Burning Waste Fuels

Properly designed high-intensity vortex burners are the key to clean combustion

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L ower fuel costs can be achieved by switching from light oils to heavy ones, or to fuel gases, or by combining conventional with unconventional fuels. Combustible-liquid wastes with high-heating values (HHV; see box on p. 69) greater than 8,000 Btu/lb can sustain combustion without support fuel in a standard commercial burner. Although some organic waste can have a HHV of 18,000 Btu/lb, the calorific value of aqueous wastes can be less than 2,160 Btu/lb.

The use of such lower-grade fuels and wastes requires specially designed combustion equipment. Because wastes often contain compounds with chlorine, nitrogen or sulfur, additional equipment to comply with safety and environmental requirements may also be required.

Retrofitting thermal systems with properly designed and operated burners can achieve the required efficiencies and offer an alternative to buying new units. Although improvements in combustion-system design and control may require an increase in capital costs, they will ultimately lower the operating costs.

The high-intensity vortex burner shown in Figure 1 handles waste fuels without producing soot or residue. However, such sophisticated burners must be properly specified and designed to operate reliably. This article offers some tips to design and operate efficient-combustion systems for heavy fuels and wastes. The design considerations presented are based on proven, commercially available technology.
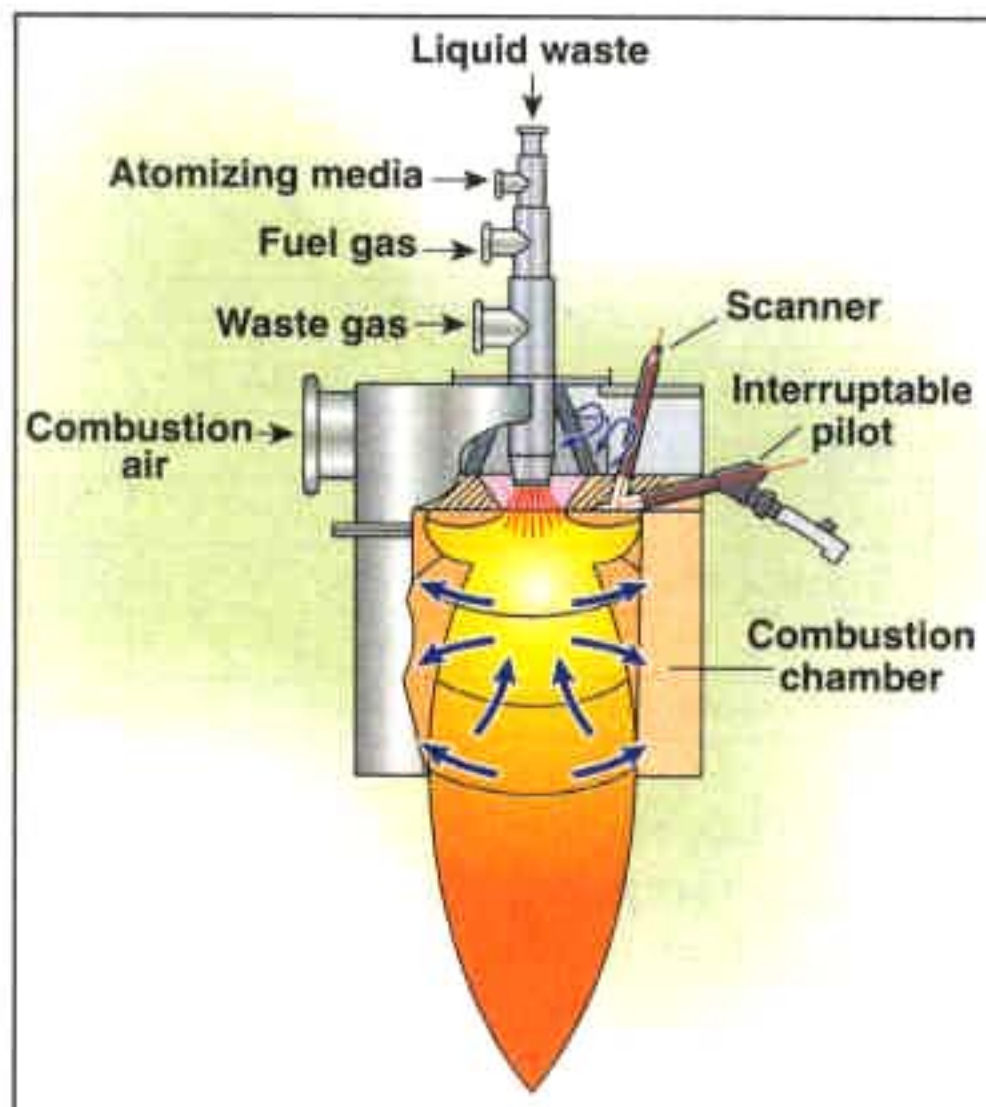

FIGURE 1. The efficient mixing achieved in high-intensity vortex burners ensures good combustion efficiency of waste fuels

Combustor design

There is no set procedure to specify or design a combustor system, but any approach must include:
- Characterizing the fuel and waste used
- Establishing performance objectives
- Developing heat and mass balances
- Evaluating the combustor configuration and burner dynamics

Combustors suitable for heavy fuels and halogenated liquid wastes are high-intensity-vortex burners with refractory-lined combustion chambers, operating at low excess air with a very compact and bushy flame. These combustors can efficiently burn certain fuels, such as heavy fuels with a high C:H ratio, without excessive soot formation, as well as fuels with a high percent of inert components that generate low theoretical flame temperatures.

Although such combustors are designed to operate with low excess air, they unfortunately can produce high NOx levels. Nevertheless, they may be the best choice for some applications. Low-NOx versions of these combustors will be discussed after first considering the burner itself.

Burner design considerations

At the heart of any combustion system is the burner, with properly designed and operated injectors. In general, the burner serves to:
- Atomize and vaporize liquid fuel
- Deliver and mix fuel, waste and air to the combustion zone at the required ratio and rates
- Provide for continuous ignition of the fuel and air mixture
- Give direction and shape to the flame

Proper burner design takes into account the fluid mechanics, combustion chemistry (including pollution emissions), and heat transfer of the system. The interface between the burner and furnace, with the recirculation of the combustion products, also plays a major role in the design.

To achieve efficient combustion, a properly designed burner has:

Proper burner flame turndown: Flame turndown is the relation between the maximum and minimum flows of fuel, using proper controls, good flame stability, acceptable emissions of unburned products, and freedom from combustion pulsation. Factors that influence burner turndown include the type of burner and fuel used, as well as the size and layout of the piping.

Good flame stability: A burner must be designed to maintain the flame over the entire range of operating pressure and temperature, within the limits of control.

Adjustable flame dimensions: The shape and length of the flame are governed by the type and size of the burner, and the operating conditions used. A short, intense flame is generated by operating conditions that provide good mixing of fuel and air. Although large amounts of excess air decrease the flame length, a short flame with reduced excess air can also be achieved by high turbulence and partial recirculation of the combustion products. Increasing the combustion air pressure across the burner typically reduces the flame length, due to

A COMBUSTION PRIMER

Combustion is the rapid chemical reaction of oxygen with the combustible elements of a fuel, resulting in heat release. Hydrocarbons (HCs) are chemical compounds of carbon and hydrogen, and their combustion result in carbon dioxide and water vapor. The oxygen comes from the air, which is 21 vol.% O_2 (23 wt.%) and 78 vol.% N_2 (76 wt.%) and other inerts. Complete combustion is the combination of fuel with O_2 without fuel left over, and requires adequate time, turbulence and temperature high enough to ignite all the combustible elements.

Stoichiometric (or theoretical) air is the amount of air required to provide the exact amount of O_2 for complete combustion. If enough O_2 is supplied, the mixture is lean and the resulting clear and short flame is said to be oxidizing. If excess fuel or insufficient O_2 is supplied, the mixture is rich, the long and smoky (a consequent of incomplete combustion) flame is reducing. A shortage of O_2 can lead to the formation of carbon monoxide, hydrogen, hydrocarbons and free carbon. Regardless of the fuel-oxygen ratio, N_2 in the air absorbs some of the heat, resulting in lower flame temperatures.

The combustion excess air (vol.%), X_S, can be determined from the fluegas dry composition by the Orsat method

$$X_S = 100(O_2 - CO/2)/(0.264N_2 - O_2 + CO/2) \quad (a)$$

where the molecular formulas are the weight fractions of these compounds, and the fuel coefficient K_f is 90 for natural gas and 95 for fuel oils. Alternatively one can measure the volumetric percent O_2 content (dry basis), Y, and use the expression

$$X_S = K_f Y/(21 - Y) \quad (b)$$

The heating (or calorific) value of a fuel can be determined experimentally, either by calorimetry or chemical analysis. The high (or gross) heating value (HHV) is the total heat released of a perfect mixture of fuel and air originally at 60°F and then cooled to the same temperature after combustion. The low (or net) heating value (LHV) is equal to the HHV minus the heat released by condensation of the water vapor in the fluegas, i.e., it assumes all products to remain gaseous. These two heating values are related as

$$LHV = HHV - Q_F, \text{ Btu/lb} \quad (c)$$

where the formation heat, Q_F, is 1,040w, and w is the weight ratio of condensed H_2O to fuel. For HC-gas mixtures, both HHV and LHV, commonly in Btus per standard cubic foot, can determined from the following fitting equations,

$$HHV = 54.4 \, MW + 127, \text{ Btu/scf} \quad (d)$$

$$LHV = 51 \, MW + 94, \text{ Btu/scf} \quad (e)$$

where MW is the average molecular weight of the HC gas mixture. Estimates for the HHV for most fuels can be obtained using the DuLong expression

$$HHV = 14,544C + 62,028(H_2 - O_2/8) + 4050S, \text{ Btu/lb} \quad (f)$$

A modified DuLong equation can be used to approximate the HHV of chlorinated-organic waste

$$HHV = 14,544C + 62,028(H_2 - O_2/8) + 4,050S - 760Cl, \text{ Btu/lb} \quad (g)$$

To calculate the gross calorific value of waste, the following equation is also widely used

$$HHV = 15,410 + 323.5H_2 - 200.1O_2 - 120.5N_2 - 115S - 162Cl - 190F, \text{ Btu/lb} \quad (h)$$

Fuel oils are HCs containing 84–85% C, 12–14% H_2, and small amounts of oxygen, nitrogen and sulfur, as well as traces of moisture and ash. They vary from the light, volatile kerosene type to the heavy, viscous Banker C type. Oils with higher specific gravity (SG) have lower calorific values on a weight (Btu/lb) basis, but their calorific value is higher on a volumetric (Btu/Gal) basis.

The HHV of fuel oils can be determined by the use of the U.S. Bureau of Standards formula,

$$HHV = 22,320 - 3780(SG @ 60°F/60°F), \text{ Btu/lb} \quad (i)$$

or by the Sherman-Kropf formula

$$HHV = 18,250 + 40(°API - 10), \text{ Btu/lb} \quad (j)$$

The specific gravity (SG) and °API are related by

$$SG = 141.5/(°API + 131.5) \quad (k)$$

Ignition temperature is eventually reached when more heat is generated by the chemical reaction than is lost to surroundings, and combustion becomes self-sustaining. The temperature of a flame depends on the type of fuel, the starvation or excess air and the initial temperatures of both fuel and air. Maximum temperature is reached when a fuel is burning near stoichiometric conditions.

The flame temperature of a mixture of HCs can be expressed as a function of the excess air (decimal X_S), by the following fitting equation, within +100 on the high end and −100 on the low end, as

$$T_F = 6,305/(1.85 + X_S), \text{ °F} \quad (l)$$

The flame temperature of a mixture of combustibles and inerts can be expressed as a function of the excess air (decimal X_S) and the LHV (Btu/lb) as

$$T_F = 60 + LHV/\{0.3 [1 + (7.5)(10^{-4})(1 + X_S)LHV]\}, \text{ °F} \quad (m)$$

Flammability limits are the lean and rich mixtures of a fuel with air or O_2 outside of which practical combustion is impossible. Typically, an unstaged flame (see main text) is unstable when the combustion temperature is below 2,200°F. With good mixing, no inerts, and using methane as fuel, the theoretical flame temperature corresponding to the lower limit of flammability (LFL) of 5.4% methane in air is 2,066°F. Adding inerts lowers the flame temperature, as well

as reducing the flame-propagation velocity.

HC gases are important fuels, typically measured in standard cubic feet per hour or sometimes in therms (the volume of gas equivalent to 100,000 Btu). With gaseous fuels, instead of breaking down the HCs into their elements it is easier to use the stoichiometric air required for the different compounds. The stoichiometric air:fuel volumetric ratio is given by

$$V_{air}/V_{fuel} = 2.38(CO + H_2) + 9.53CH_4 + 11.91C_2H_2 + 14.29C_2H_4 + 16.68C_2H_6 + 23.8C_3H_8 + 31.0C_4H_{10} - 4.76O_2 \quad (n)$$

Usually, the theoretical air required for combustion varies slightly for each particular fuel. As an average, each mole of stoichiometric O_2 demand will produce 184,000 Btu on the HHV basis. Typically natural gas is assumed to be equivalent to CH_4, No.2 fuel oil to CH_2 and No.6 fuel oil to C_4H_5.

Stoichiometric combustion (SAR) requires a fairly constant air-to-gas weight ratio of about 16. This leads to the fairly constant value of each 105 Btu of gross-heat release from a gas requiring 1 scf of air. (See the table provided at www.che.com/ceextra). A widely used rule of thumb to approximate SAR for a HC mixture is

$$SAR = Q_G/105, \text{ air flow in scfh} \quad (o)$$

where Q_G is the gross-heat release (Btus per hour), based on HHV. The same approach can be used on the LHV basis, resulting also in a fairly constant value of each 96 Btu of net heat release requiring 1 scf of air (see table at www.che.com/ceextra). We can simplify the calculation of the SAR of a mixture with several components by using the approximation

$$SAR = Q_N/96, \text{ air flow, scfh}$$

where Q_N is the net-heat release (Btu/h).

The amount of water vapor in the air, which reduces the O_2 content, should be taken into account. A humidity factor as well as a temperature factor should be applied to correct the air requirements. Saturated air at 100°F contains 6.45% water vapor, reducing the oxygen volume percentage from 21% to 19.7%. Also, an absolute-pressure correction should be used at significant (500 feet or more) elevations above sea level.   ❏

FIGURE 2. The shape and length of the flame have a big impact on the combustion temperature and residence times

the increase in velocity and turbulence. The temperature along the envelope of the flame is the practical parameter for controlling flame length.

Adequate space for combustion: This goal is related to the shape of the flame, so the furnace should be designed around the flame of a given burner. The furnace volume is governed by the heat released by the burner.

The basic design formula for the burner thermal input, Q, is

$$Q = K_p wVD^2 \qquad (1)$$

where w and V are the density and velocity of the combustion air, D is the burner's throat diameter, and K_p is a proportionality constant.

The two most widely used criteria for scaling industrial burners assume either a constant velocity or a constant-flame-residence time. Constant-velocity scaling assumes Q is proportional to the burner throat area (typically called the burner characteristic) for a given burner geometry. This scaling results in a burner-throat diameter that is proportional to $Q^{1/2}$. The constant-residence-time scaling assumes D/V is constant, which leads to a burner-throat diameter that is proportional to $Q^{1/3}$.

Both of these scaling criteria have limitations, in particular when scaling the interaction between gaseous and liquid phases. Constant-residence-time scaling is less commonly used because it results in low windbox pressures for smaller burner sizes and excessive pressures for larger ones.

Other considerations that influence the burner design and operation include the following:
• A proper ratio of furnace-to-burner diameters
• A constant inlet swirl, to ensure recirculation of the combustion products
• A proper momentum ratio of fuel to air
This last point is especially problematic when heavy oil is used, because the size distribution of atomized oil droplets can be too wide, making it difficult to adjust and control the fuel momentum.

Achieving good mixing

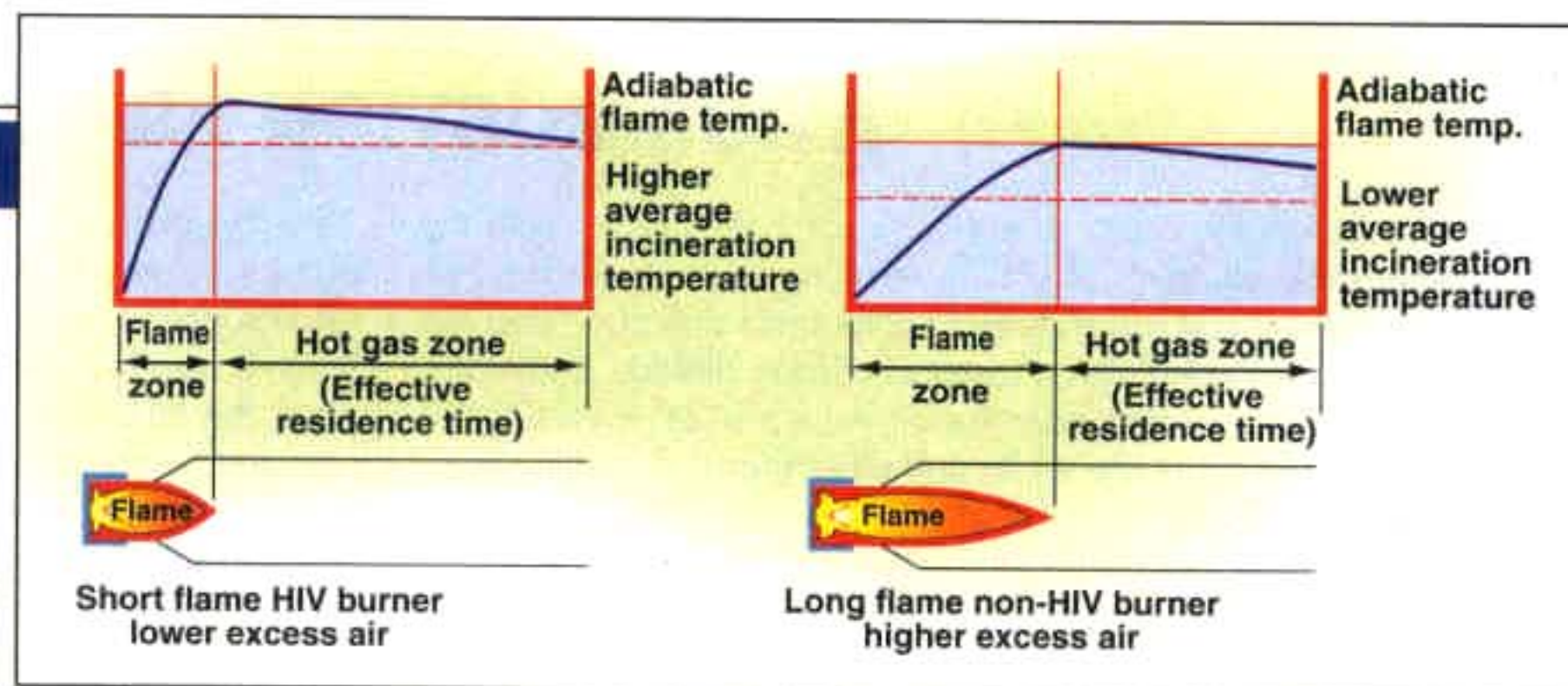A properly designed burner provides turbulent mixing of the fuel and air to achieve efficient combustion at low excess air rates. The swirl number, S, is a dimensionless number that can serve as the main indicator to quantify the turbulent mixing in the burner: The greater the value of S, the better the mixing. The swirl number is given by

$$S = G_o/R_e G_x \qquad (2)$$

where G_o and G_x are respectively the angular and axial momenta of the flux, and R_e is the exit-throat radius. This equation shows that better mixing will be achieved with a smaller R_e, other things being equal.

The swirl also contributes to a dynamic effect taking place in a furnace chamber. In this case, the burner not only facilitates the reaction of the air-fuel mixture, but also stirs up the combustion products within the furnace chamber. The dimensionless number that characterizes the turbulent interrelationship of a flame in a furnace chamber is called the Craya-Curtet number, C_t, which is the square root of the ratio of the initial velocity of the flame in plug flow to the intensity of non-uniformity of the initial flame. This can be expressed as

$$C_t = [(U_d^2 - 0.5U_k^2)/U_k^2]^{-1/2} \qquad (3)$$

where U_d is the dynamic mean entry velocity, and U_k is the total uniform flow velocity, of the products of combustion. A long, lazy flame has a larger C_t number than a short, well-stirred flame.

The combination of high S and low C_t number in a given burner-chamber combination results in efficient mixing with a uniform temperature profile in the chamber, and faster achieves a more uniform heat flux, thus improving the effective residence time. This is shown in Figure 2.

Additional design factors and features are available that can be used to compensate for unanticipated conditions. Computer modeling, combined with experimental data to improve the simulation models, is a useful adjunct. In many cases, the best burner design will be based on experience rather than theory.

Atomized liquid injectors

Liquid fuels must be converted to a gas form, a process facilitated by increasing the exposed surface area through atomization, before combustion occurs. Atomization is the physical breaking up of a liquid into a fine spray, using either the oil's own pressure or the pressure from steam or air. Good atomization minimizes non-evaporated droplets, to achieve efficient combustion.

The liquid-evaporation rate varies with the droplet surface area, so the droplet size distribution is an important parameter. Smaller droplets are easier to vaporize and thus provide better mixing with air. Ideal droplet size is in the range of 40–100 μm. Atomized-liquid nozzles, such as those shown in Figure 3, are used to inject liquid waste through the main burner or into the furnace. The maximum efficiency occurs when the liquid waste passes through the burner itself, rather than into the furnace chamber.

Steam and compressed air are the atomizing media most utilized for burner injectors that are fed different grades of fuel oil and organic wastes. Atomizing steam should be dry and slightly superheated at the burner injector inlet. A steam pressure of 70–100 psig is normally required, but can be as high as 400 psig when atomizing extremely heavy residuals.

Any released gas should be trapped and vented to avoid unstable fuel feed and burning. If air is used for atomizing, it must be heated to prevent cooling the oil. Alternatively, the oil preheat temperature can be raised to compensate for such cooling.

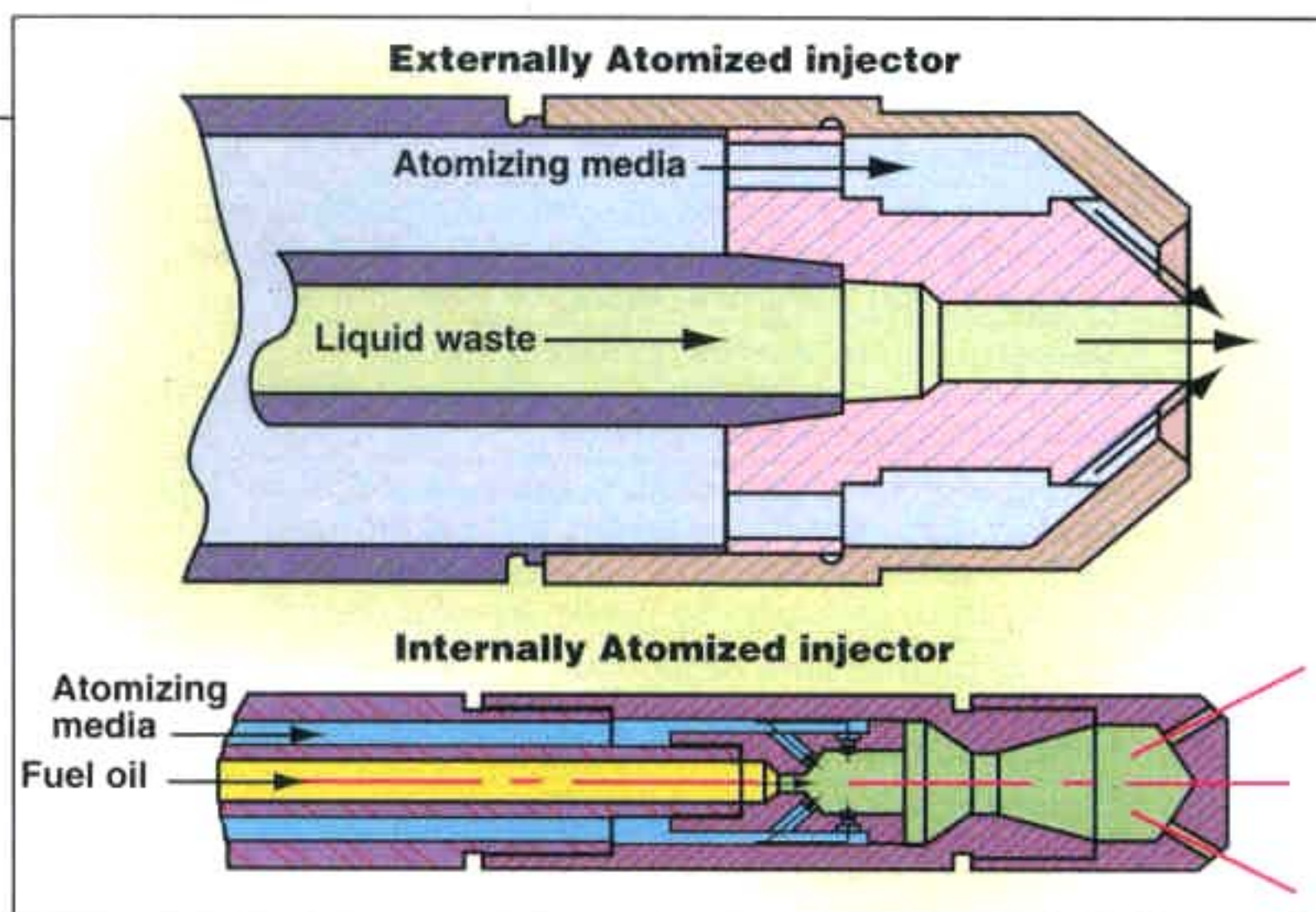Mechanical burner injectors require a higher preheating temperature to

FIGURE 3. Liquid wastes must first be atomized before entering the burner or furnace. In the two designs shown here, the fuel is either atomized within (lower picture) or just downstream (upper picture) the injector nozzle
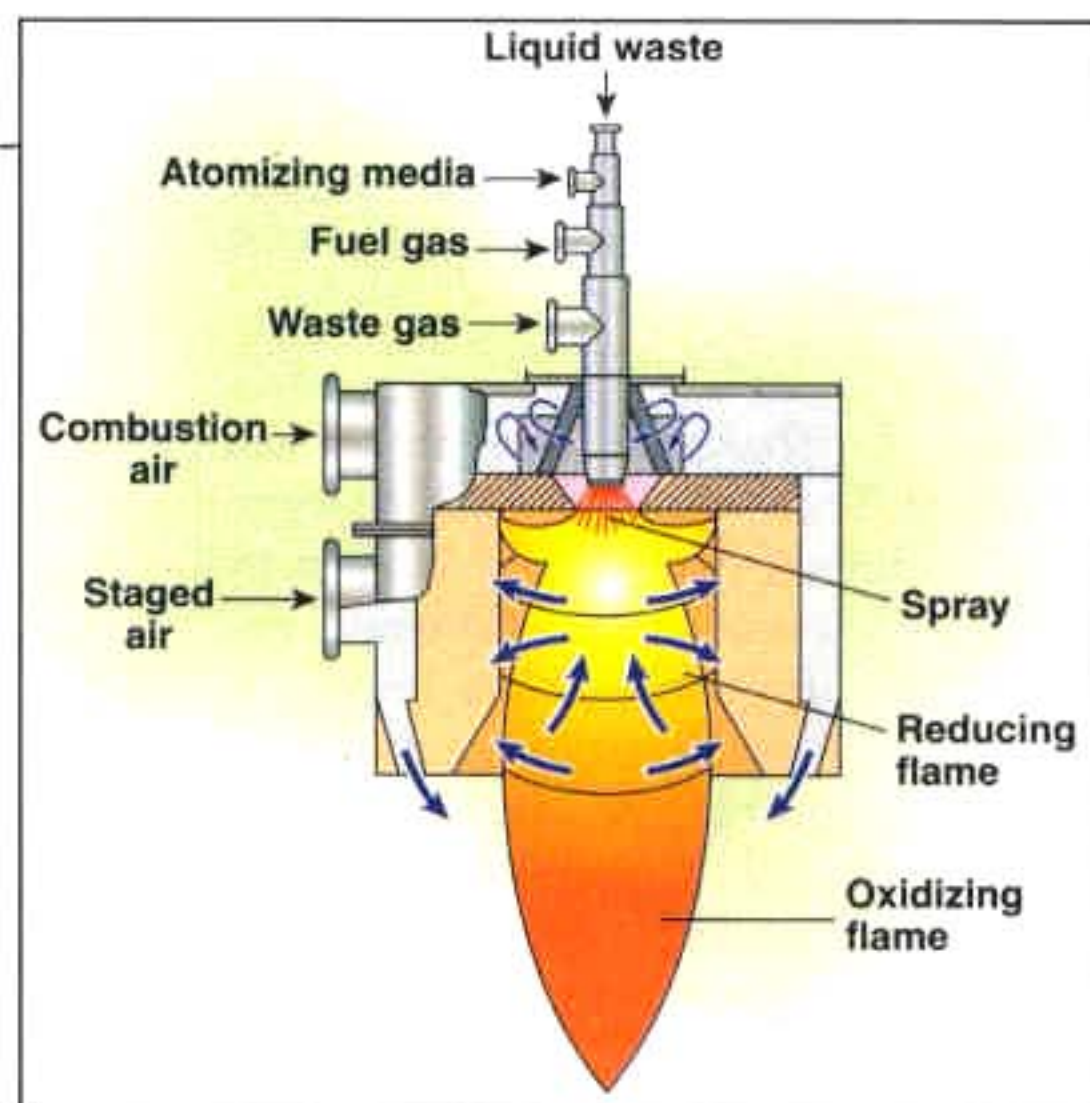

FIGURE 4. The staged-air vortex burner is used to achieve low NOx emissions during the combustion of waste fuel

reduce the viscosity. In this context, note that oil flowrates should be measured on a mass basis. Otherwise a temperature correction needs to be applied, because the volume of the oil varies with the temperature. Also, the specific heat of the oil varies slightly with the temperature. The rule of thumb for the specific heat of oil is about 0.5 Btu/(lb)(deg.F).

The oil's viscosity should be reduced to the proper point by elevating the temperature or thinning with lighter oil. The desired viscosity, depending on the type of atomizer used, ranges from 100 to 200 SSU, with a most common value equal to or less than 150 SSU. The normal temperature limit to reduce viscosity is typically about 200–270°F, but the figure can be as high as 400–500°F. It is common practice to overheat the oil by 30–50°F to compensate for unaccounted heat losses in the piping. However, the preheating temperature should be monitored to avoid polymerization, nitration or oxidation.

Insufficient preheating of the oil leads to poor atomization, resulting in poor burner ignition, unburned fuel, smoke and carbonization.

Excessive preheating can cause cracking and coking of the oil in the preheater, blocking the burner injectors and causing loss of output heat, as well as provoking vaporization and pulsation in the pumps and burners. Excessive superheated steam can vaporize the oil in the injector gun, causing vapor lock at the injector tip.

In some cases, even preheating the oil to to just below a low flashpoint will not be sufficient to reduce the viscosity to the desired level, introducing

the additional danger of preignition. This complication can lead to the formation of light vapors, causing the burner to puff and pulsate. If the preheat temperature cannot be reduced to avoid this problem, light oil can be added to reduce the overall viscosity.

Heat transfer

Heat transfer in both the flame and post-flame zones influences the emissions from the burner. In the flame zone, both the flame length and radiation reduce the adiabatic flame temperature. In the post-flame zone, the temperature of the combustion products is affected by the radiation and convection losses. Radiation losses reduce the NOx emissions.

The heat loss increases as the size of the burner and flame decrease. Also, heat recovered from the post-flame zone and recirculated back to the burner root reduces the flame temperature, and, thus, NOx emissions. This is the basis of new burner technology used to achieve very low NOx emissions with fuel gas.

Low-NOx burners

To properly design a low-NOx burner, due consideration of the flame-to-furnace interaction must be taken. Small flames have short combustion times. The products of combustion remain at high temperature longer in large furnaces than in smaller ones. Minimizing the peak temperature, the O_2 content at peak temperature or the time at the peak temperature can reduce the formation of NOx.

Modifying the combustion process itself is widely used to reduce NOx formation. Such modifications include:

Staged air burners: In staged-air combustion, a part of the required combustion air is added downstream of the burner (Figure 4). This creates two zones: a fuel-rich (reducing) zone and a second (oxidizing) zone. The flame is cooler and longer than that of standard burners. A lower flame temperature in both zones, caused by heat losses to the surroundings, reduces thermal NOx. By reduction in the amount of primary air — optimally to 70% of stoichiometry — NOx reductions of 50% are achieved with natural gas, and 38% with fuel oil.

Staged-fuel burners: These burners also operate in two different zones, but now the fuel instead of the air is introduced in stages. In the fuel-poor (oxidizing) zone, the NOx is reduced by heat losses. This is followed by a second zone, where the balance of the fuel is introduced to combine with the leftover O_2. Staged-fuel burners generally have a cool, long flame.

Water injection: Applied as steam or atomized liquid, water can be used to lower the flame temperature. This method also reduces the formation of particulate matter and allows a reduction in excess air. The combined result can be up to 65% NOx reduction. On the other hand, water injection decreases thermal efficiency and can increase problems with corrosion.

Internal fluegas-recirculation burners: These utilize the burner design for the combustion air or fuel streams to generate a low-pressure zone that induces the downstream fluegas into the combustion zone, reducing the flame temperature. Both flame stability and flashback can be of concern with fluegas-recirculation burners.

External fluegas recirculation burners: These utilize the cooled inert fluegas ducted back to the burner to produce the same effect as internal re-circulation. Both the flame temperature and the resultant oxygen concentration are decreased, resulting in an overall NOx reduction. Both flame stability and flashback can be a concern with external fluegas recirculation.*

Burner operating conditions

The goal of combustion is to convert organic compounds to CO_2 and water, while minimizing the release of pollutants in the fluegas. In practice, combustion is never complete, and wastes can change their chartacteristics at any time. So, excess air is required. However, any excess air beyond that required to complete combustion will waste fuel by putting more sensible heat into the atmosphere.

Burners with greater swirl can operate with low (below 5%) excess air, thus minimizing the SO_2-to-SO_3 conversion. This significantly reduces acid formation and subsequent equipment corrosion. Normally, heavy liquid fuels generate more particulate matter than light ones, and high swirl reduces soot formation. The degree of atomization also influences soot formation.

Over-firing a burner can lead to erosion and lack of recirculation. Fuel injectors should be designed to operate properly at reduced flow to achieve the required burner turndown. The flame length increases when co-firing a gas and liquid, due to reduced recirculation. Also, low-energy wastes burn with a longer and cooler flame.

The following considerations will ensure good flame stability:
- Good fuel-air mixing is the key for stable combustion, lowering the required excess air, reducing ignition temperature and increasing the speed of burning
- Higher pressure drop across the burner throat increases the mixing due to the additional turbulence. Excessive velocity reduces the mixing
- The flame temperature should be above the auto-ignition temperature

*For more on minimizing NOx, see *CE*, July 2001, pp. 66–71, and February 2001, pp. 78–82 and 95–99

A SUMMARY OF THE MOST COMMON BURNER PROBLEMS AND THEIR CAUSES

Problem	Possible causes
Difficult to start	Wrong liquid fuel:air ratio, incorrect atomization, excessive atomizing steam, wet steam, injector partially blocked or plugged, high flashpoint, wrong oil flow, dirty oil
Combustion inefficiency	Deficient combustion air, incorrect fuel:air ratio, incorrect atomization, low flashpoint, low calorific value of organic waste, dirty oil
Lack of atomization	Wrong atomizing temperature, pressure or flow, incorrect flow of atomizing media, wet steam, viscosity too high, incorrect oil-flow settings, excessive dirt, injector eroded, plugged or carbonized, unstable oily waste, injector dripping
Smoke	Lack of atomization, wrong fuel:air ratio, condensate in the steam-atomizing media, flame impingement, cold furnace, pre-ignition, wrong injector position
Soot	High oil ash or carbon particles, incorrect injector position
Unstable or pulsating flame	Oil or atomizing media pressure fluctuations, lack of atomization, wrong injector position, damaged burner throat, low flash point, excessive preheat, low injector turndown, air in the oil line, excessive free water
Burner pre-ignition	Low flash point, excessive preheat, atomizing media temperature
Flame flashback	Low oil flashpoint, atomizing pressure fluctuations, high oil pressure, water content, dirty oil
Carbonization of burner injector	Deficient combustion-air flow, wrong injector position, excessive viscosity, improper oil blend, deficient atomization
Burner injector erosion	Corrosive oil, excessive free water, high content of dirt or salts in the oil or waste
Burner refractory erosion	Excessive dirt, salts and ash content in the oil, wet atomizing steam, flame impingement, undersized combustion chamber

of the fuel-air mixture. The burner block and chamber provide heat back to the mixture to keep the temperature above the auto-ignition point
- Flame instability varies with burner design, but deficient O_2 (< 15 vol.%) will require combustion support to maintain flame stability

Selection of the burner size and operating settings is normally based on the burner excess air, X, and the percent O_2 released in the stack, Y (see box, p. 69, on how to calculate these quantities). Burner standard-selection charts are typically based on pressure drop, dP and heat output. The corrected burner pressure drop, dP_c, is typically a square function of the total combustion air flowrate. Based on the same burner excess air and correcting for actual operating pressure and temperature, it is given by

$$dP_c = dP(14.7/P)(T/530)$$
$$= 0.028TdP/P \quad (5)$$

The burner equivalent heat output, Qe, is a direct function of the total combustion-air flowrate, and it is commonly used to size the burner to provide a proper dP. Based on constant burner excess air, and correcting for actual operating pressure and temperature, it is given by

$$Qe = Q[(14.7/P)(T/530)]^{1/2} \quad (6)$$
$$= 0.167Q(T/P)^{1/2}$$

$dP_c = dP$ and $Qe = Q$ if $T = 70°F$ (= 530°R) and $P = 14.7$ psia.

In practice, the Wobbe index ($W_{H,L}$), an indication of the heat flux of the burner, is used to evaluate the ability to interchange two different fuel gases without changing the settings of burner geometry, valves and controls. A gas burner delivers the same heat output if operating with two gases having similar Wobbe index at similar gas pressures.

$$W_H = HHV/S_g^{1/2} \quad (7)$$
$$W_L = LHV/S_g^{1/2} \quad (8)$$

Burner noise

The unsteady nature of the combustion process leads to pressure pulses that give rise to acoustical noise. Some possible mechanism contributing to the total noise are:
- Combustion roar and super-turbulent combustion noise
- Resonance in the furnace chamber
- Resonance in the burner throat
- Flow-noise amplification and combustion-driven oscillations

The generated sound-power level is affected by the burner-injector geometry and the aerodynamic design. The noise also increases with the following operating conditions:
- Burner size, firing rate and air:fuel ratio

- Combustion intensity
- Higher speed of fuel-air mixing.
- Flame speed (fuels with higher hydrogen content).
- Pressure drop and fluid density
- Smaller liquid-droplet sizes
- Longer flames

Burner troubles and causes

Some of the most common burner troubles and causes when using liquid fuel (or organic waste) are summarized in the table on p. 72. In general, oil firing should generate a bright yellow flame. A dull orange flame is an indication of insufficient excess air

and a white flame denotes too much excess air.

When the burner injectors get eroded or partially clogged, droplets become coarse and inhomogeneously distributed, which reduced combustion efficieny. Also, the liquid can hit the refractory, eventually causing failure. High-pressure and low-flow conditions are indications of a clogged tip. Low-pressure and high-flow conditions indicates the tip is lost or mechanically deteriorated. Liquid-injector guns should be removable to ease cleaning or replacement and to avoid costly shutdowns.

Ashes containing high levels of vanadium and sodium will attack both the combustion-chamber refractory and metallurgy. V_2O_5 acts as a catalyst for conversion of SO_2 to SO_3 adding to sulfuric acid corrosion. Furnaces can operate safely if the vanadium concentration is below 20 ppm.■

Edited by Gerald Ondrey

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