

**OPTIMIZATION OF COMBUSTION GENERATED PARTICULATES  
IN A RESIDUAL FUEL OIL-FIRED FURNACE**

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**SYNOPSIS**

This paper is concerned with the experimental work carried out in a horizontal tunnel type furnace to study the optimization of production of particulates during the process of combustion and to find out whether or not this optimization is achieved at the expense of thermal efficiency of the furnace. The furnace was fired with residual fuel oil of viscosity 3500 seconds Redwood 1 using industrial burners. Thermal input to the furnace was varied between 3MW and 4MW.

During the plant trial, the influence of combustion air preheat, quantity of atomizing steam, swirl in the combustion air and oil-water emulsion firing on combustion generated particulates and thermal efficiency of the furnace was studied. The test results showed that the operating parameters and added water in the emulsion fuel played an important role in the formation of particulates and heat transfer to the furnace cooling load.

## 1. INTRODUCTION

Burner manufacturers are faced with the problem of manufacturing industrial burners to meet the requirements of a low level of production of particulates and a high thermal efficiency. In order to meet these dual requirements, the combustion equipment must be capable of burning fuel close to stoichiometric conditions before particulates are formed.

In an oil-fired combustion system, liquid fuel is injected into the combustion chamber or furnace in a spectrum of minute droplets so that the surface area of the oil exposed to the combustion air is greatly increased. The oil droplets come in contact with the combustion air and recirculated hot combustion products. The droplets receive heat from the hot gases by convection and heat from the flame and hot walls by radiation. The fuel vapour resulting from the evaporation of droplets mixes with the air and combustion products to form an inflammable mixture. The ignition of the inflammable mixture in the vicinity of the droplets increases the evaporation rate of the droplet due to heat transfer from the flame.

During the process of combustion of an oil droplet, swelling followed by contraction takes place resulting in some instances in the formation of soot and coke particles. Soot is formed from vapour phase cracking after the initial evaporation from the liquid drops and is made up of individual particles well below 1mm diameter<sup>(1)</sup>. Coke particles are formed from liquid phase cracking of heavy oil droplets, and they are mostly in the size range of 5-100mm<sup>(2)</sup>. Soot is necessary for luminous flames and hence, high heat transfer rates. In an oil-fired boiler, if the soot is not all burned, it can cause air-heater fires due to its high reactivity, and it also contributes to visible smoke at the chimney<sup>(3)</sup>. On the other hand, coke or grit particle contributes little or nothing to the flame luminosity or visible smoke, but it is emitted from the chimney in the form of acid smuts.

The most important factors controlling coke formation are the fineness of atomization of oil, flame temperature and air/fuel ratio. Generally, the higher the distillation range of the oil the higher will be its coking tendency but the cost of the oil is lower.

Particulates in flue gases consist of carbon, ash and unburned solids. During start-up of a cold furnace, or in low excess air firing, the formation of carbon is almost inevitable. The unburnt carbon can deposit on low temperature surfaces of the combustion system, and if the back end temperature of the system is below the acid dew point, condensation of sulphuric acid, due to  $\text{SO}_3$  in flue gases, results. This sulphuric acid will be absorbed by any cool deposits. Due to the gas velocity, these acid contaminated carbon deposits may flake away to produce the undesirable phenomenon of acid smuts. Also the unburnt carbon that settles on cool surfaces of the combustion system or leaves the chimney constitutes a combustion loss<sup>(4)</sup> and thus, reduces the thermal efficiency of the furnace or boiler.

This paper deals with the experimental work carried out in an oil-fired furnace using industrial burners to study the influence of operating parameters and oil-water emulsion firing on optimization of combustion generated particulates and to determine the effect of this optimization on thermal efficiency of the furnace.

## **EXPERIMENTAL FACILITY**

### **Experimental Furnace**

The furnace is a horizontal tunnel type, the walls and roof of which are lined with refractory bricks. Seventeen cooling plates formed the floor of the furnace. It is approximately square in cross-section, 2 metres x 2 metres, with an arch roof and has a length of 6.25 metres. The construction of the furnace is shown in Figure 1.

The burners deliver the fuel and air into the furnace and the fuel is fired axially along the furnace length, exhausting through a chimney at the far end of the furnace. The furnace walls are constructed of 450 mm. thickness of refractory materials and consequently, the wall heat losses are kept to a minimum.

There are a number of slots through the furnace walls for gaining access to the flame for measurement purposes. A single horizontal slot, which serves the first two metres of the furnace, provides access to all axial stations in the region close to the burner. Subsequently, the vertical access ports are spaced every 0.75 metre. When the access ports are not in use, each port can be closed by means of a vertically sliding water cooled door. Thus it is possible to reduce gas leakage to a minimum whilst carrying out measurements. The furnace can be maintained at a positive pressure by means of a manually operated chimney damper to prevent air leakage.

### **Test Burners**

#### **A. Industrial Burner A**

The main features of this burner are shown in Figure 2.

It is a fixed thermal release baffle burner, normally used in reheating furnaces, boilers and recirculating furnaces, suitable for operating furnace temperatures up to 1450 °C.

The burner uses the principle of refractory flame stabilization, that is the stabilizing influence of the hot refractory burner quarl, employing a concentric row of air holes surrounding a central fuel nozzle. A jet at high velocity, about 40 m/s,

sets up a recirculating zone around the jet and creates a suction at the face of the baffle to stabilize the flame and to promote the mixing of the fuel and air. The baffle design is tailored to suit each application and the air velocity through the baffle holes determines the flame shape. Fuel oil atomization is achieved by using steam or compressed air. The burner design is such that the burner imparts a constant swirl to the combustion air for various throughputs.

## **B. Industrial Burner B**

Figure 3 illustrates the main principles of this burner. This is a variable swirl burner. The air to the burner is supplied through two ducts, the primary and the secondary ducts which are connected to each other.

Adjustable swirl vanes are mounted in the secondary air duct and the angle of the swirl vanes controls the tangential momentum of the secondary air and, at the same time, controls the ratio of primary to secondary air distribution as the two ducts are fed via a common wind box. Atomization of the fuel is achieved by either pressure-jet or steam.

### Swirl Output of Burner B

The measurements of the various swirl outputs of the burner were carried out using a swirl-meter. The meter was bolted to the burner and the swirling flow, corresponding to the various levels of swirl-setting of the burner, was directed into the narrow axial channels of the swirl-meter, the flow finally issuing in the radial direction. The idea is to remove all swirl from the jet on passage through the swirl-meter, thus giving rise to a couple on the meter equal to the angular momentum flux of the flow which is measured by the angular deflection of the apparatus, knowing the mass and radius of the counter balancing weight.

The swirl-meter gives

$$G_t = GL_1(\sin q) g, \text{ where}$$

$$G_t = \text{Angular momentum flux of the flow (kg.m}^2\text{/s}^2\text{)}$$

$$G = \text{Counter balancing wt (kg)}$$

$$L_1 = \text{Arm of the couple (m)}$$

$$q = \text{Angular deflection of the apparatus (degrees)}$$

$$g = \text{Gravitational constant (m/s}^2\text{)}$$

$$S = G_t/(G_x R), \text{ where}$$

$$S = \text{Swirl number (dimensionless)}$$

$$R = \text{Radius of the burner (m)}$$

$$G_x = \text{Linear momentum of the flow (kg m/s}^2\text{)}$$

$$G_x = MaVa, \text{ where}$$

$$Ma = \text{Mass flow rate of air (kg./s)}$$

$$Va = \text{Air velocity (m/s)}$$

## **METHODS OF MEASUREMENTS**

The operating conditions of the furnace were monitored continuously. They were largely controlled automatically from a control room where the recording instruments and a number of pen-recorders for use with the measuring probes were located.

### **Particulate Measurements**

Flue gas samples were drawn from points in the chimney for measurements of particulate concentrations.

A Beta-dustmeter was used for the particulate measurements during the testing of the two industrial burners. The dust-meter measured the concentration of particles in  $\text{mg./m}^3$  of wet gas. To find the concentration of particles, the volume of gas sample was measured and the mass of particles collected was determined. The quantity of particles was determined by measuring the attenuation of beta rays in the collected particle sample. The sample volume was measured by means of a venturi tube connected to a differential pressure controller.

The dust-meter, initially, gauges the mass of the filter at a particular spot. That value is considered to be zero. The measurement cycle is then initiated by the filter housing opening and the filter tape moving between the beta radiation source and a tube. A motor driven tape drive pulls the tape to the exact predetermined location for sampling. After locating a particular spot, the beta gauge measures the attenuation of the spot and stores the impulses collected over the selected time interval. These impulses act on a digital computer and are passed on to a precision potentiometer by way of an impulse-regulated step motor. The stretch of filter is transported into the filter housing by the back pulse motor so that the same spot that was zeroed is now ready to collect the sample.

During the sampling cycle, the gas flow is maintained constant by the differential pressure controller connected to the venturi tube and the vacuum pump by-pass valve. The built-in maximum and minimum contact switches in the differential pressure controller are adjustable for maintaining flow at various flow rates. The dust-covered spot of the filter is moved forward into the beta gauge by the measuring roller. Rays from the radioactive source pass through the entire dust-covered area so that any unevenness in the distribution of the dust is accounted for. By comparing the ratio of the signals from the zero rate and the dust attenuated rate, errors due to variation in weight in the filter material, fluctuations in temperature and other interferences are eliminated.

During the measuring cycle, the following readings were taken:-

- (i) Overpressure reading,  $P_0$  (mm Hg) taken at the end of the suction time .
- (ii) Rotameter reading,  $R_a$  ( $\text{Nm}^3/\text{hr}$ ) taken during the suction time.
- (iii) Temperature of venturi reading,  $T$  ( $^{\circ}\text{C}$ ) recorded during the suction time .
- (iv) Sampling time,  $t$  (minutes).

The rotameter was calibrated so that 100 divisions on the rotameter =  $y$  ( $\text{Nm}^3/\text{hr}$ ) of the gas flow and therefore, the rotameter reading,

$$R_a = (y/100) R_a \text{ (Nm}^3/\text{hr.) of the gas flow.}$$

The flow was then corrected for the overpressure and temperature. The correction factor ( $w$ ) was obtained from a correction graph and the total gas flow ( $z$ ) during the sampling time was calculated as follows:-

$$z = (y/100)(R_a t w/60) \text{ (Nm}^3\text{)}$$

On a multiple-point recorder chart, a signal (in mV) was recorded. The value of the recorder signal was taken and the corresponding value of the weight of the particulates(  $W$  in mg. ) was obtained from a table.

The content of the particulates of the flue gas is thus =  $W/z$  ( $\text{mg./Nm}^3$ )

#### Furnace Heat Balance

Heat was removed from the furnace in the following ways: Chimney losses, water cooling to the access doors, cooling plates on the furnace floor and heat conduction through the furnace walls. Rotameters and thermocouples were used for measuring the cooling water flow rates and the temperature rise of the water through the system respectively. For steam atomized burners, the furnace heat balance is expressed as:

$$\begin{array}{ccccccc} \text{Chemical and} & \text{Heat in steam} & & \text{Heat in} & \text{Heat} & & \text{Heat conducted} \\ \text{sensible heat} & + \text{ used for atomising} & = & \text{chimney} & + \text{ absorbed by} & + & \text{by refractory} \\ \text{of input fuel} & \text{liquid fuels} & & \text{gases} & \text{cooling water} & & \text{walls} \end{array}$$

In the case of pressure jet burners, the heat balance is :

$$\text{Chemical and sensible heat of input fuel} = \text{Heat in chimney gases} + \text{Heat absorbed by cooling water} + \text{Heat conducted by refractory walls}$$

$$\text{Thermal efficiency of furnace} = \frac{\text{Heat transferred to furnace cooling load}}{\text{Thermal input}}$$

### Air Preheat

Combustion air was preheated in an air-preheater fired by natural gas and temperature measurements of the preheated air were taken using thermocouples.

### Oil System

Fuel oil from the tanks was drawn and heated to the required temperature by means of a heater using steam and pumped to the burner. The oil flow rate was measured by means of an orifice working on a recording controller

#### A. Emulsifier

In the case of oil-water emulsion as a fuel, water was added to the oil and the mixture was homogenised in a simple vibrating blade emulsifier.

#### B. Fuel Oil Analysis

Residual fuel oil of 3500 seconds Redwood 1 viscosity was used for the experimental work. Its composition was as follows:-

Carbon, %	86.29
Hydrogen, %	11.17
Oxygen, %	0.71
Sulphur, %	1.81
Density, @ 15°C	0.967
Asphaltenes, %w	1.29
Higher Calorific Value, kcal/kg.	10,300

### RESULTS AND DISCUSSION

The experimental results obtained on combustion generated particulates and thermal efficiency of the furnace are shown in Figures 4, 5, 6 and 7.

### **Influence of Atomizing Steam**

Figure 4 shows that the particulate emission dropped when the quantity of atomizing steam was increased from 10% to 15% of the oil flow rate, but a further increase in atomizing steam, however, increased the particulate burden of the flue gas at combustion air preheat of 20 °C with a loss of 2.8% thermal efficiency. As evident from figure 5, the particulate content of the flue gas decreased with increasing quantity of atomizing steam at air preheat of 400 °C at the expense of a drop in thermal efficiency of 1.6%.

The reason for the drop of particulate production at the air preheat of 20°C when the quantity of atomizing steam was increased from 10% to 15% is probably due to improvement in the degree of atomization of the oil, and hence, in the reduction of the burning time of the oil droplets. A further increase of the atomizing steam flow to 23% of the oil flow rate at this air preheat has lowered the flame and combustion temperatures because a greater volume of gas had to be raised to the combustion temperature for no additional heat generation, resulting in an increased production of particulates.

At the air preheat of 400°C, the reduction in particulate burden of the flue gas with increasing atomizing steam quantity may be due to the combined effect of the higher air preheat temperature, better oil atomization, and hence, lower oil droplet burning times. All these factors may have contributed to the betterment of the combustion and off-set any adverse effect the increased quantity of atomizing steam may have had on the flame and combustion temperatures. The steam also contributes towards the reduction of solid production by increasing the OH concentration which decreases the atomizing fluid density which in turn lowers the drop diameter and thus, results in reduction of the total solid production<sup>(5)</sup>.

### **Influence of Swirl**

During the Burner "B" trial, the degree of swirl imparted to the combustion air was varied in order to study its effect on particulate emission and thermal efficiency of the furnace.

Figure 6 shows that in steam atomized oil combustion, there was a drop in particulate emission with increasing degree of swirl below  $S = 0.5$  with a loss of 1.1% thermal efficiency but, further increases in swirl however, increased the solid burden of the flue gas. The increased level of swirl in the combustion air, may have increased the mixing efficiency, and thus, improved the rate of combustion.



At higher swirl levels ( above  $S = 0.5$  ), the reduction of flame length may have caused some unburnt oil particles to escape from the flame, resulting in a higher level of particulates in the flue gas.

Drake and Hubbard<sup>(6,7)</sup> showed that there is an optimum value of swirl for a given combustion system. The influence of swirl on solid emission was shown to be as follows:

- i) Low values of swirl produced large quantities of carbon in the size range of 10-40mm.
- ii) Increasing the degree of swirl decreased the quantity of particles in these large size ranges.
- iii) Overswirling produced a large increase in the production of sub-micron soot.

#### **Influence of Added Water on Oil-Water Emulsion Firing**

Figure 7 shows that in the pressure-jet flame, the particulate emission decreased as the water content of the oil-water emulsion increased, the maximum increase in thermal efficiency of 1.8% attained with 10% emulsion water. The reduction of particulate emission and the increase of thermal efficiency may be due to a higher degree of oil atomization in the furnace caused by micro-explosions of the added water into the steam. These explosions may have caused a second-stage atomization of the oil particles.

#### **CONCLUSION**

In the steam atomized combustion system, the particulate emission decreased with increasing quantity of atomizing steam in the range of 15%-35%, expressed as % oil flow rate, at combustion air- preheat of 400°C with a drop in thermal efficiency of 1.6% or less, but at 20°C air preheat, the reduction of particulate emission was only associated with an increase in atomizing steam quantity from 10% to 15% with a loss of thermal efficiency of 2.8%.

Variation of swirl in the combustion air has an effect on the optimization of combustion generated particulates, an optimum level achieved in the weakly swirling range ( $S < 5$ ) with a loss of 1.1% thermal efficiency.

In the pressure-jet atomized water-in-oil emulsion fuel combustion, particulate emission levels dropped considerably with added water, an optimum value obtained with 10% emulsion water with an improvement of 1.8% thermal efficiency.

#### **ACKNOWLEDGEMENT**

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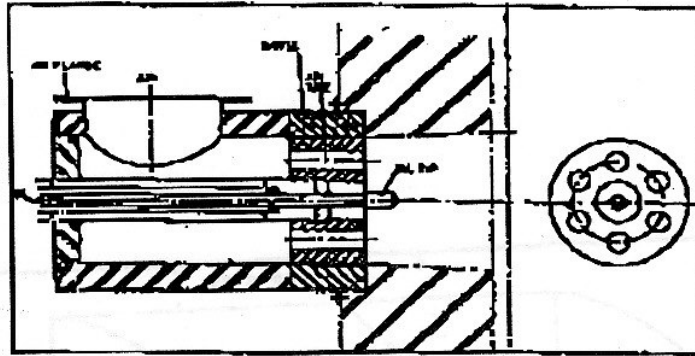


Figure 2: Schematic diagram of burner "A"

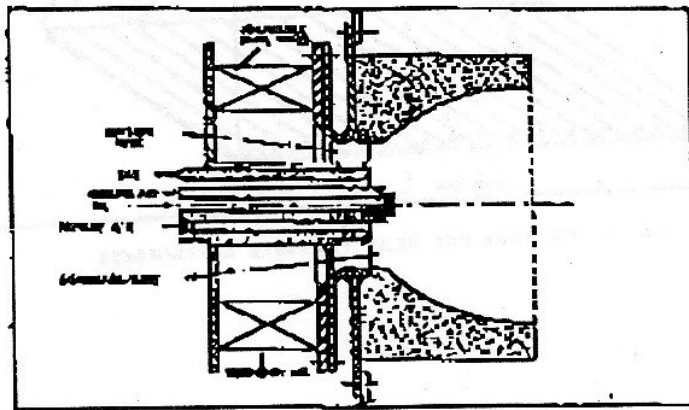


Figure 3: Schematic diagram of burner "B"

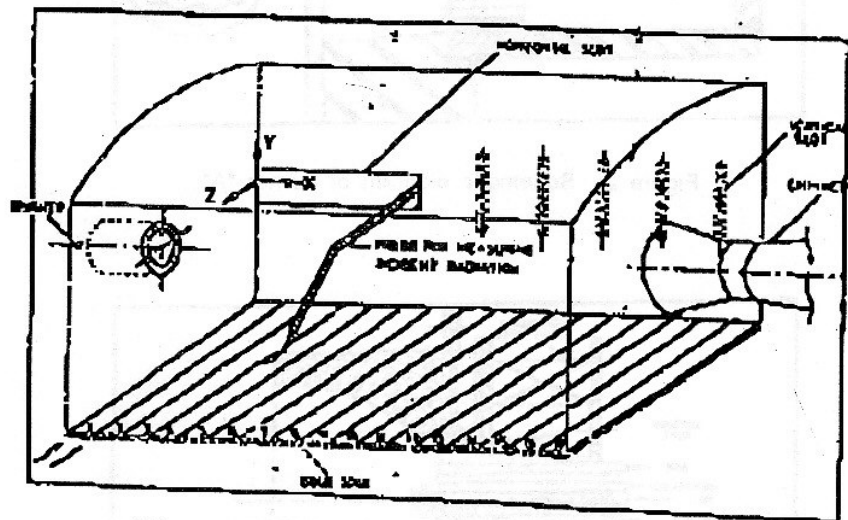


FIG. 3: FURNACES FOR HEAT TRANSFER EXPERIMENTS

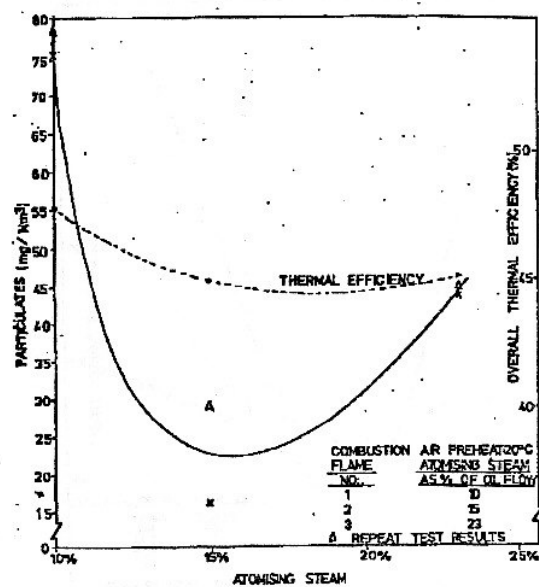


FIG. 4 BURNER A-EFFECT OF ATOMISING STEAM ON PARTICULATE EMISSION AND THERMAL EFFICIENCY.

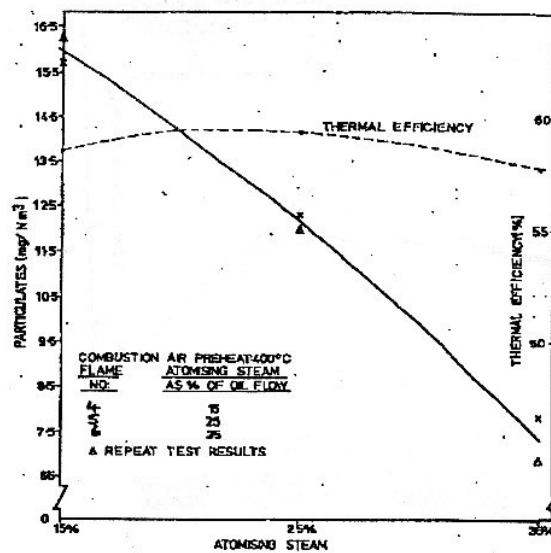


FIG. 5 BURNER A-INFLUENCE OF ATOMISING STEAM ON PARTICULATE EMISSION AND THERMAL EFFICIENCY (%)

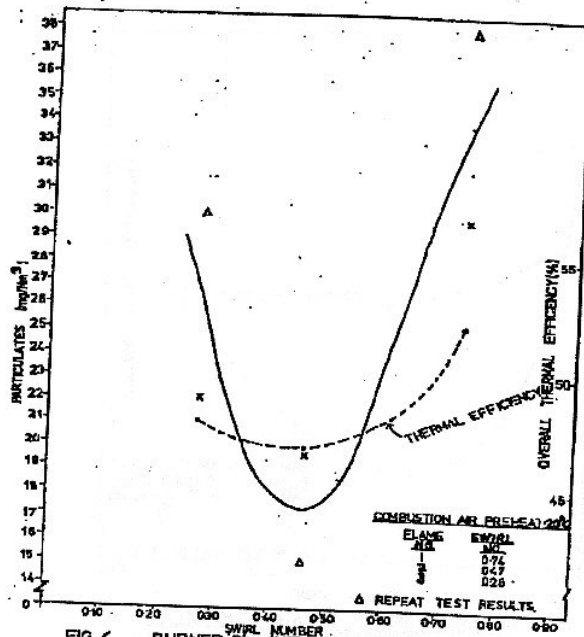


FIG. 6 BURNER B - INFLUENCE OF SWIRL ON PARTICULATE EMISSION AND THERMAL EFFICIENCY.

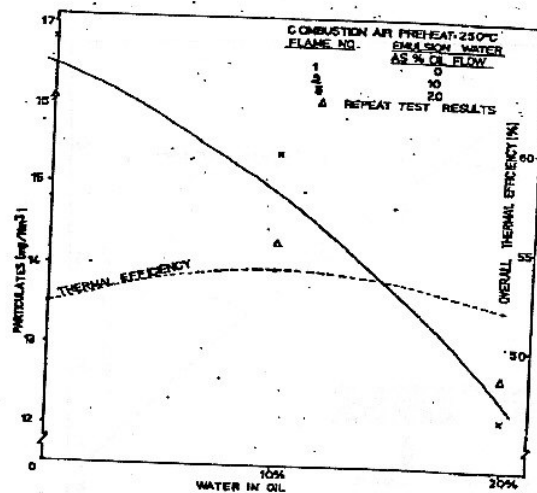


FIG. 7 BURNER B - EFFECT OF OIL-WATER EMULSION ON PARTICULATE EMISSION AND THERMAL EFFICIENCY.