NOTATION

B = dimensionless condensation driving potential, $C_p(T_s - T_\infty)/h_{fg}$
C_p = liquid specific heat, J/kg·K
D = injector diameter, m
G = vapor mass velocity, kg/m^2·s
h = heat transfer coefficient, J/m^2·s·K
h_{fg} = heat of condensation, J/kg
L = cavity length, m.
$\dot{m}$ = vapor mass flow rate, kg/s
P_c = chamber pressure, N/m^2
r' = cavity radius, m
R = rate of condensation, kg/m^2·s
S = dimensionless transport modulus, $h/G\,C_p$
T_s = saturation temperature, °K.
T_∞ = bath temperature, °K.
x' = axial distance, m.
x = dimensionless axial distance, $2x'/D$
X = dimensionless cavity length, $2L/D$

Subscripts

m = mean value for cavity
0 = injector exit conditions

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Ecological Aspects of Combustion Devices (with Reference to Hydrocarbon Flaring)

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Some fundamental factors controlling the emission of pollutants by an industrial combustion system are illustrated by reference to gaseous hydrocarbon flare stack combustion. The formation of smoke (soot), radiation, and nitric oxide may be controlled by limiting of the premixed fuel air ratio to moderately rich mixtures. The factors which determine the design of a suitable Coanda mixer are shown to be area ratio, density ratio, and pressure ratio. An important pollutant for large burners is the combustion roar. Fuel type and mixture ratio only affect the combustion noise output by about 5 db. The dominant factor in the generation of this noise is the burner turbulence, which can be controlled to reduce the combustion roar by up to 20 db.

Burner cost considerations lead to the current use of simple flare tips; however, the eventual use of more technically sophisticated low pollution units is inevitable.

Although most fuels will burn readily in air, when we wish to use the combustion process for some technological purpose the manner in which the fuel, air, and combustion products are brought together requires careful control. Thus, even the Pueblo Indians used an air deflector to control the air flow to the fire used to heat their dwellings. The next step taken early in the industrial revolution was the use of a secondary air supply above a burning solid fuel bed to consume combustible gases formed during pyrolysis of the fuel and thus achieve almost 100% combustion efficiency. Unfortunately, the overall process must be efficient, and the addition of excess air to assist in achieving complete combustion means the loss of heat due to the increased flow of hot gases through the flue. The optimization of the system therefore requires careful design of the combustor to achieve complete combustion with the minimum quantity of excess air.

Our concept of the overall system has now expanded to include the environment as well as the process, and the combustion designer must now reduce the emission of pollutants to an acceptable level. The major pollutants are: 1. particulate, for example, smoke; 2. gaseous, for example, carbon monoxide, oxides of nitrogen and aldehydes; 3. thermal and, 4. noise or combustion roar. In general, the emission of these pollutants does not significantly effect the efficiency of the process since they are mostly present in trace quantities. Further, the design of combustion systems has evolved as an art, rather than as a science, due largely to the complexities of the quantitative analysis of turbulent flow of dissimilar gases combined with the elaborate chain of chemical reactions involved in even simple combustors.

Fortunately, combustion research has progressed steadily and we are now in a position where fundamental combustion theory can be used in the design of combustion systems with improved ecological properties. In this presentation, the discussion will be confined to gaseous fuel although many of the concepts may be applied readily to liquid and solid fuels.

POLLUTANTS

Smoke

When the available oxygen in a hydrocarbon fuel/air mixture is progressively reduced, the combustion products change:

Mixture	Products
Stoichiometric	Carbon dioxide and water
Slightly rich	Carbon monoxide and water
Rich	Carbon monoxide and hydrogen
Very rich	Carbon, carbon monoxide, and hydrogen

In general accordance with this concept of relative affinity, it is found that for most gaseous premixed fuel/air systems carbon formation occurs when the oxygen to carbon weight ratio is less than $16/12 = 1.35$. This corresponds to equivalence ratios (ϕ) weaker than $\phi = 3$ (where ϕ = (actual fuel/air ratio)/(stoichiometric fuel/air) ratio). In practice, a safe limit is $\phi = 1.5$ (*1*); however, this depends on other factors such as the rate of secondary air addition.

The elimination of soot from the flame decreases the radiant heat transfer from about 40% of the total heat for a luminous flame to about 12% for a nonluminous flame.

Gaseous Pollutants

Carbon monoxide is formed when there is insufficient air in the reaction region to complete combustion.

Aldehydes are formed when the flame is quenched before the reaction is complete. Quenching may be due to the presence of cold walls, or mixing with excess cold air, or flame stretch effects.

Oxides of nitrogen are formed when the flame attains extremely high temperatures followed by relatively rapid cooling.

The carbon monoxide and aldehydes can be limited by correctly designing the burner aerodynamics and operating at the overall correct mixture ratio. These design factors will be discussed later.

Oxides of nitrogen present a difficult problem since higher temperature and intensity burners tend to produce higher concentrations of NO. If the burner can be operated on very weak mixtures $1/\phi > 1.4$ then Figure 1 shows that low NO formation is feasible. A slight decrease in $1/\phi$ from stoichiometric to about 1.1 results in an increase in NO since the temperature is still high and more oxygen is available to form oxygen atoms and hence NO. Unfortunately, operating the burner very weak results in a severe loss of heat in the flue gases and this technique must be confined to combustors such as gas turbine combustion cans where overall mixture ratios less stoichiometric are required by material limitations. In industrial combustion systems an alternative technique is to operate the burner with a non-adiabatic, staged flame. This is achieved by operating with a rich primary flame combined with radiant or convective heat transfer from the flame region. After some heat has been removed, more air can be added to complete combustion without the attainment of extremely high temperatures. A third technique to limit NO production is by the addition of an inert diluent to the air thus reducing flame temperature. In this case the 'cold' exhaust products from the flue are a convenient diluent, and about 15% recirculation can be employed effectively.

Thermal Pollution

In a large part of the Trent Valley in England, the weather has been very adversely affected by thermal pollution produced largely by power station cooling systems causing extensive fog formation. At present, an attractive solution appears to be the use of the low grade heat for district heating (central space heating) even if this is uneconomic. A heat exchanger in each building could be used with a ducted hot air central heating system, and arrangements could be made to dump the hot air to atmosphere when not required for heating. In this way, pollution from many local central heating furnaces could be reduced as hot air is the least objectionable form of thermal pollution.

Noise

Combustion noise takes two forms:

1. Periodic pressure oscillations, commonly referred to as combustion instability and associated with an acoustic mode of the combustor.

2. Random fluctuations due to the statistical nature of the combustion process known as *combustion roar*. The periodic oscillations are usually driven by changes in flow pattern, induced by the acoustic velocity component which cause periodic fluctuations in the heat release rate.

In the past, combustion noise or roar has received much less attention than combustion instability. However as the size of burners and their combustion intensity increases, this phenomenon is becoming a very real problem.

Noise at any distance from a source is measured as the sound pressure level (L_p) in decibels relative to 2×10^{-5} N/m^2. Since this noise level is a function of the distance from the source, it is convenient to convert to the actual sound power level (SPL) of the source measured in dbW

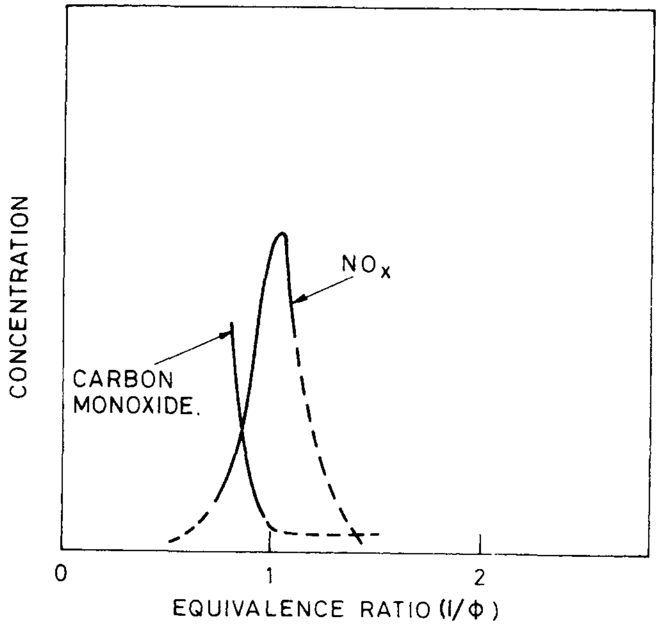

Fig. 1. Typical variation of oxides of nitrogen and carbon monoxide with equivalence ratio.

relative to 10^{-12} W. In the case of hemispherical radiation over a nonabsorbing surface, these two are related by

$$SPL(dbW) = L_p(db) + 10 \log_{10} 2\pi r^2 \quad (1)$$

where r is measured in meters. This equation is inaccurate for high frequency sound at distances greater than 50 meters due to atmospheric absorption (see the upper part of Figure 2).

The application of Equation (1) will be illustrated by an example: Suppose a burner is operating in the open air at 1 kg./sec. flow rate of fuel of calorific value 46,560 kJ/kg. and a noise efficiency of 10^{-7}. What is the sound pressure level at 10 meters?

Total burner output = 46,560 kJ/s (kW)

$\therefore$ Burner sound power output = 4.656 W.

$$\therefore \quad SPL(dbW) = 10 \log_{10} \frac{4.656}{10^{-12}}$$
$$= 10(0.668 + 12)$$
$$= 126.68 \text{ dbW}$$

$$\therefore \text{ At 10 meters } L_p(db) = 126.68 - 10 \log_{10} 200\pi$$
$$= 98.7 \text{ db}$$

Thus evaluating the noise level is straightforward once we know the noise efficiency, and Figure 2 presents both the sound power level and the distance attenuation effect for a typical hydrocarbon fuel.

To use Figure 2 one simply subtracts the db due to distance from the sound power level of the source corresponding to the appropriate fuel flow and noise efficiency. Similar curves can be drawn for more complicated environments or other fuel calorific values.

It now remains to evaluate the combustion noise generation efficiency.

Noise consists of pressure pulses and results from the unsteady nature of the combustion process, thus it depends on the *rate of change* of the rate of heat release rather than on the actual rate of heat release. Following an analysis first given by Bragg (2) and modified by Swithenbank (3) the efficiency can be evaluated as follows.

When a mass of fuel m burns with $1/\alpha$ times its mass of air, the products expand to x times their original volume, and a mass of gas m/α is displaced to occupy $1 - x$ times its original volume. Since the sound radiation depends on the rate of change of mass displacement the power radiated is

$$\dot{E} = [(x-1)\dot{m}/\alpha]^2/4\pi\rho c \quad (2)$$

where $\dot{m}$ is the rate of consumption of the fuel, ρ is the unburned density of the fuel-air mixture and c is the velocity of sound in the unburned gas. Now in a turbulent flame zone, we consider a small volume $\Delta V'$ of similar dimensions to the flame thickness in which the rate of combustion may vary from zero to the flame propagation speed. If the mean rate of fuel consumption per unit volume is $\dot{m}_f$, and the flame is convected by turbulence through the volume element at root mean square velocity u', then the mean rate of change of burning rate will be of the order $\pm \dot{m}_f \Delta V' u'/d_f$ (kg/sq. sec.) where d_f is the flame thickness.

Thus the mean square of the rate of change of combustion rate in the element is

$$\left(\frac{\dot{m}_f u' \Delta V'}{d_f}\right)^2 \quad (3)$$

Hence from Equation (2) we obtain

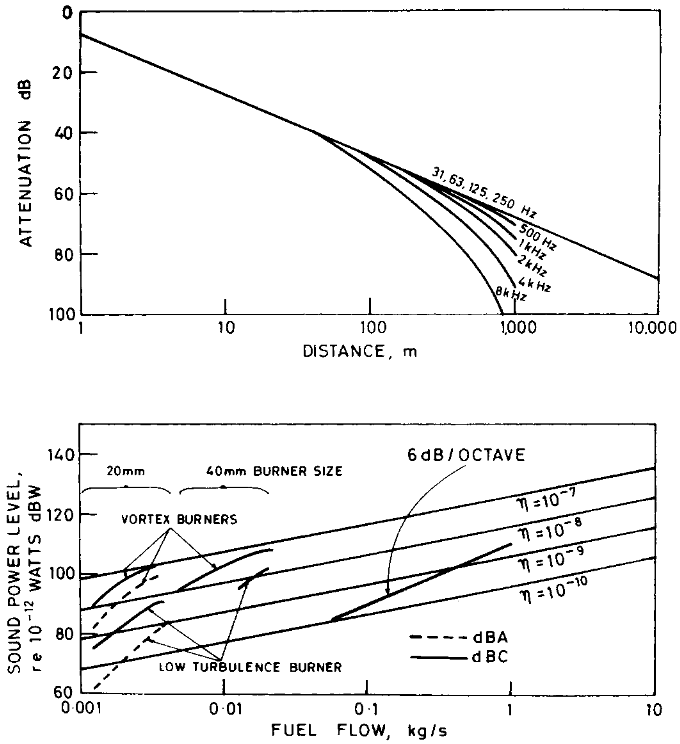

Fig. 2. Noise level of burners and attenuation.

$$\dot{E} = \left(\frac{x-1}{\alpha}\right)^2 \left(\frac{1}{4\pi\rho c}\right)\left(\frac{\dot{m}_f u' \Delta V'}{d_f}\right)^2 \quad (4)$$

The noise generation efficiency for a fuel of calorific value H is then

$$\eta = \frac{\dot{E}}{\Delta V' \dot{m} H} = \frac{(x-1)^2}{4\pi\alpha}\left(\frac{\dot{m}_f d_f}{\alpha\rho c}\right)\left(\frac{\Delta V'}{d_f^3}\right)\left(\frac{u'^2}{H}\right)$$

Now $\Delta V'/d_f^3 \doteq 1$, and this term can be neglected. $\dot{m}_f d_f/\alpha$ is the rate of consumption of air per unit area which is equal to ρS_L, therefore the second term is the dimensionless flame speed S_L/c and the equation becomes the product of three dimensionless groups

$$\eta = \left\{\frac{(x-1)^2}{4\pi\alpha}\right\}\left\{\frac{S_L}{c}\right\}\left\{\frac{u'^2}{H}\right\} \quad (5)$$

For a typical hydrocarbon fuel

$$x = 5 \quad a = 0.067 \quad S_L/c = 10^{-3}$$

$$u'^2/H = 10^{-5} \text{ to } 10^{-7} \text{ and hence } \eta = 10^{-7} \text{ to } 10^{-9}$$

For hydrocarbon fuels, the first term increases slightly with weaker mixtures while the second term decreases by a factor of four. As a result, a noise reduction of about 4 db is commonly observed from stoichiometric to the weak (or rich) limit. Changing the fuel type from methane to ethylene increases the flame speed by a factor of two (with little change in the other variables), giving about a 3 *db* increase in noise output.

The most sensitive factor controlling noise production is the dimensionless turbulence energy (u'^2/H). From the energy balance equation across the flame stabilizer

$$\frac{\Delta P}{q} = \zeta = 3\left(\frac{u'}{U}\right)^2_{\max} \quad (6)$$

where ΔP = total pressure loss across the stabilizer, q = dynamic head, that is, $\frac{1}{2}\rho U^2$, ζ = pressure loss factor, and

u'/U = turbulence intensity; we see that for any given stabilizer ζ is a constant and hence the burner turbulence intensity (u'/U) is a constant. It follows that if we double the throughput to the burner, we will double the value of U and hence also the turbulence velocity u'. The noise output would therefore increase by a factor of 4, that is, 6 *db*. The validity of this conclusion is illustrated on Figure 2 where it can be seen that for a variety of different combustors, the measured noise varied by 6 *db* per 'octave' (that is, doubling) of fuel flow.

In order to produce a low noise burner, we must modify the turbulence. Unfortunately if we reduce the turbulence we also reduce the mixing and the combustion intensity. The nearest approach which we can make is to transfer the turbulence energy to the wavenumber range of the dissipation eddies k_d (that is, we modify the turbulence energy spectrum so that $k_e \rightarrow k_d$). In this way, it is suggested, the mixing can be maintained at a finite level—thus avoiding a laminar flame, while the noise can be reduced to an acceptable level. In practice, we must modify the flow from the fuel/air mixer, which usually has a high turbulence level ($u'/U \doteq 50\%$) and $k_e \doteq R/10$ where R is the radius of the mixer exit/burner entry port. Reducing the turbulence to 10% would reduce the noise by a factor of 25, that is, 14 *db*. The use of a wiggle strip stabilizer (as discussed later) to modify the turbulence also modifies the size of the eddies by two or more orders of magnitude. This results in a corresponding change in the peak frequency of the noise, and the combustion roar which is usually obtained at a few hundred cycles frequency changes to a hiss which can be easily suppressed or deflected.

Some experimental results with low turbulence burners are plotted on Figure 2 which illustrates the validity of the above conclusions. For example the low turbulence burner is 10 *db* to 20 *db* quieter than a conventional burner at the same throughput.

APPLICATION OF FUNDAMENTAL CONCEPTS TO BURNER DESIGN

To illustrate the techniques whereby these fundamental concepts can be applied to the design of industrial combustion systems, the problem of flare tip design will be considered. The purpose of the study will be to demonstrate the significance of the various factors rather than produce an overall design which is commercially feasible. The flare stack combustor is used to dispose of waste gases from refineries and hydrocarbon plants, usually on an intermittent basis. In the event of a failure in some component in the process plant, relief valves pass the process flow, at rates up to several hundred tons/hr., to the flare stack for safe clean disposal. A major design feature is therefore a very large turn-down ratio.

Reference to the previous discussion on pollutants suggests that simply burning the gas in a diffusion flame with the air will at some conditions result in local carbon formation—particularly with unsaturated hydrocarbons—giving intense radiation and possibly smoke formation. Attempts to inject air at high velocity into the gas in order to reduce carbon formation will produce high turbulence levels and thus noise. If the peak temperature in the flame is very high, oxides of nitrogen will be formed, and the only convenient solution to this problem is to produce a non-adiabatic flame. In this case a luminous flame would help but is precluded on other grounds, hence we must use a flame of low combustion intensity. Fortunately this is compatible with a low noise flame as discussed above. The problem therefore consists of designing a combustion system which operates on rich premixed fuel:air mixture, having wide turndown characteristics, and low turbulence.

Air Entrainment

If the waste gas is available at high pressure (one or two atmospheres), then it contains enough energy to entrain the required premix air by means of the jet pump principle (momentum exchange)and give a small pressure at the burner tip. If the gas is only available at atmospheric pressure, then steam injectors are a convenient source of the required air. Steam is a more reliable source of the large quantity of energy required than electrically driven fans, especially in a plant emergency.

Although the performance of injectors (or ejectors) working over a significant delivery pressure ratio is well documented, the performance when the pressure rise of the air stream is negligible is less widely available. This entrainment ratio can be evaluated as follows.

Assuming approximately constant area mixing between fuel supplied through a small jet area A_j into a large air stream area A, velocity U_1, and passing to an exit at station 2. Then mass balance gives

$$\rho_1 \int U_1 \, dA + \rho_j V_j A_j - \rho_2 \int U_2 d(A_j + A) = 0 \qquad (7)$$

Momentum balance

$$\rho_1 \int U_1^2 \, dA + \rho_j V_j^2 A_j - \rho_2 \int U_2^2 d(A_j + A) = \int (p_2 - p_1) d(A_j - A) \qquad (8)$$

In these relations, it is assumed that the densities of the jet fluid and air may be different, and the velocity profiles of the external and exit streams may be nonuniform. If we further assume that the Bernoulli relation may be applied between air entry ($p_0 = p_2$) and station 1,

$$p_2 - p_1 = \tfrac{1}{2} \rho_1 U_1^2 \qquad (9)$$

Substituting Equation (9) in Equation (8) and taking $A/A_j >> 1$ gives

$$\rho_1 \int U_1^2 dA + \rho_j V_j^2 A_j - \rho_2 \int U_2^2 dA = \tfrac{1}{2} \rho_1 \int U_1^2 dA \qquad (10)$$

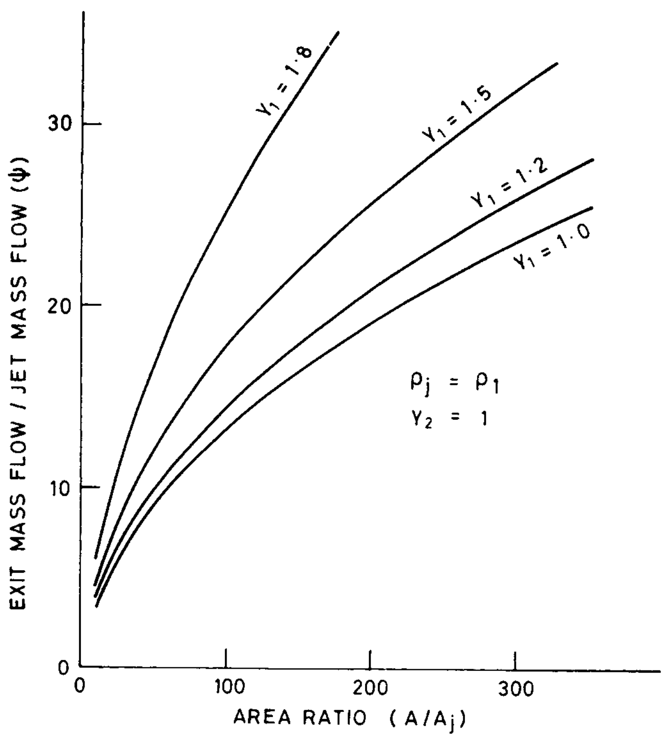

Fig. 3. Variation of entrainment with area ratio for various values of skewness factor Y_1.

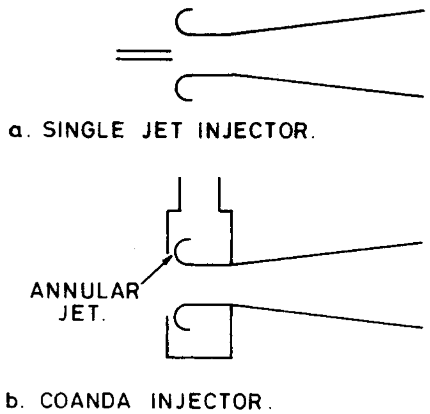

Fig. 4. Injector mixing configurations.

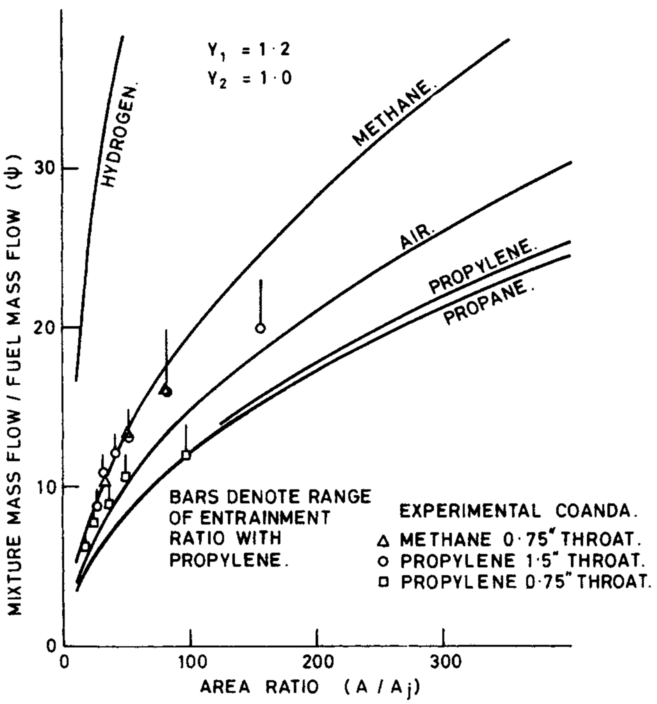

Fig. 5. Injector mass entrainment ratio (ψ) versus area ratio.

$$\therefore V_j = \left\{ \frac{1}{\rho_j A_j} \left(\rho_2 \int U_2^2 dA - \tfrac{1}{2}\, \rho_1 \int U_1^2 dA\right) \right\}^{1/2} \quad (11)$$

We now define a parameter Y which describes the degree of nonuniformity (that is, velocity profile) of the entry and exit streams.

$$\text{Taking } Y = \frac{\dfrac{\rho^2}{A} \int U^2\, dA}{\left(\dfrac{\rho}{A} \int U\, dA\right)^2} \quad (12)$$

we find that $Y \to 1$ for a flat profile

$Y = 1.5$ when velocity decreases linearly with radius,
$Y = 1.125$ when velocity increases linearly with radius,
$Y = 1.02$ when the velocity at the centerline is half the velocity at the wall (as at the exit of a Coanda injector), and
$Y = 2.0$ when there is no flow inside half radius and a constant velocity outside.

From the definition of Y we get $\int \rho U^2 dA = YW^2/\rho A$, and inserting this in Equation (11) gives

$$V_j = \left\{ \frac{1}{\rho_j A_j} \left(\frac{Y_2 W_2^2}{\rho_2 A_2} - \frac{Y_1 W_1^2}{2\rho_1 A} \right) \right\}^{1/2} \quad (13)$$

The mass entrainment ratio is then given by

$$\psi = \frac{W_2}{W_j} = \left\{ \rho_j A_j \left(\frac{Y_2}{\rho_2 A_2} - \frac{Y_1 W_1^2}{2\rho_1 A W_2^2} \right) \right\}^{-1/2} \quad (14)$$

But

$$\frac{W_1^2}{W_2^2} = \left(1 - \frac{1}{\psi}\right)^2 \text{ and } \rho_2 = \rho_1\left(1 + \left(\frac{\rho_j}{\rho_1} - 1\right) \Big/ \psi\right) \quad (15)$$

Substituting Equation (15) in Equation (14) gives

$$\psi = \frac{W_2}{W_j} = \left\{ \left(2 \frac{A\rho_1}{A_j \rho_j}\right) \Bigg/ \left[\frac{2Y_2}{\left(1 + \left(\frac{\rho_j}{\rho_i} - 1\right) \Big/ \psi\right)} - Y_1\left(1 - \frac{1}{\psi}\right)^2 \right] \right\}^{1/2} \quad (16)$$

To solve Equation (16), assume $1/\psi \doteq 0$ on the right-hand side. This yields an approximate value of ψ which may then be inserted on the right-hand side to obtain ψ. (Further iterations are unnecessary).

It will be noted that when $Y_1 = Y_2 = 1$ and $\rho_j = \rho_1$ Equation (16) reduces to the conventional ejector relation

$$\psi \doteq (2A/A_j)^{1/2} \quad (17)$$

When ψ is large, Equation (16) can be approximated by

$$\psi = \left\{ 2 \frac{A\rho_1}{A_j \rho_j} \Big/ (2Y_2 - Y_1) \right\}^{1/2}$$

and it follows that decreasing jet density or increasing Y_1 will increase the entrainment ratio (see Figure 3). On the

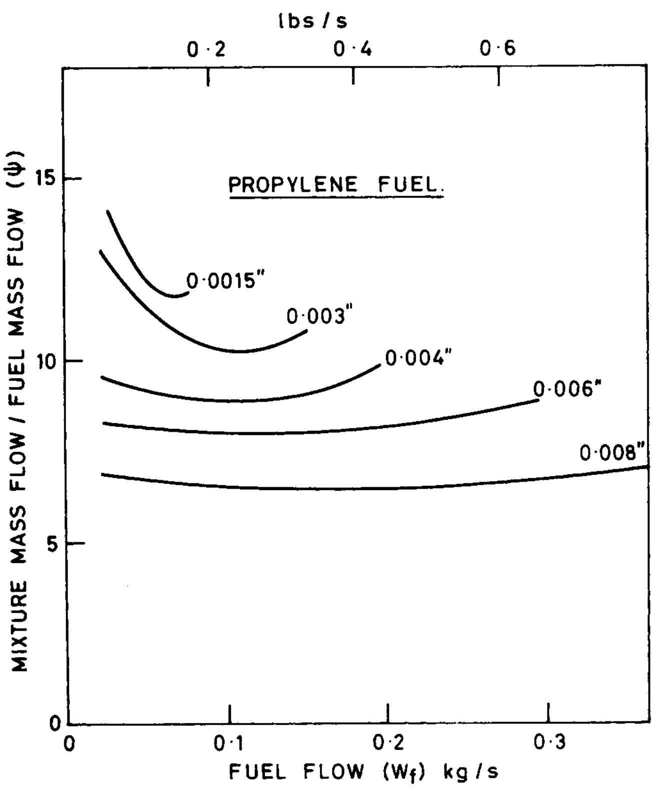

Fig. 6. Variation of entrainment ratio with fuel flow.

other hand increasing Y_2 decreases entrainment. We therefore require the lowest possible value of Y_2 (that is, flat exit profile) and a high Y_1 (that is, peaky entrance velocity profile) to get the highest possible entrainment.

For a typical single jet or Coanda fuel injector as illustrated in Figure 4, the value of $Y_2 \simeq 1$ and $Y_1 = 1.2$ to 1.8 (increasing with size). The corresponding values of entrainment ratio for various 'fuel' gases are plotted against area ratio in Figure 5. As shown in Figure 6, this mixture ratio is approximately constant independent of throughput since it is controlled by stream momentum.

In order to avoid soot formation (that is, minimize radiation and smoke) the mixture ratio for a typical hydrocarbon such as propylene should be weaker than $\psi = 7$ to 10. The area ratio should therefore be about 30, and the fuel throughput capability is determined simply from the fuel pressure and jet area A_j.

If a steam ejector system is used to entrain the air (instead of high pressure fuel), very similar considerations apply except that the steam density is now much less than the fuel density. The steam requirement will therefore be less than the fuel requirement by approximately the square root of the density ratio and area ratio. In the case of propylene we obtain:

$$\frac{\text{Steam mass flow}}{\text{Fuel mass flow}} = \sqrt{\frac{0.0615\, A_{\text{fuel}}}{1.45\, A_{\text{steam}}}} = 0.65\sqrt{\frac{A_{\text{fuel}}}{A_{\text{steam}}}}$$

The steam is usually available at pressures greater than the fuel (for example, 5 atm., 75 lb./sq. in.), permitting the steam ejector system to have a higher area ratio than the fuel ejector system. Steam to fuel ratios of 0.4 to 0.5 are therefore achievable; however the lowest ratios are associated with either high steam pressure and hence high steam jet noise or low burner tip loading and hence a larger and more expensive unit with limited turndown capability.

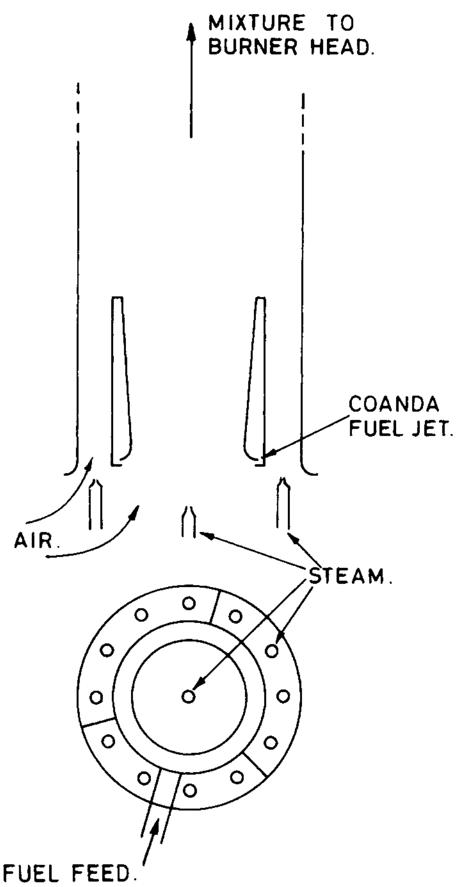

Fig. 7. Fuel injector configuration pilot scale burner.


Fig. 8. View of 5 ton/hr low noise flare system.

Coanda Injector

The Coanda effect is the name given to the tendency of a gas jet to adhere to an adjacent wall. The effect is simply caused by the wall restricting entrainment into its side of the jet which results in a reduction of the local pressure and hence curvature of the jet to follow the wall surface. The Coanda injector behaves very much like a conventional single co-axial jet injector except that the mixing is faster and hence shorter. The radius of curvature of the wall should be greater than 25 times the annular slot width. The performance of a multijet injector is generally superior to either the single jet or the Coanda injector; however, the mechanical design is much more complicated.

The Coanda injector slot does not interfere significantly with the design of a second injection system. Thus for this flare design investigation a combination of a multinozzle injector with a Coanda injector was employed so that fuel or steam could be introduced with either or both injection systems. The injection configuration is illustrated in Figure 7 and 8. A constant area or diverging (7° semiangle) mixing tube, with a length to diameter ratio of 5 is required between the injector and burner head. Although this distance can be reduced, the possibility of flame being blown externally by wind gusts down to the air inlet (causing combustion in the mixing section) means that a short mixer is not advisable. Extensive details of Coanda combustion systems are given in (*4*).

Combustion Head Design

In order to satisfy the stringent pollution requirements posed in this project, conventional combustion systems were precluded. In particular, the required turn down ratio of about 1,000 to 1 eliminated any simple burner inlet port design, since flash back could be encountered at low throughput. Figure 9 shows that for a simple port burner operating at $\phi = 1.5$ the critical boundary velocity gradient varies over two orders of magnitude between flashback and blow off. For turbulent flow, this corresponds to a throughput range of about 10:1. Although the upper limit (blow off) can be extended very considerably by stabilizer design, the possibility of flashback still exists. The solution is to use the fact that for any gas/air mixture, a critical quenching distance exists so that flame propagation down a passage of smaller dimensions is impossible. For propylene or propane/air mixtures this distance is about 2.0 mm. in diam.

It has already been pointed out that turbulence velocities at the burner head must be kept low if noise is to be minimized. In the case of an injector system of the present type, the turbulence intensity at the end of the mixing tube (u'/U) is about 28%. This is a moderately high level and gives a noisy flame if not modified. A conventional way to reduce the turbulence intensity is by means of a honeycomb screen. This reduces the scale of the turbulence from an order smaller than the diameter of the mixer to an order smaller than the dimensions of the honeycomb. Fortunately, the requirement to suppress turbulence and prohibit flashback can both be met by the same technique. Instead of a honeycomb, a wiggle strip structure consisting of corrugated shim strip interleveled with flat strip can be fabricated simply to meet the requirement. This structure may be wound spirally to form a cylindrical burner head. At low throughput, the flame stabilizes on the head with small conical flames at each exit. At high throughput the flame must be piloted at a number of points. This can be achieved by locating low velocity wake regions at various points on the head. A simple configuration consists of an annular pilot fed with mixture from the mixing tube by a series of holes as shown in Figure 10.

Experimental Results

To test the concepts proposed above, a series of combustors were made at different scales. The smallest had a mixer throat diameter of ¾ in. (20 mm) and a flame stabilizer diameter of 2 in. (50 mm). A second version was exactly twice the size of the first. A third unit was built at

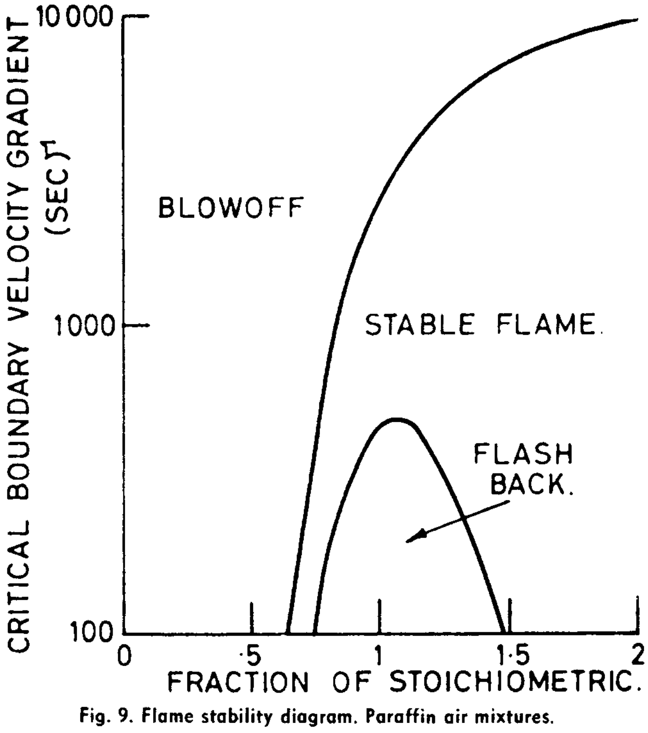

Fig. 9. Flame stability diagram. Paraffin air mixtures.

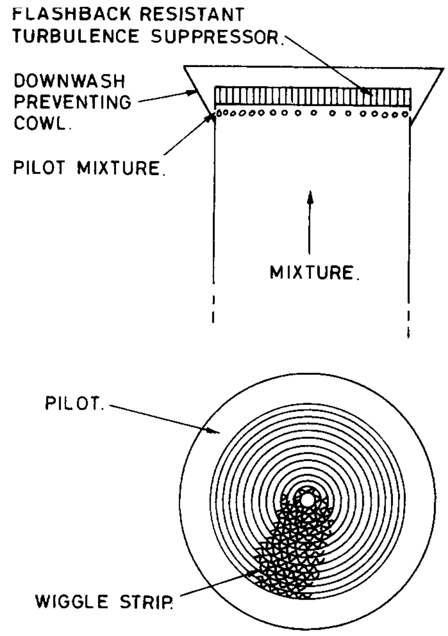

Fig. 10. Burner head design.

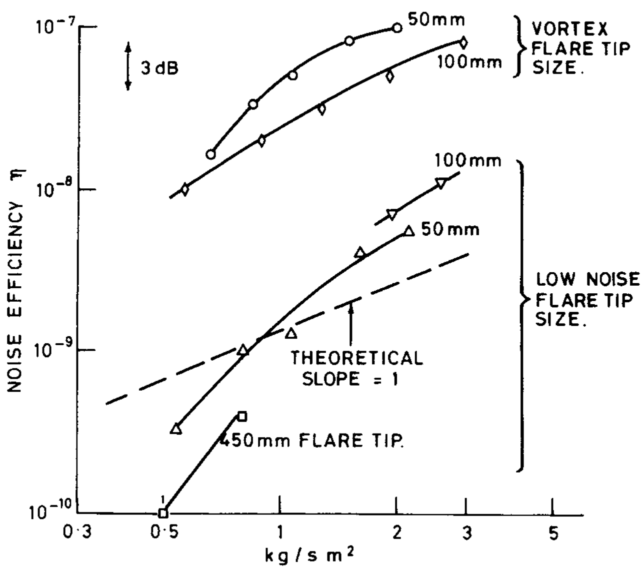

Fig. 11. Noise efficiency versus burner loading (standard and low noise burner tips).

pilot scale to handle 5 tons/hr., and had a constant diameter mixer/burner head as discussed previously. Tests were conducted with methane, propane, and propylene.

The results of these tests confirmed that the inclusion of $\phi < 1.5$ air premix eliminated smoke and radiation, even for unsaturated hydrocarbons such as propylene. The minimum amount of premix air depended on the fuel type, and as expected was slightly larger for propylene. The flame stabilizer also affected the minimum air requirement, such that a strongly swirling flame entrained secondary air from the surrounding atmosphere more rapidly than the low noise burner head. As a consequence, it required a few percent less primary air to avoid smoke.

The noise output of the three sizes of burner head is shown in Figure 2. It is apparent that the noise of any one configuration varies in accordance with the 6 *db*/(doubling of fuel flow). The ¾ in (20 mm.) and 1½ in (40 mm.) throat burners follow a variation with size which is in accordance with theory in that the noise efficiency per unit area loading of the stabilizer should be independent of the burner size for geometrically similar burner heads. In Figure 11 the noise efficiency is plotted against stabilizer loading confirming this result. It may be surprising at first that when a vortex stabilizer is scaled up, the fact that the eddies are now larger does not influence the noise efficiency. In fact this is reasonable since in the analysis presented above it was assumed that the eddies were large compared with the flame thickness (which is true) and the eddy size only effects the spectral distribution of the noise.

The deviations of the actual measurements from the theoretical predictions are generally in the directions anticipated. Thus as the burner throughput is increased, secondary turbulence will be created between the jet and the surrounding atmosphere. This could well explain the slightly steeper slope of the experimental results than the theory. The second factor is the tendency to observe a lower noise efficiency with a larger burner operating at the same loading. This effect is probably due to the fact that the statistical noise production process is modified when the dimensions of the noise producing regions are comparable with the wavelength of the sound. In the case of the 450 mm. burner head, the local speed of sound is about 850 m./sec. giving a reduction in noise output at frequencies of 1 kHz and above. A normal burner only has a small proportion of its output at these frequencies; however the low noise burner has a larger proportion of these high frequency components.

At this point it should be noted that the *dbC* results represent an unweighted noise measurement whereas the *dbA* results are weighted approximately to the response of the human ear.

Safety

During the experimental tests it was found that strong crosswinds led to downwash on the leeward side of the flare stack, especially at turndown conditions. In the worst case this could lead to external flashback of the flame, and in a less severe case would cause heating of the outside of the stack. A simple conical divergence on the top of the stack will prevent downwash so that this feature was combined with the annular pilot.

A more fundamental problem in a commercial flare tip burner of the type studied is the existence of combustible mixture upstream of the flameholder. In the case of a single burner head operating at a maximum rating of 500 tons/hr. (140 kg/s), at 5 kg/sm^2 loading, the diameter would be 6 meters. The mixer length would be 30 meters containing the equivalent of about 1 ton TNT. One solution to this problem is to use many small mixing channels in parallel, which will reduce the overall mixing volume and minimise the effects of combustion in any one mixer. In the event of failure of one or more elements, the relative merits of a fail-safe design, or a control system, would have to be considered. The blockage of the turbulence suppressor by gums present in some gas streams has not been considered in this study.

CONCLUSIONS

1. The fundamental factors controlling the emission of pollutants by a flare combustion system are fuel/air ratio controlling smoke and radiation, maximum flame temperature controlling NO formation, combustion quenching controlling aldehydes, and burner turbulence controlling noise.

2. The parameters which determine the performance of the injector system are area ratio, density ratio, and steam pressure (if used). If steam is used to provide the critical amount of combustion air then steam/fuel consumption will be better than 0.5 (by mass). Steam or fuel jet noise can be a problem when high pressures are used. The production of turbulent combustion noise has been analyzed quantitatively and related to the physical variables.

3. Fuel type and mixture ratio only affect the burner noise output by about 5 db for typical hydrocarbons. The dominant factor is burner turbulence velocity which can be controlled to decrease the combustion roar by up to 20 db.

4. Safety and burner cost considerations lead to the current use of simple flare tips. These, however, can easily form aldehydes, smoke, noise, and other pollutants and the eventual use of more technically sophisticated low pollution units is inevitable.

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NOTATION

A = area = m^2
c = velocity of sound = m/s
d_f = flame thickness = m
E = energy = J
k_d = wavenumber of turbulence dissipation eddies = m^{-1}
k_e = wavenumber of turbulence energy containing eddies = m^{-1}
L_p = sound pressure level db re 2×10^{-5}(N/m^2)
m = mass of fuel = kg
$\dot{m}_f$ = fuel consumption rate = kg/s
p = static pressure = N/m^2
P = total pressure = N/m^2
q = dynamic head = N/m^2
r, R = radius = m
SPL = sound power level db re 10^{-12} watts
u' = root mean square velocity = m/s
U, V = velocity = m/s
V' = volume = m^3
W = mass flow = kg/s
x = volume expansion ratio
Y = velocity profile nonuniformity parameter (Equation 12)
α = fuel/air ratio
ρ = density = kg/m^3
ζ = pressure loss factor
ϕ = equivalence ratio = (actual fuel/air ratio)/(stoichiometric fuel/air ratio)
ψ = entrainment ratio

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