15% O ₂ |
| D | µm | Droplet diameter |
| FBN | % (w/w) | Fuel bound nitrogen content |
| NO _x | vppm | NO _x in dry exhaust gas at 15% O ₂ |
| O ₂ | % (v/v) | O ₂ concentration in dry exhaust gas |
| p | bar | Pressure |
| t | s | Time |
| T _{ad} | K | Adiabatic flame temperature |
| T | °C | Temperature |
| U | % (w/w) | Conversion ratio of cracking reactions |
| λ | | Air/fuel ratio |

| | | |
|----------|-------|-------------------------------|
| Ω | kg/kg | Water/fuel mass ratio "omega" |
| UHC | vppm | Unburned hydrocarbons |
| Z | mm | Distance along the axis |

INTRODUCTION

As described by Aigner et al. (1990) and Döbbling et al. (1994) it is possible to reduce the NO_x and CO emissions of gaseous fuels by generating lean premixed combustion conditions. The premixed combustion of liquid fuels however, is more complicated as the fuel must additionally be atomised and vaporised before being mixed homogeneously with the combustion air.

In some cases liquid fuel is injected into the hot combustion air stream and vaporised (Sattelmayer et al. 1990, Poeschl et al., 1994). This concept, often called LPP (lean combustion of Prevaporized/ premixed fuels), is a most promising method for reducing NO_x and CO emissions. However, under typical modern gas turbine conditions the ignition delay time of fuels is very short (Spadaccini et al, 1975 and 1982). High combustor inlet temperatures and pressures can consequently activate self-ignition and flame stabilisation in the premixing zone, especially if using heavier hydrocarbon fuels. In addition, further difficulties arise from carbon deposits on hot metal surfaces caused by pyrolysis of hydrocarbon fuels and from soot formation due to incomplete evaporation of fuel droplet. Furthermore, a separate supply and injection system for liquid fuels becomes necessary if the combustor is to operate in a dual fuels powered gas turbine.

To overcome such complications, Stoffel and Reh (1995) investigated the feasibility of a new approach: VPL (vaporised premixed lean) combustion. It consisted of a separate, externally heated fuel vaporiser in which the liquid fuels were fully converted to gaseous fuel, and then injected using a mixing device for generating a homogeneous mixture of fuel vapour and air. Using a 15 kW surface burner in premixed mode, air / fuel vapour was burned, and reduced NO_x emissions were obtained.

In this paper, a modified VPL combustion technique is proposed which has been tested under gas turbine conditions. A pressurised fuel evaporation system heated externally by simulated exhaust gas was designed, assembled and tested. Experiments were conducted with mixtures of fuel vapour and steam ($0.3 < \Omega < 1.0$) so as to maximise combustor performance and reliability. By utilising equipment available at ABB the emission characteristics of vaporised fuel oil No. 2 have been determined under atmospheric and high pressure conditions. Atmospheric tests were conducted using an ABB EV (EnVironmental) burner whilst high pressure tests made use of burner geometries with simple flame stabilisation techniques.

CONCEPT

The combustion of liquid fuel can be split into four essential processing steps, namely atomisation, evaporation, mixing and combustion; each of which may be investigated and optimised

independently. Traditionally, fuel is atomised by injecting it through a nozzle into hot combustion air, where it then rapidly evaporates as a result of heat transfer with the surrounding air. Following the formation of a sufficiently homogeneous mixture ignition is initiated via one of many flame stabilisation techniques. A problem that may result from this type of injection is that incomplete evaporation occurs due to poor atomisation and poor mixing of the fuel spray and air.

In order to preclude such behaviour liquid fuel can be atomised and evaporated separately in the absence of air, before being injected into the combustion air. Part of the waste heat from a gas turbine exhaust (550°C to 700°C) can be utilised in heating the liquid fuel vaporiser as well as a steam generator. Thus, a proportion of the exhaust heat is recycled to the high temperature zone of the process, and hence raises overall turbine efficiency according to fundamental thermodynamics. Figure 1 shows the concept of a VPL combustion process for a gas turbine.

To achieve flash vaporisation and avoid the build up of carbon deposits on the vaporiser walls, superheated steam is used as atomisation gas. This has several advantages:

1. Steam-assisted atomisers are very suitable for highly viscous liquid fuels and operate with minimal amounts of steam over a wide turndown ratio. Due to the immediate and intensive heat exchange between steam and fluid a portion of the fuel can be vaporised during atomisation resulting in a very fine spray.

2. By adding steam to fuel oil the partial vapour pressure and hence the boiling points of the oil are lowered. A mixture of steam with fuel oil therefore results in a complete evaporation of the fuel oil at a lower overall temperature than would have been the case had the steam not been present. Figure 2 illustrates this point for the boiling range of fuel oil at 30bar. The two curves show a difference in boiling temperature of approximately 120°C for steam / fuel mass ratios of 0 and 0.5.

3. Reactions of carbon and hydrocarbons with water can remove carbon deposits by reforming into H_2 and CO.

4. Combustion temperatures are decreased by steam addition which results in lower NO_x emissions.

During evaporation and superheating in the vaporiser, a small portion of fuel is thermally cracked into smaller molecules. In general the cracking reactions of hydrocarbons are strongly endothermic. Fuel vapour consequently absorbs extra heat chemically. The cracking rate is dominated by temperature and mean residence time in the vaporiser. By incorporating atomisation and evaporation with superheated steam into one vaporisation process and including cracking reactions, the modified VPL process can more effectively recover available exhaust heat than conventional recuperation.

The vaporised fuel is then burned analogously to gaseous fuel in a low NO_x gas burner.

TEST FACILITY

Fuel vaporiser system

A liquid fuel vaporiser unit has been designed and built to convert the liquid fuel to a gaseous state. It consists of a vaporiser, a steam generator, two 10kW air heaters, two high pressure piston membrane pumps, a special fuel atomisation nozzle and an operating control system.

The vaporiser shown in Figure 3 is in essence a double pipe, high pressure heat exchanger. Fuel oil and steam are injected into the top of the vaporiser and the fuel oil is evaporated as it passes down through vertical channels. Additional ducts surround the fuel channels for the transport of hot air which is used to simulate the gas turbine exhaust gases. This is depicted in the cross section shown in Figure 3. By injecting the air at the base of the vaporiser the two streams flow in opposite directions and hence an efficient heat exchange between the hot air and fuel oil is achieved. This heat exchange consequently evaporates the fuel oil. In this case, air is preheated to temperatures ranging from 400 to 700°C using an electrical heater. Inner kernels of differing lengths allow the volume and length of the vaporiser channels to be set to achieve the desired evaporation time and fuel oil throughput. A throttle at the exit regulates the operating pressure of the system between 5 and 30bar.

The fuel injector at the head of the vaporiser consists of a steam assisted atomiser in which superheated steam is used for atomisation, as shown in Figure 4. The steam generator supplies superheated steam to the nozzle at temperatures up to 650°C and pressures attaining 60bar. By introducing a layer of ceramic insulation fuel and hot steam are kept separate up to the mixing positions inside the nozzle. Hence, build up of possible carbon deposits inside the fuel tube is avoided. The nozzle can be operated at steam/fuel weight ratios, Ω between 0.3 and 1.0. The internal mixing inside this steam assisted atomiser has the advantage of providing efficient and direct heat exchange between the fuel and superheated steam, and consequently improves fuel atomisation. This is attributable to two effects resulting from heat exchange with the hot steam:

- fuel oil viscosity is reduced due to the rise in liquid temperature,
- evaporation of light hydrocarbons already begins in the nozzle.

Additionally, with this apparatus flash evaporation of fuel droplets can be achieved.

Fuel and water mass flow rates and their ratio are independently controlled by high pressure metering pumps and coriolis flow meters. The vaporised and superheated mixture of fuel and steam is either fed to the combustor through heated, thermally insulated pipes, or alternatively to a fuel condenser for analysis. For safety reasons, the second line also serves as an emergency vapour release mechanism. All temperatures are measured with NiCr/Ni thermocouples.

COMBUSTORS

The combustion of oil vapour has been investigated with two test rigs, one operating at atmospheric pressure, the other at various pressures up to 30bar. The sampling of the exhaust gases was accomplished by using traversable, water-cooled probes connected to gas analysers with heated tubing. The gas samples were analysed for CO, CO₂, O₂, NO, NO₂ and UHC with infrared absorption, paramagnetic, chemiluminescence and flame ionisation detectors. Flame and probe temperatures were measured with Pt/Rh and NiCr/Ni thermocouples.

The atmospheric test rig is shown schematically in Figure 5. A scaled down model of an ABB double cone EV burner with an exit diameter of approximately 100mm was used for the first experiments. The ABB EV burner consists of two halves of a cone that are shifted to form two inlet slots of constant width. Combustion air entering through the slots is mixed with gaseous fuel injected through many small holes, which are situated on the inlet ports of the slots. A vortex breakdown is formed near the burner outlet as a result of the central recirculation, which stabilises the flame in free space (Sattelmayer et al, 1990). The burner was installed in a fibre ceramic combustion chamber of approximately 320mm inner diameter. Combustion air was supplied at temperatures up to 450°C. Initially the flame was ignited and run using natural gas, before switching to vaporised oil. By monitoring the entire process with the aid of a video system mounted downstream of the combustion chamber, it could be seen that transition between the two fuels occurred without flame extinction or noticeable pulsation.

The high pressure test rig depicted in Figure 6 consisted of modular elements which enabled effective modification of important combustion characteristics. Following the preheating section, (capable of achieving air temperatures in excess of 600°C), the fuel vapour is injected into the air stream in a prescribed manner with a nozzle and allowed to mix homogeneously with the hot air over a specified mixing section. If necessary, it was possible to introduce static mixing elements into the ensuing pipe to improve the uniformity of fuel vapour concentration, in the air. The flow then enters a combustion chamber, equipped with a ceramic liner, and the flame is stabilised by a sudden expansion in diameter as the mixing section enters the liner. The vastly reduced flow velocities in the combustion chamber enables the investigation of sufficient residence times to guarantee complete combustion prior to the point where the flow turns through 90°, is throttled and exhausted to the atmosphere.

In order to maintain the fuel in an evaporated state between vaporiser and respective test rigs, heated supply lines were employed with operation temperatures in excess of 500°C. In addition, a specially insulated injection flange was utilised in conjunction with the high pressure tests, which prevented the fuel oil vapour temperature from dropping below the upper boiling point.

FUEL PROPERTIES

Compared with other liquid fuels such as gasoline, kerosene, naphtha and ethanol, Oil No.2 has high values of liquid viscosity, a broad boiling range as well as low values of autoignition temperature. It is also more difficult to achieve flash evaporation of Oil No.2 than for other liquid hydrocarbon fuels and therefore this fuel was used in all experiments.

Natural gas represents a typical turbine fuel and was hence also used as a reference gaseous fuel for the combustion tests.

TABLE 1 COMPOSITION OF FUEL OIL NO. 2

| Properties | Oil No.2 |
|---|----------|
| Analysis, % (w/w) | |
| C | 86.44 |
| H | 13.41 |
| O | 0 |
| N | 0.0124 |
| S | 0.140 |
| H / C, mol/mol | 1.85 |
| NO _x , vppm 15% O ₂ dry from FBN source | 5.2 |

*Analysis by Swiss Federal Laboratories for Materials Testing and Research (EMPA)

RESULTS AND DISCUSSION

Fuel Evaporation

Since the evaporation of fuel takes place in a hot steam environment, the problem of auto ignition during this process is overcome. Figure 7 shows typical bulk residence times of the fuel/steam mixtures in the vaporiser between limiting operating conditions, namely pressures between 5 and 30bar and oil mass flow rates from 0.5 to 3 g/s. As can be seen, residence times vary from approximately 2 to 10 seconds, which is much longer than the evaporation time of most droplets. This short evaporation time of droplets is simulated in Figure 8 for hexadecan droplets in a hot stream of steam under high pressure conditions. Figure 9 presents inlet temperatures of fuel and steam, temperatures of the mixture inside the vaporiser and outlet temperatures. Due to the high residence times and temperatures applied, full evaporation can be achieved in the vaporiser. This is substantiated by the fact that all outlet temperatures are above the point at which the fuel vapour reaches a superheated state.

When hydrocarbons are heated in absence of oxygen, chemical reactions known as cracking or pyrolysis take place. In order to investigate the amount of thermal cracking reactions

during fuel preparation, the fuel vapour stream leaving the vaporiser is passed through a water cooled condenser. With cooling water at 14°C the large molecules recondense, whereas lighter hydrocarbons formed during the cracking process remain gaseous. By measuring the amount of condensate produced it is possible to estimate the proportion of thermal cracking. The gaseous cracking products constitute a complex mixture of low molecular weight hydrocarbons such as methane, ethane, ethylene and propylene. As a first approach, the thermal cracking of hydrocarbons can be described using a conversion ratio U, which is defined as the ratio of the mass flow rate of non-condensed gaseous fuel at the condenser exit to the mass flow rate of fuel entering the vaporiser. In Figure 10 results are shown for the cracking conversion ratio of Oil No.2 at vaporiser outlet temperatures of 530°C to 580°C. Based on chemical kinetics for endothermic cracking reactions with high activation energy, it can be shown that temperature is the dominant parameter governing reaction rates; as temperature increases so does the conversion ratio - this is confirmed by the results in figure 10. Furthermore, an influence of residence time on this ratio can be observed. In practice however, some of the cracked fuel components are also collected in the condensate, which leads to overall conversion ratios much higher than those actually measured.

Combustion

The applicability of oil vapour for premixed combustion with an ABB EV burner was confirmed with a number of tests. A smooth and uninterrupted transition was also possible between operation with normal gaseous fuel and liquid fuel vapour. During testing the oil vapour flame was stable over a wide range of fuel/air ratios, resulting in lean blowout (LHC > 1 vppm) at $\lambda > 3.5$. The fuels burnt with a non luminous (blue) flame downstream of the combustor.

The NO_x emissions for operation at atmospheric pressure are presented as a function of adiabatic flame temperatures in Figure 11. The burner thermal loading ranged from 80 to 150 kW for a steam/fuel ratio of 0.4. Due to chemical kinetics, NO_x emissions decrease with lower adiabatic flame temperature. In the low flame temperature range, thermal NO_x formation plays an unimportant role in NO_x emissions. Very low NO_x values of 0.5 vppm (15% O₂) were obtained for natural gas at both loads. The combustion of oil vapour resulted in NO_x values of 4 to 4.5 vppm (15% O₂) at the same flame temperatures. This is due to the contribution of fuel bound nitrogen, quoted as 124ppm from the fuel analysis. If the entire FBN were converted to NO_x, 5.2 vppm NO_x would be generated. By comparing with test results, it is hence estimated that the conversion ratio of FBN to NO_x is approximately 70%, assuming that thermal NO_x formation for combustion of oil vapour and for natural gas are equal.

Part load combustion was also investigated for both fuels with adiabatic flame temperatures varying between 1600K and 1900K. For operation at low burner loads (85 kW for natural gas or 100 kW for oil vapour) high NO_x emissions were observed,

whereas at full load (150 kW) the measured values were noticeably lower. This phenomena is understood by the fact that at part load the air inlet velocity is too low to achieve perfect premixing of fuel vapour and air. Nonetheless, NO_x emissions remained below 20 vppm (15% O_2), even at high flame temperatures (1900K).

The data plotted in Figure 12 show that NO_x emissions are almost independent of water/fuel ratio in the range $0.4 < \Omega < 0.9$.

Figure 13 shows CO emissions as a function of adiabatic flame temperature for oil vapour and for natural gas. Although there is some data scatter, measurements indicate similar CO emissions for both fuels. CO concentration is a minimum at a flame temperature of 1500K. Above a flame temperature of 1600K there is a rise in CO emissions due to CO_2 dissociation. For very lean combustion near blowout, (1300K-1400K) a sharp increase in CO can again be observed, since the residence time is too short for complete CO oxidation at these low flame temperatures.

The UHC emissions generally follow the CO emissions. Observed concentrations of UHC were less than 1vppm under typical combustion conditions ($T_{ad} > 1500\text{K}$). The results of UHC measurements also represent an indicator for combustion efficiency. In this paper, the flame lean-blowout is defined as the point at which UHC emission exceed 1vppm.

Due to the complete conversion of the liquid fuel to a gaseous phase, no carbon deposits on the wall of the burner or soot in the exhaust could be detected.

Results from the high pressure tests showed that there was no perceptible pressure dependency on NO_x formation for combustion pressures up to 23 bar. NO_x generation at high pressure was practically identical to the formation rates measured under atmospheric conditions. The trends at high pressures were also similar; in particular the fact that thermal NO_x formation was only significant at adiabatic combustion temperatures above approximately 1600K (characterised by a steeper curve, Figure 14).

A variation of the steam concentration in the fuel was conducted at combustion pressures of 5 and 21bar. For $0.4 < \Omega < 1.2$ a slight effect on NO_x emissions was detected, namely a reduction of approximately 15% in measured NO_x values over this range. The relatively small variation is accountable to two characteristics, the first being that the high temperature steam does not produce as an appreciable reduction in peak flame temperature as would be the case for injection of cooler water through a conventional nozzle. A second point is the fact that for a VPL combustion process an optimisation of the mixture homogeneity has already been achieved and it is therefore unlikely that the NO_x generation rate can be influenced by water addition, since the fuel is already well premixed with the combustion air. Conversely, for droplet combustion in liquid fuel injection processes the fuel equivalence ratio varies considerably in space and time; i.e. a fuel rich region surrounds each droplet

during combustion which causes higher local flame temperatures and hence an increase in NO_x emissions.

A radial profile was taken using a traversable probe mounted perpendicularly to the combustion chamber axis. The results showed no detectable variation in exhaust gas concentrations (in particular, low UHC emissions) across the combustion chamber, and hence it was concluded that complete flame burnout had occurred. In this case the residence time between flame stabilisation and probe was approximately 50ms for this profile.

CONCLUSIONS

Perfect premixed lean combustion without flame instability or flash back has been achieved for the case of a prevaporised liquid fuel employed in a gas burner. Furthermore, it is possible to switch from gaseous to prevaporised fuel without interruption.

Conversion of liquid fuel to a gaseous state and burning using premixing lean combustion technology results in low pollutant emissions, namely NO_x below 20vppm and CO less than 10 vppm (corrected to 15% O_2) under gas turbine operating conditions. The difference in NO_x emissions between combustion of natural gas and vaporised Oil No.2 originates mainly from the fuel bound nitrogen content of the liquid fuel.

Steam addition in the system lowers the vapour pressure of the hydrocarbons and therefore enhances their evaporation. Nonetheless, steam addition to the fuel mass has little appreciable effect on the NO_x formation rate for VPL combustion. Reforming reactions between hydrocarbons and water change the composition of liquid fuel vapour and hence prevent formation of carbon deposits on the vaporiser wall.

Little or no pressure dependency on the NO_x formation rate has been observed for VPL combustion at pressures up to 23 bar. This result is in accordance with similar observations made for perfect premixed lean combustion under pressure (G.Leonard and J.Stegmaier, 1993) and shows that it is possible to design high pressure combustors up to 30bar with NO_x emissions comparable to atmospheric versions.

In addition the VPL combustion concept can be advantageously exploited to improve the thermal efficiency of gas turbine processes. It can be applied as an alternative fuel option, running without modification to the burner already in place for gas combustion.

ACKNOWLEDGEMENTS

The authors would like to thank the „Nationaler Energie-Forschungs-Fonds“, Basel (Switzerland) as well as ABB Corporate Research for their continual support.

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FIGURES

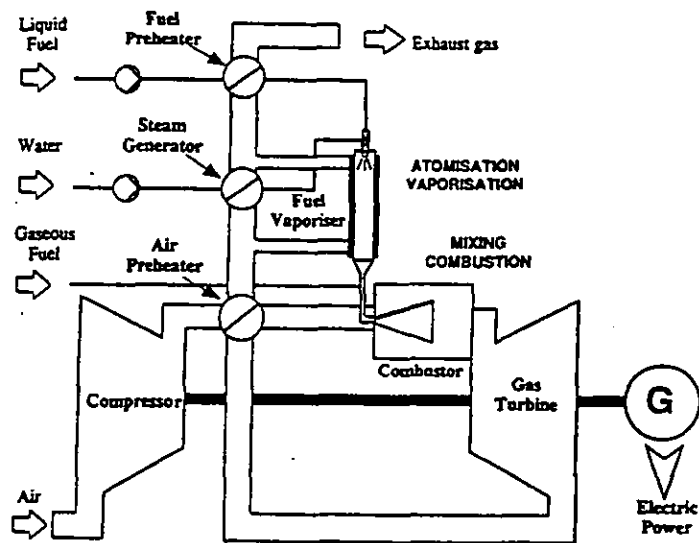


FIGURE 1 VPL COMBUSTION CONCEPT

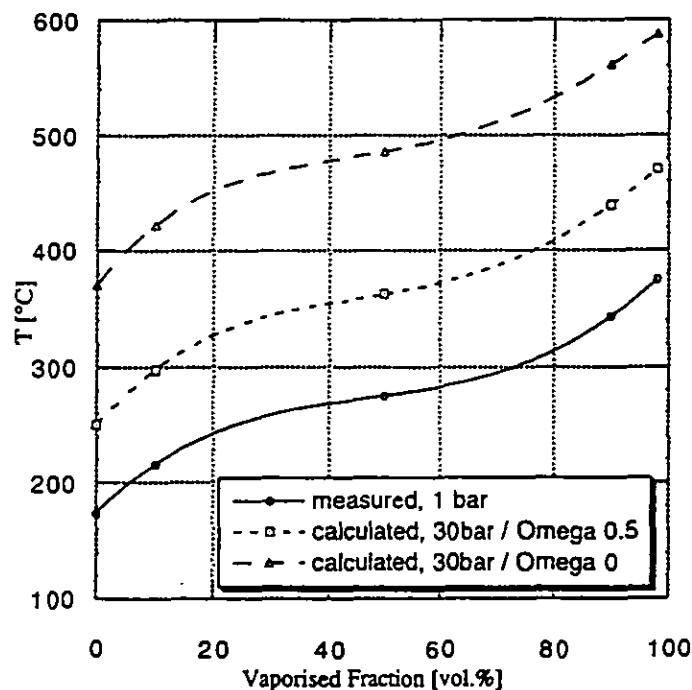


FIGURE 2 MEASURED AND CALCULATED BOILING RANGE OF OIL NO. 2

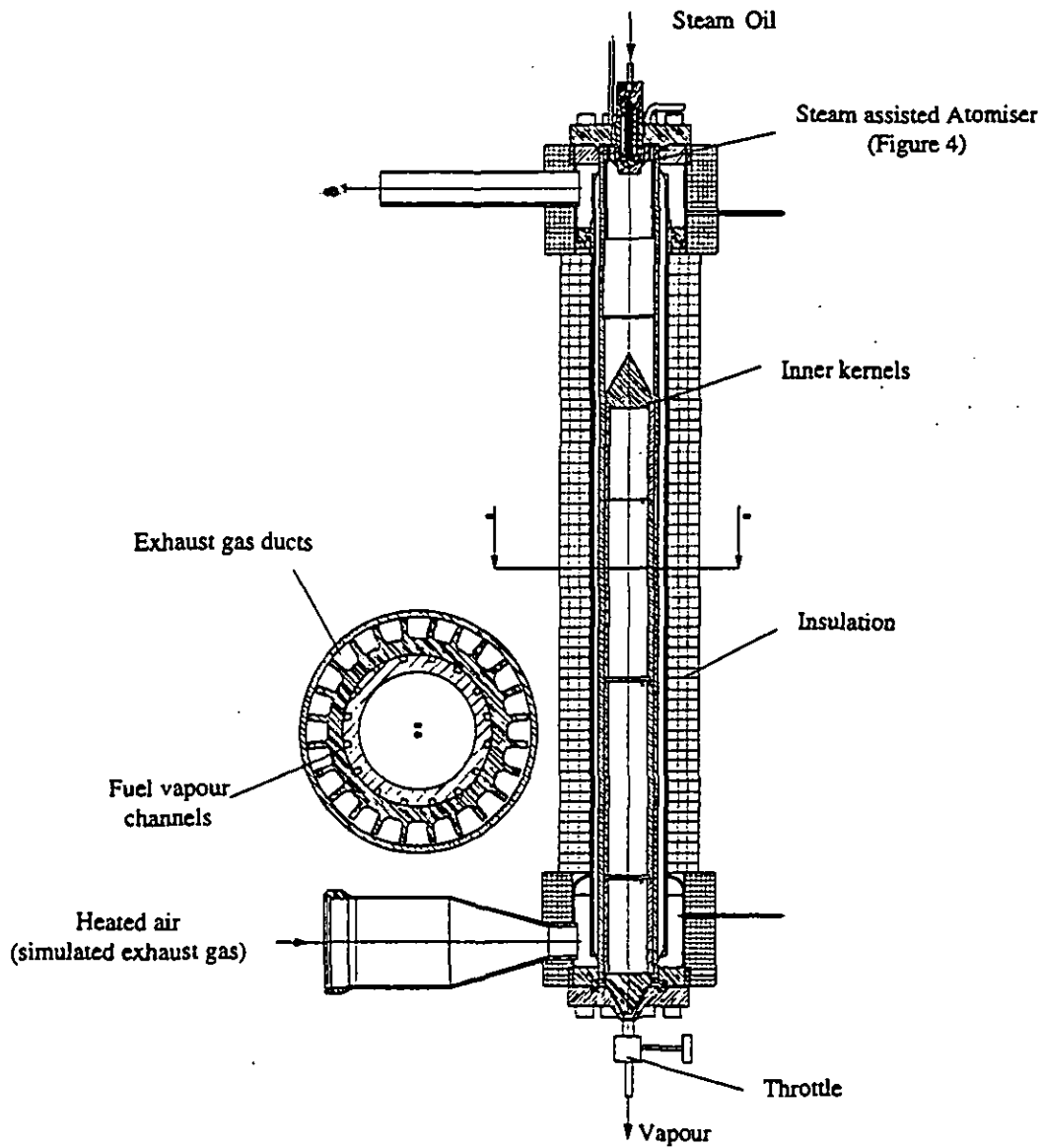


FIGURE 3 VAPORISER

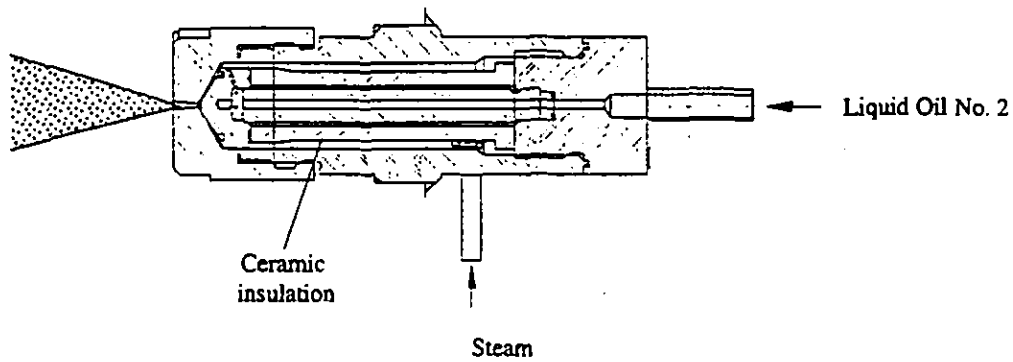


FIGURE 4 STEAM ASSISTED ATOMISER

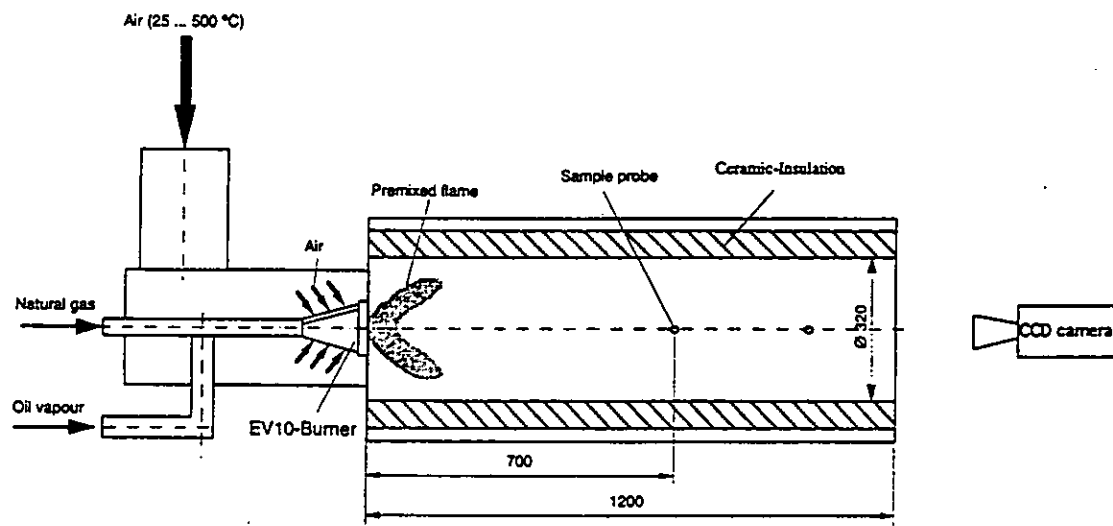


FIGURE 5 ATMOSPHERIC TEST RIG

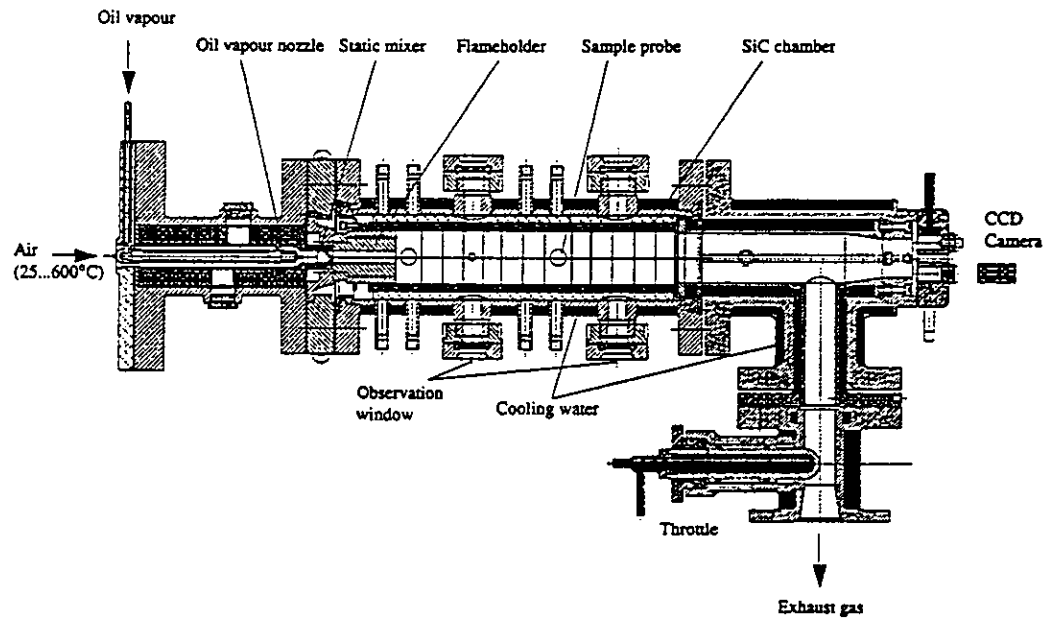


FIGURE 6 HIGH PRESSURE TEST RIG

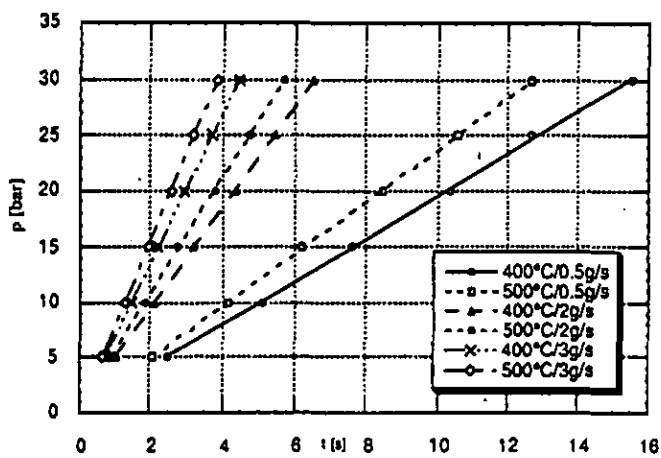


FIGURE 7 APPROXIMATE VAPOUR RESIDENCE TIME IN VAPORISER AS A FUNCTION OF OPERATING PRESSURE AND TEMPERATURE FOR $m = 0.5$ & 3.0 g/s

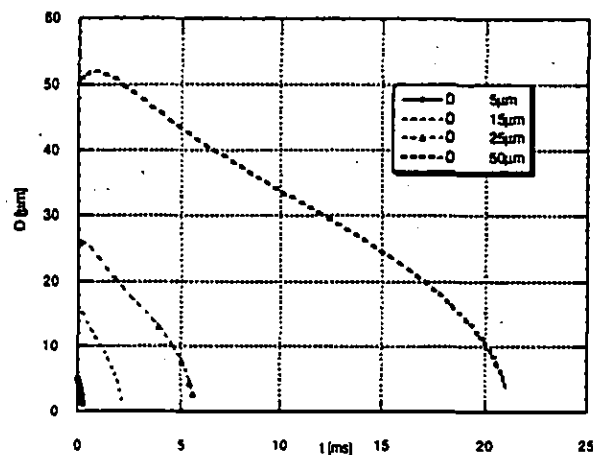


FIGURE 8 SIMULATED EVAPORATION OF DROPLETS OF HEXADECAN IN STEAM 350°C AND 30BAR

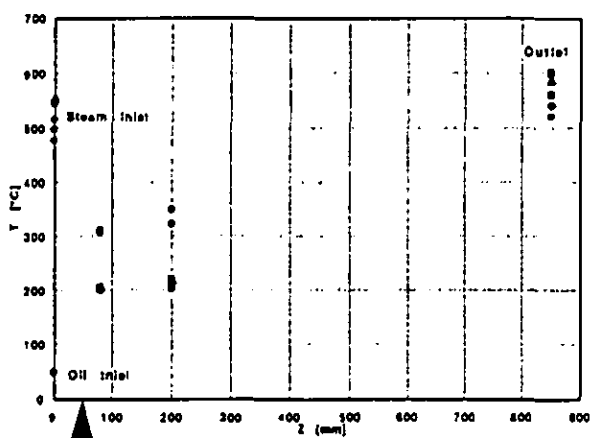


FIGURE 9 TYPICAL TEMPERATURES AT DIFFERENT LOCATIONS IN VAPORISER (Z: DISTANCE ALONG VAPORISER AXIS)

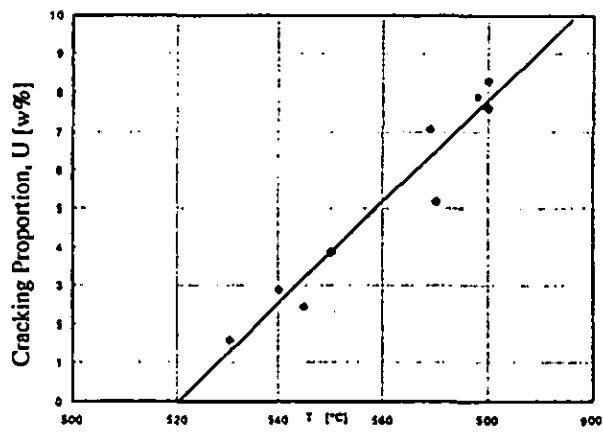


FIGURE 10 MEASURED THERMAL CRACKING RATE OF OIL NO.2 (T: OUTLET TEMPERATURE OF VAPORISER)

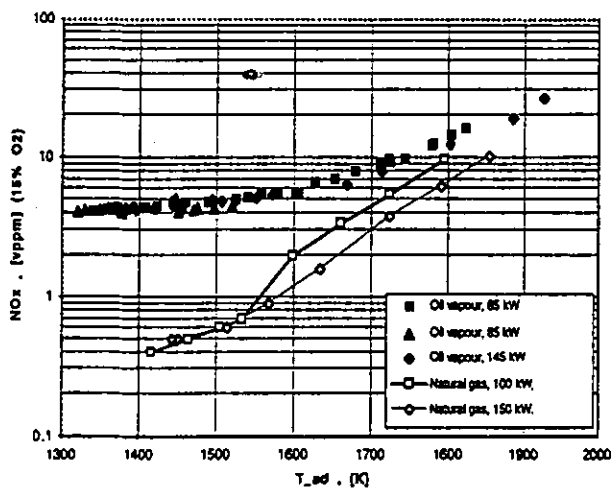


FIGURE 11 NO_x EMISSIONS FOR COMBUSTION OF OIL VAPOUR AND NATURAL GAS AS FUNCTION OF ADIABATIC FLAME TEMPERATURE UNDER ATMOSPHERIC PRESSURE

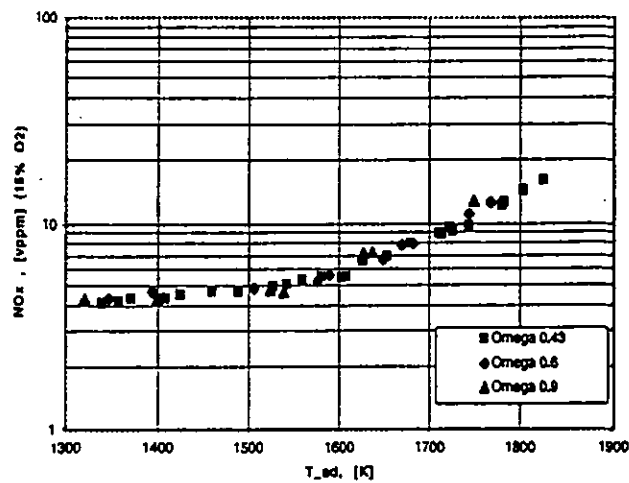


FIGURE 12 NO_x EMISSIONS FOR COMBUSTION OF OIL VAPOUR AT VARYING STEAM ADDITIONS AS FUNCTION OF ADIABATIC FLAME TEMPERATURE UNDER ATMOSPHERIC PRESSURE

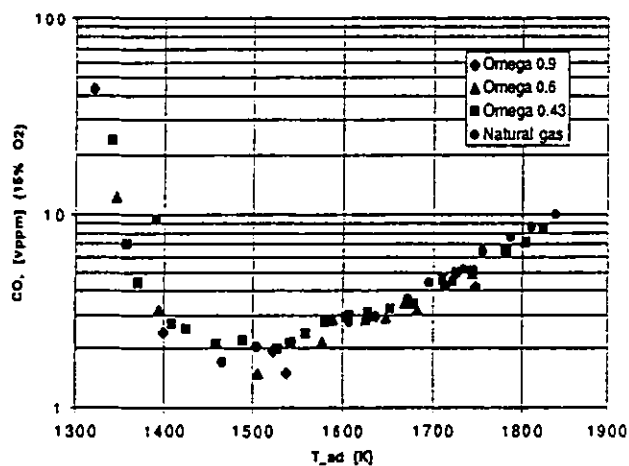


FIGURE 13 CO EMISSIONS FOR COMBUSTION OF OIL VAPOUR AND NATURAL GAS AS FUNCTION OF ADIABATIC FLAME TEMPERATURE UNDER ATMOSPHERIC PRESSURE

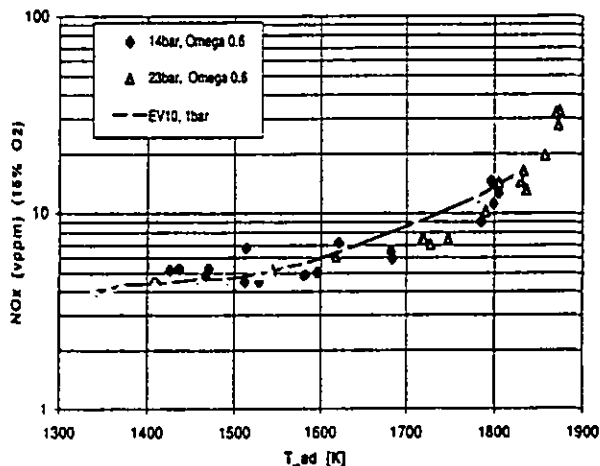


FIGURE 14 NO_x EMISSIONS FOR COMBUSTION OF OIL VAPOUR AND NATURAL GAS AS FUNCTION OF ADIABATIC FLAME TEMPERATURE UNDER HIGH PRESSURES