MECHANISMS OF PARTICLE ENTRAINMENT AND COMBUSTION AND HOW THEY AFFECT EMISSIONS FROM WOOD-WASTE FIRED BOILERS

T. N. ADAMS
University of British Columbia
Vancouver, B. C. Canada

ABSTRACT

An analytical model of a spreader-stoker wood waste fired boiler has been developed for predicting the size of the largest particle which can be entrained by the upward motion of the furnace gases. This model is based on an entrainment model which equates particle in-situ settling velocity to furnace gas velocity and uses a stirred reactor model to evaluate furnace conditions. The effects of excess air, furnace size, fuel moisture content, and steam generation rate on furnace temperature and entrained particle size are demonstrated. Calculation of the burning rate of small wood particles under furnace conditions are presented and it is shown that the larger particles entrained by the gases are not significantly reduced in weight by combustion during their passage upward through the furnace.

INTRODUCTION

In regions where there is an active forest industry, by-products such as bark, sawdust, and general wood waste have long been used as fuel for power boilers. This material is usually passed through a crushing or slicing machine called a hog which reduces the size of the largest pieces. Because of this device the material is called hog fuel. Hog fuel is a very inexpensive and therefore attractive substitute for fossil fuels [1, 2] despite the difficulties associated with its relatively low heating value, 19,850 kJ/dry kg (85000 Btu/bone dry pound) and high moisture content, typically 50 percent moisture on a wet basis. This has become even more the case recently as fossil fuel prices have rapidly increased and a degree of uncertainty has been experienced with its delivery.

Particulate emissions from hog fuel boilers have always been a problem. Legislated limitations on particulate emissions from these sources has required either substantial derating of the boilers or increased reliance on flue gas cleaning equipment. Dependence on these approaches will undoubtedly continue, however, understanding of the factors which cause particulate carryover could lead to more economical solutions or improvements to this situation.

Fuel, furnace and operating parameters which affect particulate emissions have been investigated by many workers, primarily for coal [3-6] but also for hog fuel [7, 8]. These studies have been mainly empirical and although they have led to some important changes and improvements to furnace design, they have not led to a generally acceptable method of predicting the effect of changes in operation on particulate carryover. The purpose of this paper is to present a relatively simple model for particulate entrainment which, when coupled with a furnace model, can be used to predict the effect of the major fuel and operating parameters on particulate carryover.

The paper is broken into three sections. In the first section the particle entrainment and combustion model will be presented and the major assumptions in the model will be discussed. Furnace
PARTICLE ENTRAINMENT AND COMBUSTION

The mechanism responsible for entrainment of particulate matter by furnace gases is aerodynamic drag. Any particle for which the aerodynamic drag force due to its relative motion in the furnace gas is greater than its weight will be carried upward through the furnace. For these particles, the in-situ settling velocity is less than the upward velocity of the furnace gases.

Visual observation of the furnace of straight walled, fixed or dumping grate spreader-stoker fired hog fuel boilers leads to the following impression. From the grate to about the level of the overfire jets (approximately the lower third of the furnace) there is a region which appears to be quite turbulent and contains a luminous flame. Above the overfire jets the flame quickly disappears and a uniform looking plug flow pattern develops with some recirculation in the corners. The furnace Reynolds number for the bulk flow is relatively low, so it is expected that turbulence of high intensity and on the scale of the size of hog fuel particles will be quickly damped out and not have a major part in the mechanism of entrainment of the particles above the level of the overfire jets. Depending on the location of the overfire jets, the fuel thrown into the furnace by the spreaders will either be immediately subjected to the relatively uniform upward plug flow or, be subjected to the turbulent zone which will quickly carry the

gas temperature and mass flow rate will be required in order to apply the entrainment model. In the second section a model for predicting these will be presented for the lower third of a spreader-stoker hog fuel fired boiler. In the final section the two models will be combined and the predicted effect of steam generation rate, excess air, fuel moisture, and furnace size on particulate carryover will be presented.
"lighter" particles up to the uniform flow region. Based on these observations the carry-over will simply be the fraction of the incoming fuel which has an in-situ settling velocity less than the plug flow velocity minus the amount of particulate which will be burned during the passage from the turbulent zone to the furnace outlet.

Normally the term settling velocity refers to conditions of still air at normal temperature and pressure. In-situ settling velocity used here is meant to apply to the furnace gas conditions. There is no simple relationship which can be used to relate these two quantities over the entire range of furnace and fuel conditions. The size of particles which are likely to be entrained by normal furnace gas flows are small enough to make particle Reynold's numbers relatively low i.e. less than a few hundred (based on spherical particles with a specific gravity of one whose in-situ settling velocity is just equal to the bulk flow velocity).

Unfortunately, this is well outside the range of applicability of the Stoke's flow model. A somewhat more complex empirical expression for drag coefficient as a function of Reynold's number which is valid over the range from \( 3 < \text{Re} < 400 \) is

\[
C_D = \frac{24}{\text{Re}} + \frac{4}{(\text{Re})^{1/3}} \tag{1}
\]

where

- \( C_D = \) drag coefficient
- \( \text{Re} = \) Reynold's number
This equation applies to smooth spherical particles. Hog fuel particles either fresh or partially burned are neither smooth nor spherical. Fortunately, the drag coefficients for non-spherical particles based on a characteristic dimension follows the same general relationship as for spherical particles in the Reynold's number range of interest [10]. The value of drag coefficient calculated for spherical particles can be multiplied by a shape factor which is characteristic of the shape of the non-spherical particle. It will be shown below that shape factor has the effect of changing the effective particle specific gravity. All calculations have been made for spherical particles of various specific gravities so the results can be interpreted for either spherical particles of the same specific gravity or for non-spherical particles with actual specific gravity greater in proportion to the shape factor.

The bulk flow velocity in the uniform flow region is given by

\[
V_b = \frac{G_T}{A_F \rho_g}
\]  

(2)

where

\[
V_b = \text{Bulk flow velocity}
\]
\[
G_T = \text{Total mass flow rate}
\]
\[
A_F = \text{Furnace crossectional area}
\]
\[
\rho_g = \text{Density of the furnace gases}
\]

If: 1) the in-situ settling velocity of the largest particle to be entrained by the furnace gases is equated to the bulk flow velocity, 2) the density of the furnace gases is neglected when compared to the particle density, and 3) Equation (1) is used for drag coefficient, then it can readily be shown that

\[
\left[ 24 + 4 \left( \frac{G_T D}{\mu A_F} \right)^2 \right] \frac{G_T \mu}{A_F} = \frac{4}{3} \rho_p g D^2 \quad \text{(S.F.)}
\]

(3)

where

\[
D = \text{Diameter of the largest particle entrained by the furnace gases}
\]
\[
\mu = \text{Absolute viscosity of the furnace gases}
\]

\[\nu = \text{Kinematic viscosity of the furnace gases}\]
\[\rho_p = \text{Density of the particle}\]
\[\text{S.F.} = \text{Shape factor of the particle}\]

It can be seen from Equation (3) that the easiest interpretation of shape factor is that it modifies the apparent particle density.

In order to apply the equation it is not only necessary to have values for mass flow rate and furnace area but also furnace gas viscosity. For a wide range of furnace gas composition spanning both dry, low volatile coal to wet, high volatile hog fuel it can be shown that furnace gas viscosity is not substantially different from that for dry air at the same temperature [11, 12]. The absolute viscosity of air is well correlated by the following equation in the temperature range encountered in normal furnace operation

\[
\mu = 182 + 0.245 T_g \text{ micro g/cm-sec}
\]

(4)

where \[T_g = \text{Furnace gas temperature, } ^\circ\text{K}\]

Kinematic viscosity is the absolute viscosity divided by the density. For a furnace pressure of one atmosphere, the furnace gas density depends only on temperature and average furnace gas molecular weight.

It can be seen that the entrainment model must be coupled to a furnace model which predicts furnace temperature, \[T_g\], if estimates of particulate carry-over as a function of operating conditions are to be obtained. The furnace model is the subject of the following section.

The entrainment/furnace model will predict the size of the largest particle which will be entrained by the furnace gases. Some of the entrained particles will burn during their passage through the furnace. In order to assess the effect of burning of particles that are entrained, calculations were made of the burn time to half mass for various size particles. The approach for these calculations was that of reference [13]. Additional data on combustion of particles and wood particle properties were taken from the literature [14-17]. Figures 1 and 2 are plots of particle burn time to half initial mass as a function of initial diameter for dry, devolitilized spherical wood particles and are parametric in furnace gas temperature and oxygen partial pressure. Furnace wall temperature is specified and the
emissivity of the wall is taken as unity. Figure 1 is for a particle in the luminous zone where the combined emissivity of the soot, water vapor, and carbon dioxide is taken as 0.65. Figure 2 is for a particle in the non-luminous uniform flow region where the furnace gas emissivity is due only to water vapor and carbon dioxide and is typically about 0.3.

In a following section it will be shown that wood particles as large as a few millimeters can be entrained. It is clear from Figures 1 and 2 that for typical gas residence times in the furnace of between 2.5 and 4 seconds, significant combustion occurs for only the smallest size entrained particle. Larger particles escape the furnace with perhaps only a charred surface.

The entrainment model will predict the size of the largest particle entrained by the furnace gases. From Figures 1 and 2 it is possible to estimate the size of the largest particle which will be completely burned in the furnace. Carryover will then consist of material which has an initial size between these two limits. Considering the differences in emissions between spreader-stokers and other types of stokers, it can be concluded that carryover is due primarily to raw fuel which is sized between these two limits rather than from partially burned material which is spouted from the bed.

**FURNACE MODEL**

In order to make use of the particle entrainment model it is necessary to obtain a value of furnace gas temperature. There has been a great deal of work on models for boiler furnaces with the primary goal of predicting furnace heat absorption efficiencies. Starting back in the 1920’s both theoretical and empirical models have been put forward [19-23]. These have been applied and improved over the years [24-25] while in the last two decades, stirred flow and stirred flow-plug flow reactors, as well as zone models, have been developed [26].

The significant improvement in making a priori calculations with these later models is to a large degree offset by their computational complexity. For any but the most complex models there has been the chronic problem of correctly identifying a furnace radiant mean temperature and a luminous flame emissivity. The approach adopted here makes use of the simplest stirred reactor model and circumvents some of the past difficulties by using an empirical expression of soot emissivity and by limiting the stirred flow assumption to the luminous zone which occupies roughly the lower third of the furnace.

The approach adopted is conceptually identical to that of Wohlenberg [21]. An energy balance is made involving inputs from fuel and preheated air and outputs due to sensible heat of the furnace gas at the exit, convection to the walls, and radiation to the walls and the furnace exit opening (or in the case of the lower-third-model, to the walls and non-luminous gas above the luminous zone). Sensible enthalpies are evaluated with analytical expressions which reflect present knowledge of thermodynamic properties at elevated temperatures. The convective heat transfer coefficient is taken at 11.4 watts/m²K (2Btu/hr ft²°R) with temperature difference evaluated as bulk gas temperature less wall temperature. Evaluation of the radiation term is discussed below.

Comparison of boiler furnace heat absorption data from stoker fired units with predictions based on the Wohlenberg approach, or one of its modifications, has nearly always shown the predictions to be low. This is due to two assumptions: 1) that the mean radiant temperature is equal to the furnace exit gas temperature and 2) the emissivity of the furnace gas is due to water and carbon dioxide only, while the emissivity of the fuel bed is unity and its temperature is that of the gas above it. These assumptions have been relaxed in more recent work, however treatment of grate burning furnaces with the new approaches has been limited.

Review of the experimental data available on stoker fired units and visual observation leads to three conclusions: 1) the discrepancy between theory and experiment increases as the volatile component of the fuel increases [22], 2) temperature of the radiating gas is approximately 90°C higher than that of the exit gas [24] (see also reference 26 pages 463-468) and, 3) the bed on overfeed stoker units appears to be well below the gas temperature. The last observation is based on the fact that fuel beds (for hog fuel boilers) appear dark in color which is not consistent with temperatures in the range of 1200°C to 1400°C for relatively high emissivity surfaces. Based on these conclusions a very simple model of the radiation term has been developed which has the form

\[
Q_{RAD} = 1.1 \ A \ \epsilon \ g \ \left( T_g^4 - T_w^4 \right) 
\] (5)
where

\[ Q_{\text{RAD}} = \text{Radiation from the furnace gases and the solid fuel} \]
\[ A = \text{An area. See discussion below.} \]
\[ \varepsilon = \text{Effective emissivity} \]
\[ \sigma = \text{Stefan-Boltzmann constant} \]
\[ T_g = \text{Furnace gas temperature. See discussion below.} \]
\[ T_w = \text{Furnace wall temperature.} \]

The effective emissivity is that due to soot and to water vapor and carbon dioxide. The emissivity of the latter two has been evaluated using Hadvig's approximation [27] as reported in reference [26]. The effective emissivity is taken as the sum of the emissivities of soot and the non-luminous gases, minus their product. The emissivity of soot was evaluated with an expression of the form

\[ \varepsilon_{\text{soot}} = 1 - e^{-k_vL} \]  \hspace{1cm} (6)

where

\[ \varepsilon_{\text{soot}} = \text{The emissivity due to soot in the luminous zone of the flame} \]
\[ k = \text{A constant equal to 0.024 m}^{-1}. \text{See discussion below.} \]
\[ v = \text{Fraction of volatile matter in the fuel.} \]
\[ L = \text{The effective gas radiation path length}. \]

The constant \( k \) was taken as the value which provided the best fit of the data reported in reference [24] when using Equation (5) to evaluate the radiation term. A value of 0.024 m\(^{-1}\) was found by trial and error.

The values of \( A \) and \( T_g \) have a slightly different meaning for the full furnace and the lower-third of furnace case. For the full furnace, \( A \) is the total wall area plus the projected area of the furnace exit opening and \( T_g \) is the exit gas temperature plus 90\(^\circ\)C (This is essentially the incompletely stirred reactor assumption). In the lower third case, \( A \) is the wall area only and \( T_g \) is the gas temperature which is equal to the temperature of the gas leaving the lower third of the boiler. A cold wall fraction of 0.80 is assumed in all cases and the results are not substantially different for values of fraction cold above this value.

Equations (5) and (6) are so simple that they cannot model the very complex physical situation which exists in the furnace in any but an empirical way. The redeeming feature of this approach is that it does improve the correlation between the predicted and measured heat absorption efficiencies of straight wall overfeed stoker furnaces which have a large fraction of cold wall surface. Application of this model to hog fuel boilers of similar design (size, shape, rating, overfire, air, etc.) will not lead to serious error. This is particularly true in the case of wet hog fuel. Adiabatic flame temperatures for wet hog fuel are typically less than 1600\(^\circ\)K so that radiation is a relatively small term under normal furnace conditions (1100\(^\circ\)K to 1400\(^\circ\)K) despite the predicted high emissivities of the model. Radiation in the following calculations typically accounts for only 10 percent of the total energy leaving the lower third of the furnace.

The furnace model has been used to predict \( T_g \), the temperature of the furnace gas leaving the luminous zone, under various operating conditions. Figures 3, 4 and 5 show the effect of operating conditions on the predicted value of \( T_g \). The following assumptions were made for these calculations: Hog fuel properties — volatile matter = 75 percent, carbon = 50 percent, Hydrogen = 6 percent, heating value = 18,850 KJ/dry kg, stoichiometric air-to-fuel ratio = 5.97, Furnace-wall temperature = 533\(^\circ\)K.

**RESULTS AND DISCUSSION**

The furnace model and entrainment model have been combined to predict the effect of various
operating parameters on the size of the largest particle entrained by the furnace gases. Figures 6 through 9 show these effects for spherical particles of three specific gravities, 0.1, 1, and 3. Dry hog fuel particles have specific gravities between 0.1 and 0.5 [6] but their shape, typically elliptical disks, tends to change the effective density. Thus the actual size of particles of wood which would be entrained will be between the values predicted for 0.1 and 1 specific gravity. Sand and ash tend to have higher specific gravities and will be better predicted by the curves for specific gravity 3.

For the low density combustible component of carryover it can be seen that particles as large as 5 mm to 10 mm can be entrained. These particles will not be significantly reduced in size due to combustion as they move through the furnace, based on the burn times of Figures 1 and 2. The smaller particles which will be entrained along with these large particles will lose a larger fraction of their weight due to combustion. The net effect of a change in one parameter would then depend on its effect on both combustion and entrainment and, as well, the fuel size distribution. For example, from Figures 1, 2 and 6, the effect of an increase in excess air is to increase both combustion and entrainment. The net effect then depends on the relative amount of large and small particles. For hog fuel with a high proportion of sawdust, which is typically about 1 mm in size, the net effect of increasing excess air would be to reduce the carryover. If there is little sawdust or other small, light
FIG. 6 EFFECT OF EXCESS AIR ON PARTICLE ENTRAINMENT FOR THREE PARTICLE SPECIFIC GRAVITIES, S.G., AND TWO STEAM GENERATION RATES, $M_s$, GRATE AREA, $A_F = 23.8$ m$^2$, FUEL MOISTURE, $m = 50$ PERCENT.

FIG. 7 EFFECT OF STEAM GENERATION RATE ON PARTICLE ENTRAINMENT FOR TWO PARTICLE SPECIFIC GRAVITIES, S.G., AND THREE FUEL MOISTURES, $m$, GRATE AREA, $A_F = 23.8$ m$^2$, EXCESS AIR, $= 50$ PERCENT.
material, then the effect would be to increase carryover.

No such uncertainty exists for fuel moisture. From Figures 1, 2 and 8, an increase in moisture causes an increase in entrainment and a decrease in combustion. Therefore the net effect of an increase in fuel moisture is to increase carryover.

CONCLUSIONS

The analytical model of a spreader-stoker type hog fuel boiler that has been developed can be used to predict the size of the largest particle which can be entrained by the upward motion of the furnace gases. Combustion calculations of the burning rate of wood particles indicates that only the smallest particles entrained by the gas will be significantly reduced in mass before they leave the furnace. Because the two effects, combustion and entrainment, are both affected by changes in operation it will be necessary to use a fuel size distribution to assess the net effect of a change in operation on carryover.

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REFERENCES


Key Words

Particulate Emissions
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