Modeling Ignition and Burning Rate of Large Woody Natural Fuels

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Abstract. As part of the development of a model for predicting fuel loading reductions by and intensity histories of fires burning in large woody natural fuels, it was necessary to develop separate models for the processes of ignition and rate of burning of individual fuel elements. This paper describes the derivation of predictive equations for ignition delay time and burning rate (from diameter reduction rate) of large woody natural fuels in a fire environment. The method consists of deriving approximate functional forms using fuel component properties and a measurable "fire environment temperature" and then fitting these forms to data taken in laboratory fires using a large propane burner. The equations describe the calibration data with precision adequate for the purpose for which they were designed.

Keywords: Ignition; Burning rate; Woody fuels; Natural fuels.

Introduction

Wildland fires sometimes burn in fuel complexes that include various amounts of the larger components of woody plants, such as limbs and boles of trees or woody shrubs. While alive, these large plant parts seldom burn in a wildland fire, whether deliberately set or unintended. But when they are dead, and especially when dead and arranged in accumulations on the surface, they can burn so vigorously as to concern fire control and fire use planners. Heavy concentrations of dead large fuel components can arise from causes such as windthrow or avalanche in forest stands, breakage and cull during timber harvest, or mechanical clearing of shrublands.

However such fuel accumulations occur, they often demand the attention of fire control planners because they can sustain fires of relatively high intensity for a prolonged period of time, defying direct suppression efforts and serving as potential sources of spot fires.

These considerations arise during presuppression planning in assessing an area's potential resistance to control.

Another important issue raised for planners is assessment of the consequences to be expected from a fire in such fuels under various conditions. The amount of heat generated per unit area varies as the total fuel loading burned. And as heat per unit area increases, so does the depth to which it penetrates into the underlying soil and the higher the peak temperature at any soil depth. A host of prompt, thermally induced, on-site effects can be keyed to the time-temperature profile experienced in the soil. Delayed on-site effects such as soil erosion, chemical and biological ramifications of the release of sequestered minerals, and alteration of soil physical character are also influenced or controlled by the amount and rate of heat released per unit area. Off-site effects on air and water quality are also tied to the amount and rate of fuel consumption. This cascade of consequences has motivated field research over the years, to provide land managers a means of using fuel descriptors that can be measured or estimated to make a priori estimates of fire effects that they need to control. But the consequences are many and often influenced by uncontrollable environmental factors, complicating compilation of an adequate data set for each fire effect that must be quantified. Considering such variables as fuel quantity, size distribution, condition, arrangement, moisture content, etc., the variability of burn site conditions and environment, and the need for replication of experiments, a broadly comprehensive yet strictly empirical approach seems impracticably slow and costly.

Consequently research on wildland fire effects has embraced what can be called an analytical perspective (Hungerford 1990). By this we do not mean that theoretical mathematical predictions have replaced observational data. Rather we mean that the complex chain of causes and effects is being explored "link-by-link", in order to fashion a set of models, each of limited

scope, which can be used together to predict a wide variety of effects under a wide variety of conditions. Under this paradigm, some models might be theoretical and mathematical (e.g., predicting thermal response of moist soil to a specific surface heating regime) while others might be entirely empirical (e.g., correlating the probability of survival of a rhizome to its temperature-vs-time experience), yet they could be used together.

One member of such a suite of models should predict the quantity and rate of consumption of large woody natural fuels. This model should be useful to fire management planners and would serve as a driver element in predicting many fire effects. A cooperative effort by personnel of USDA Forest Service's Intermountain Fire Sciences Laboratory and Montana State University is currently under way to develop such a large fuel burnout model. The research reported here is a part of an early phase of this model development effort.

Approach to Model Development

The model being developed is intended to predict fire intensity (kW/m^2) as a function of time, by combining the burning rates ($kg/s\text{-}m^2$) of the fuel components. Fire intensity would then be used to derive an equilibrium temperature characterizing the fire environment of the fuel elements. This temperature would be used to predict the rate of heat transfer to each of the fuel elements. From its heat transfer rate history the burning rate of each fuel component is to be calculated, thus closing the model. It should also predict, incidentally, the quantity of fuel of each type that would be consumed in a fire and the burning time of each. It is intended that this new model replace an earlier one (Albini 1976) based mostly on conjecture with one based upon realistic phenomenology and a firmer empirical footing.

The investigations described here were undertaken to obtain equations for two aspects of the burning process — the time elapsed before ignition occurs and the rate of burning once ignition occurs — using fuel element properties and fire environment descriptors. We characterize the fire environment by a temperature and predict the rate of heat transfer to fuel components placed in this environment. We next express the connection between heat transfer rate and temperature rise, using findings reported by earlier investigators and simple analyses. These equations involve fuel properties as parameters.

The functional forms between parameter groups are then used to correlate experimental data from continuously weighed natural fuel specimens exposed to a propane fueled steady flame environment in the laboratory. Using these data, we find the empirical coefficients that best describe the data and thus derive the desired predictive equations. Fuel specimens used were cylindrical limbwood and small tree bole segments approximately 50 cm in length and 2.5 - 10 cm in diameter, with and without bark, in both sound and rotten conditions. The data fits rely upon (measured or estimated) fire environment temperatures; computed effective film heat transfer coefficients; observed ignition delay times; measured histories of burning fuel element weights; measured values of fuel element initial diameter, mass density, and moisture content; some putative constants such as the thermal conductivity and specific heat capacity of ovendry wood; and some nominally universal relationships such as the dependence of wood's thermal conductivity upon its mass density and moisture content. These quantities and the roles they play are described in more detail in the following three sections. Those not interested in such detail might profitably skip to the summary section.

Modeling Heat Transfer in a Fire Environment

A small inert object immersed in the flame volume of a fire in natural fuels will exhibit a temperature that depends upon the net rate of heat transfer to the object and upon its thermal mass (thermal mass being the amount of heat needed to change its temperature by 1 K). If it has little thermal mass, its temperature will fluctuate with time. For example, a bare thermocouple made of very fine wire will typically exhibit temperature swings of 100°C or more over tens of illiseconds. But if it is a few grams or more in mass, it will achieve and maintain an interior temperature that is esentially constant, depending only upon the fire's intensity, the object's position in the flame structure, and the radiative properties of its outer surface. It reaches a state of equilibrium in which the net rate of heat transfer to the object, averaged over a period of time on the order of a second, is zero. This temperature characterizes the fire environment at the location of the object.

It has been noted experimentally that, over much of the vertical extent of turbulent diffusion flames, long-term average temperature is roughly constant but declines rapidly with height above the continuous flame zone (Hasemi and Tokunaga 1983). This height in turn depends upon the chemical energy supply rate, as measured by the mass flow rate of fuel, for isolated flame structures. It depends upon the area intensity when an area is burning with an average intensity less than a critical value such that one large merged flame structure cannot be maintained (Heskestad 1991).

In any event, we assume that there exists a "fire environment temperature", T_F, a function of the fire's area intensity (kW/m²), which represents the temperature that an inert object ultimately would achieve if it were kept in the fire environment where T_F is determined. Note that this temperature need not be the time average value of the temperature of the flame gas at that location, nor a measure of the local radiation intensity, but a parameter that involves both these features of the flame environment. Nevertheless, this measurable quantity can be used to model the rate of heat transfer to a fuel component in that environment, by assuming that it is both the local flame gas average temperature and the temperature that characterizes the local radiation intensity, although it actually may be neither.

Heat will be transferred to and from the surface of a fuel element by convection (i.e., by conduction through the gas in contact with, and moving relative to, the surface of the fuel element) and by radiation. The rate of heat transfer by convection can be modeled as the product of the difference between the ultimate temperature (T_F) and the surface temperature (T_s) and a Newtonian film heat transfer coefficient, h:

$$dq''/dt \text{ |convection} = h \ (\ T_F - T_s \ ) \qquad (1)$$

where t is time and q'' is heat transferred per unit area. The parameter h has been correlated to surface geometry, fluid properties, and relative velocity using dimensionless parameters. For the present purpose, we use the form

$$Nu = 0.344 \ Re^{0.56} \qquad (2)$$

(Brown and Marco 1958) that approximates well more recent but more complicated correlations (Incropera and DeWitt 1985). Here Nu is Nusselt number, hD/k_a, and Re is Reynolds number, VD/v, where D is fuel element diameter, and k_a is the thermal conductivity of flame gas, taken to be hot air. V is the relative velocity of gas across the long, cylindrical fuel element, a function of windspeed and height within the flame structure. Parameter v is the kinematic viscosity of hot air, a strong function of temperature. When the film heat transfer coefficient given by equation (2) falls below the value characterizing natural convection (in contrast to "forced" convection, to which the equation applies), the value for natural convection is used.

The effect of heat transfer by radiation can be included by using a modified film heat transfer coefficient. The radiative heat transfer rate should approximately follow the form:

$$dq''/dt \text{ |radiation} = E \ (\ T_F^4 - T_s^4 \ ) \qquad (3)$$

with E representing the product of the Stefan-Boltzmann radiation constant (5.67×10^{-8} W/m²-K⁴), a geometrical "view factor", and the integrated emissivity of the fuel surface. A value 0.5 for view factor x emissivity was arbitrarily assumed. (Ignition time delay varies inversely with the square of the effective heat transfer coefficient and burning rate is proportional it, so the effect of such an arbitrary numerical factor assignment appears at first to be substantial. But these functional relationships are fitted to experimental data to derive working equations for the model, so the arbitrary nature of this assignment is largely negated. The numerical value influences the partitioning of the total heat transfer between convective and radiative modes, but since their combined effects are treated as equivalent to convective heat transfer, this influence is also minimized). The temperature function can be factored to yield the form

$$dq''/dt \text{ |radiation} = E \ (\ T_F - T_s \ )(\ T_F^3 + T_F^2 T_s + T_F T_s^2 + T_s^3) \qquad (4)$$

and the cubic polynomial in T_F approximated by using an appropriate average value of T_s. The product of E and the polynomial factor is then added to h to give h_{eff} and the total heat transfer rate is then approximated by

$$dq''/dt \text{ |total} = h_{eff} \ (\ T_F - T_s \ ) \qquad (5)$$

This formulation is used to calculate the rate of heat transfer to fuel elements before ignition in order to predict the time elapsed between immersion of an element into the flame environment and the attachment of flame to its outer surface. It is also used after ignition, to predict an element's burning rate. In obtaining data for correlations, we measured the laboratory flame environment temperature in about one fourth of the experiments. For the others, we assumed T_F to be the average of the measured values. To measure T_F we used a Chromel-Alumel thermocouple embedded in a crude sphere of Sauereisen cement about 2 cm in diameter. This sensor achieved a virtually constant temperature a few minutes after having been suspended in the flame.

Ignition Time Delay

Several models have been created to predict the ignition time delay of woody fuel elements under various heating regimes. An early series of experimental investigations conducted in England established that

ignition of radiatively heated small panels of wood and other cellulosic materials could be successfully correlated to their achievement of a specific surface temperature (Simms 1963, Simms and Law 1967). These investigations further showed that the time required to reach this temperature could be predicted using the intrinsic properties of the fuels, by assuming the fuels to be inert and solving the halfspace transient heat transfer problem.

The validity of using a critical temperature to characterize the ignition process was later proved for fine forest fuels (Stockstad 1975, 1976) and for rotten natural woody fuels (Stockstad 1979). Although the reported values vary somewhat, piloted ignition of sound fuels appears to occur at about 600K and for rotten woody fuels at about 550K.

Although analytical and numerical attacks on this problem have grown increasingly sophisticated and general (Niioka 1978; Moallemi, Zhang, and Kumar 1993) little has changed in the level of understanding of the processes involved. Simms and Law (1967) discuss experimental measurements by Williams (1953) that support the simplification that the moisture in "rapidly" heated wood remains essentially frozen in place until the temperature rises to about 100°C, whereat the moisture evaporates. They use the thermophysical properties of moist wood to describe the thermal response of a radiatively heated inert halfspace from the start of heating until ignition. They increase specific heat capacity to account for latent heat of vaporization of water. Using this model they reduced data from a series of experiments and obtained remarkably consistent results. Delichatsios, Panagiotou and Kiley (1991) treat the inverse problem and propose the use of time-to-ignition data to characterize thermal inertia and minimum heat flux required for ignition.

The experimental setup used by Simms and Law can be described as follows: Radiation of intensity I (W/m^2) is normally incident upon and absorbed by a thick plane slab of wood initially at uniform temperature T_a and moisture content M (dry mass fraction). The absorbed radiant energy heats the surface to temperature T_s that is greater than the ambient temperature T_a so the hot surface transfers heat to the environment at the rate $h(T_s - T_a)$. The excess of energy absorbed over energy lost increases the interior (and surface) temperature of the wood, until the heated surface reaches ignition temperature. The wood panel is modeled as infinitely thick.

We modified the analysis of this problem, hoping to model better the actual physical process. First, we calculate the time required for the surface of a fuel specimen to reach drying temperature, using the thermophysical properties of moist wood. Heating begins with the fuel component initially at ambient temperature T_a. Then we compute the time from the start of drying until ignition, assuming an inert medium with moist wood thermophysical properties, including an adjusted specific heat capacity to account for latent heat of vaporization of water, but beginning with the fuel uniformly at the drying temperature, T_d. Film heat transfer with a constant temperature environment (T_F above) is equivalent to constant radiant heating with film cooling:

$$dq''/dt \text{ ltotal} = h_{af}(T_F - T_s) = h_{af}(T_F - T_a) - h_{af}(T_s - T_a), \tag{6}$$

where $h_{af}(T_F - T_a)$ represents the constant radiant heat input, I. Because the fuel element is treated as inert, the problem as posed can be solved in either the cylindrically symmetrical case or as a halfspace. In the cylinder problem, an infinite series of Bessel functions of order zero and unity are involved in the expression for the transient temperature distribution (Abramowitz and Stegun 1972), and evaluation of the surface temperature requires finding a series of roots of the zero order Bessel function. This complexity was found to be unnecessary, as the cylinder and the halfspace show quite similar surface temperature histories to this kind of heating environment. The halfspace solution can be presented concisely as the fractional temperature rise, $\phi(x, t)$, where x is the distance from the heated surface into the halfspace and t is time since start of heating:

$$\phi(x, t) = (T(x, t) - T(x, 0)) / (T_F - T(x, 0)), \tag{7}$$

In terms of the thermal diffusivity of the moist fuel, α, and a characteristic time, τ, where

$$\tau = k \rho C / h^2, \tag{8}$$

$$\phi = 1 - erfc(u) + exp((t/\tau)^{1/2}(u + (t/\tau)^{1/2})) \\ erfc(u + (t/\tau)^{1/2}) \tag{9}$$

where $erfc$ is the complementary error function (Abramowitz and Stegun 1972),

$$u = x / (\alpha t)^{1/2}, \tag{10}$$

and

$$\alpha = k / \rho C. \tag{11}$$

Thermophysical properties (thermal conductivity, k; mass density, ρ; and specific heat capacity, C) of moist wood are modeled following Simms and Law

(1967). In the initial phase, where $T(x, 0)$ is assumed to be T_a, we use

$$k = k_o + 4.27 \times 10^{-4} \rho_o M \qquad (12)$$
and $$\rho C = \rho_o (C_o + C_w M) \qquad (13)$$

where subscript o implies ovendry and w, water. SI units are used. In the second heating phase $T(x, 0)$ is taken to be T_d, the temperature at the start of fuel drying, and C_w is increased to account for the latent heat of vaporization of the fuel moisture.

This formulation differs from the one-step process used by Simms and Law, in which they compute the temperature rise of the hypothetical fuel material from ambient to ignition. Doing so clearly overestimates the rate of heat transfer to the fuel after the surface reaches drying temperature, but it accounts for the energy needed to heat the fuel from ambient to ignition. Our formulation reduces the heat transfer rate to the fuel surface in the second phase, but artificially adds heat to some of the fuel to bring it to drying temperature. Note that the ejection of steam, the reduction of heat transfer to the fuel because of this, and the transport of heat by steam are ignored in both formulations.

Using an approximate form for the complementary error function (Hastings 1955) and — partly for historical reasons — using the effective Biot number (Bi_{eff}) and the Fourier number (Fo) for heating of a cylinder,

$$Bi_{eff} = h_{eff} D / k \qquad (14)$$
$$Fo = 4 \alpha t / D^2 \qquad (15)$$
$$t / \tau = Fo \, Bi^2_{eff} / 4 \qquad (16)$$

equation (9) is solved for $\phi(0, t)$ by binary search, to determine the time from the start of heating until surface drying begins, and from the start of surface drying until the surface achieves ignition temperature. These two times are added together to give a prediction of the ignition time delay.

Predicted ignition time delays were compared to ignition time delays observed in laboratory trials for 75 fuel specimens. In the laboratory experiments, individual fuel specimens were suspended in flame over a propane burner (see Figure 1) and the time at which flames were attached fairly continuously over the length of the specimen was recorded. The burner was a roughly square grid of pipes about 1 m on a side. The pipes were drilled with 0.18 cm holes about every 7.5 cm. A pressure regulator with output pressure preset to 1/4 psig (about 1.7 kPa excess pressure) controlled the propane flow to the burner, producing a fairly steady flame about 1 m high and burning about 7 gm/s of propane.

Fuels were selected to be representative of fuels burning in wildland fires, and consisted of limbs and boles of lodgepole pine between approximately 2.5 and 10 cm in diameter, sound and rotten, with and without bark, and at a range of moisture contents.

Preburn moisture content was sampled by sawing, weighing, and ovendrying a disk from either end of the fuel specimen. Fuel density was estimated by assuming the pieces to be cylinders of known diameter, length, weight, and moisture content.

Fuels were suspended approximately 70 cm above the propane burner, where they were immersed in the flame. In 23 of the experiments, a thermocouple embedded in a sphere of Sauereisen cement about 2 cm in diameter was suspended at the same height, and temperature recorded at one second intervals. This temperature reached a nearly constant value after about 3 minutes. This temperature was used to characterize the fire environment, and is called the "fire environment temperature", T_F. The average value for the 23 tests was 928K, with a standard deviation of 33K. For trials in which T_F was not measured, the average value was used in the analysis. Time until ignition was estimated ocularly as the time at which flame had attached along much of the length of the sample. Observed ignition delays were then regressed against predicted delays to derive an equation for use in the large fuel burnout model.

Three experiments were discarded as outliers. Two of them were replications in which four specimens were cut from the same tree stem. The ignition times observed were 26, 47, 127, and 228 seconds, while specimens of twice their diameters with greater moisture contents ignited in 45, 64, 65, and 139 seconds respectively. None of the eight specimens had bark. Obviously, undetected differences delayed the ignition of two of the smaller specimens uncharacteristically, and to have included them in the data set would have distorted the derived relationships from what is a reasonably well supported trend line. The other experiment culled from the data set was a replication in which two specimens were cut from one piece of limbwood with bark on. One ignited in a leisurely 41 seconds (about twice the time that similar specimens required), while the other delayed for 89 seconds. Again, at least this one specimen was atypical for undetected reasons.

With the exception of these three data points, observed ignition delay was well correlated with the predicted value, whether the prediction was assumed to be error free or not. The regression equation was of the form:
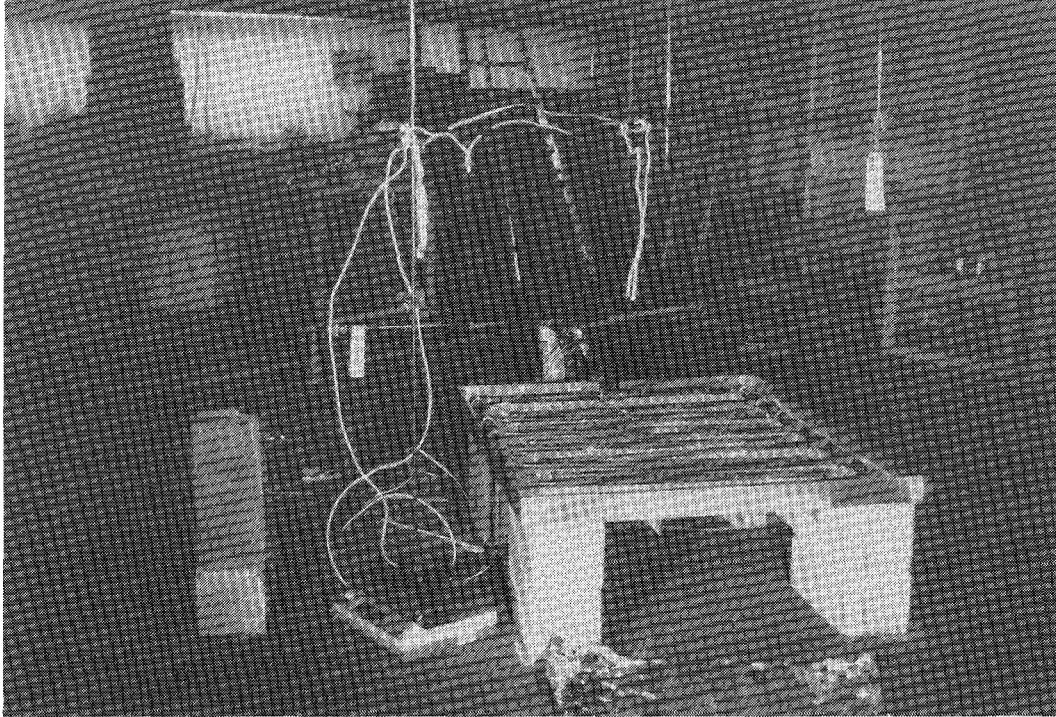
Figure 1. Experimental apparatus used in ignition time and burning rate measurements. A charred fuel specimen rests in two hooks suspended from a crossbar clamped to two stands that rest on digital readout scales. The fuel specimen and fire environment temperature sensor (the small dark sphere suspended by two insulated thermocouple leads) are suspended 60-70 cm above the array of pipes that feed a propane flame about 1 m^3 in volume.

Observed delay $(s) = a + b \, x$ Predicted delay (s)

and the parameter values found were:

$$a = 7.77 \; , \; b = 0.455 \; , \; r^2 = 0.682$$

using analysis that assumes the predictions to be error-free. In this case the 95% confidence intervals are (Ryan 1989):

$$4.72 < a < 10.82 \; ; \; 0.398 < b < 0.512$$

Assuming that the predictions are also subject to error, one can find the straight line that passes closest (in the least sum squared error sense) to the scatterplot points and derive

$$a = 6.33 \; , \; b = 0.494 \; , \; r^2 = 0.677$$

where r^2 is defined as the difference between unity and the ratio of the variance of the observations about the model to their variance about the mean. In this case, confidence intervals on the parameters are more difficult to derive, but it seems that a very small intercept (a) and a slope (b) near 0.5 would be a generally good descriptor of the scatterplot pattern.

We simplified the algorithm to use $a = 0$, $b = 0.5$, which approximates the regression equations adequately for the present purpose, and avoids the logical inconsistency that a vanishingly thin fuel element would still require six to eight seconds to ignite. Figures 2 - 4 show the scatter of predicted and observed ignition delay times for specimens with and without bark, and for all specimens combined. A line through the origin with slope 0.5 describes the data set in Figure 4 about as well as most natural fire phenomena are described by simple equations.

The slope (b) should be less than unity for this correlation because the analysis is done for a one dimensional problem in which half the universe is filled with solid wood and the heat source uniformly warms the exposed surface. The experiments are much closer to a two dimensional idealization in which an infinite cylinder is uniformly heated on its outer surface. The cylinder clearly should show a surface temperature rise rate greater than the halfspace. We speculate that the factor 0.5 is to be expected from detailed computation, but we have not attempted to derive it.
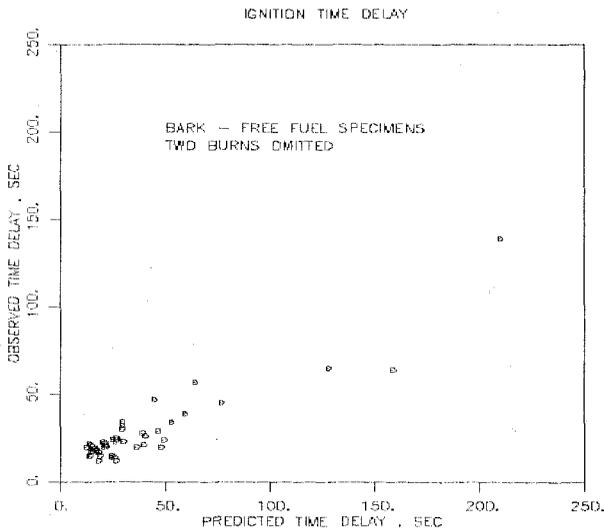
Figure 2. Scatter plot of predicted and observed ignition delay times for fuel specimens without bark. Two "fliers" were culled from the data set.
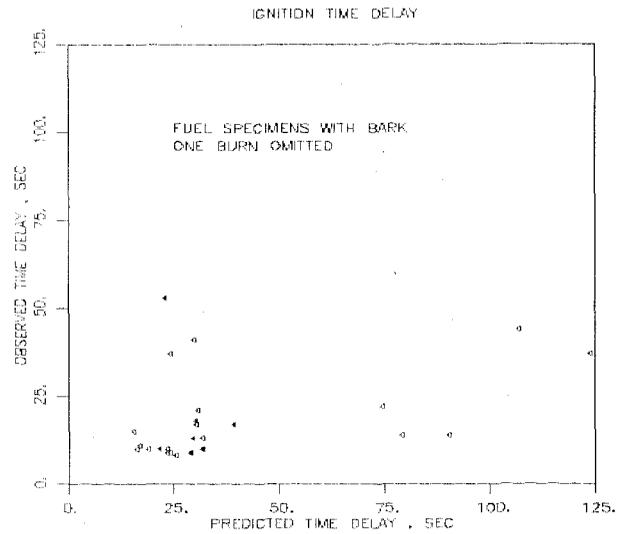
Figure 3. Scatter plot of predicted and observed ignition delay times for fuel specimens with bark. One "flier" was culled from the data set.
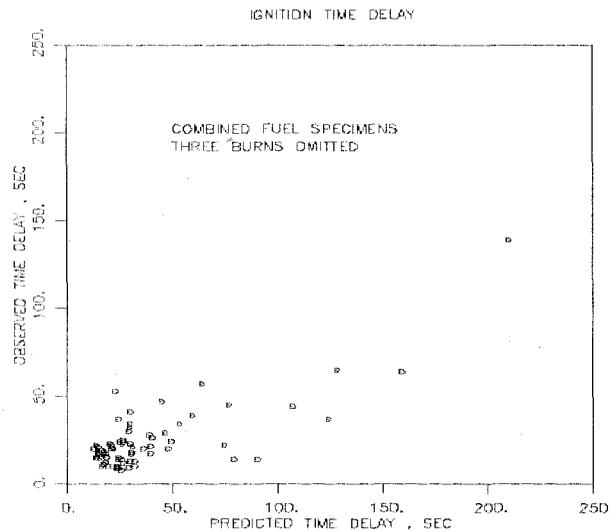
Figure 4. Scatter plot of predicted and observed ignition delay times for all data except for three culled experiments. A line through the origin with slope 0.5 describes these data well enough to provide a useable predictive model for ignition delay time.

Burning Rate

Assuming a fuel element to be a cylinder held horizontally and ignited over its entire surface, it will burn at a rate that depends upon the environment in which it is placed. If it is small enough in diameter and dry enough, it will burn in isolation by flaming combustion at approximately a steady rate until only char remains. A common wooden match does this.

But if the element is too large or too moist, it will not burn if heated only by its own combustion. An isolated section of limbwood a few cm in diameter, for example, will cease to burn even if ovendry, and a match may ignite but not continue to burn if it is saturated with moisture.

Like the ignition process, the burning of solid fuels by flaming combustion has been the subject of increasingly detailed analyses and numerical simulation modeling efforts over the years (Kansa, Perlee, and Chaiken 1977; Lee 1978; Petrella 1980; Holve and Kanury 1982; Phillips and Becker 1982; Wichman and Atreya 1987). But the fundamentals of the process can be captured roughly by a simple heat balance. This balance equates the rate at which fuel mass is raised to the burning temperature (including heating and vaporization of fuel moisture) to the rate of heat transfer.

Modeling this process as simple sublimation (which it clearly is not) at a temperature, T_c, approximately that at which the peak rate of pyrolysis occurs (Susott 1982, 1984) gives the following simple model for the rate of diameter reduction of a long cylindrical fuel element:

$$dD/dt = -h_{eff}(T_F - T_c)/\rho_o(Q_o + M Q_w) \qquad (17)$$

Here Q_o is the heat required to raise the temperature of a unit mass of ovendry wood from T_a to T_c and Q_w the heat required to evaporate a unit mass of water initially at T_a. The sublimation temperature, T_c, should probably be considered to be different for sound and rotten wood (Susott 1982), but our limited experience to date suggests that about 673K (400°C) works equally well for sound and rotten fuel within the confines of this simple model.

To extract a model equation from experimental data, we expressed equation (17) in integrated form, neglecting the small change in h_{eff} with time:

$$D(t) = D(0)(1 - t/T_c) \qquad (18)$$

Here the characteristic burning time for the fuel element, T_c, is to be found from data fitted to the form, found by equating the righthand side of (17) to the derivative of (18) and factoring out average values for $T_F - T_c$ and ρ_o:

$$T_c = ((928 - 673)/(T_F - T_c))(\rho_o/446) \\ (D(0)/h_{eff})(a' + b' M) \qquad (19)$$

The quantity 446 is the mean ovendry mass density (kg/ m³) of the fuel specimens used in our experiments, 928K is the mean fire environment temperature as estimated from the 23 experiments in which T_F was measured, $D(0)$ the initial fuel element diameter, and M the fuel moisture fraction.

We found the coefficients a' and b' in equation (19) in a two step process. First we found the value of T_c that best described the weight loss history of each fuel specimen, by ignoring the ignition time delay and fitting the functional form implied by equation (18) to the data. This form yields the fractional weight loss of a fuel specimen in our laboratory burner fire as

$$(W(0) - W(t))/W(0) = 1 - (1 - t/T_c)^2 \qquad (20)$$

We fitted fractional weight losses to this form by selecting the value of T_c that minimized the sum of squares of the prediction errors of all the weight samples for each specimen. The weight measurements were corrected to eliminate the bias (an instantaneous apparent weight loss

coinciding with ignition of the burner) resulting from the lift force exerted by the flame gases upon the fuel specimen.

Weight was recorded at one second intervals for each fuel specimen as it ignited and burned. The weight was sampled by standing the supports for a crossbar from which the fuel was suspended on two Mettler scales. Tare weights recorded by the two scales were zeroed before the fuel was suspended. The sum of the weights recorded by the two scales at each second was the remaining weight of the fuel specimen, and weight loss was the difference between the initial specimen weight and the weight at each time step.

Once the best value of T_c was established for each specimen, we performed a constrained regression of T_c against derived parameters x and y. Using (19) to identify the two parameters x and y as

$$x = 255(\rho_o/446)(D(0)/(h_{eff}(T_F - 673))) \qquad (21)$$

and

$$y = M x \qquad (22)$$

we express (19) in the form to be fitted to those values of T_c that were found to describe best the experimental data:

$$T_c = a'' x + b'' y \qquad (23)$$

In this case, we determined a'' and b'' that defined the surface passing closest in the least squares to the points (x, y, T_c) of a three dimensional scatter plot, because all three coordinates are subject to error. Comparing equation (23), with derived values of a'' and b'', term by term to equation (19) allows us to calculate the values of a' and b' in equation (19).

Values of a' and b' were determined for fuel specimens with and without bark cover separately, and for all fuel specimens combined and treated the same:

48 bark free samples: $a' = 2.06 \times 10^6$, $b' = 1.50 \times 10^6$, $r^2 = 0.89$
32 samples with bark: $a' = 2.03 \times 10^6$, $b' = 5.95 \times 10^6$, $r^2 = 0.97$
80 samples combined: $a' = 2.01 \times 10^6$, $b' = 3.36 \times 10^6$, $r^2 = 0.77$

where we have again used the generalized definition of r^2 to derive this measure of the quality of the data description by the model equation.

Because of the way the regression coefficients were defined, the units of a' and b' are J-m³-K, assuming that h_{eff} is expressed in W/m²-K and D in m. For simplicity we used the values from the combined data set for the model.

Using this formulation, the rate of "shrinkage" of a fuel element's diameter is to be predicted using the form:

$$dD/dt = h_{eff}((T_F - T_c)/255)(446/\rho_o)/(a' + b'M) \quad (24)$$

where T_c, ρ_o, and M are considered constant for the fuel component, but the fire environment temperature, T_F and h_{eff} can be expected to vary with time. The laboratory fires are simple in the sense that T_F can be considered to be constant because the gas flow rate to the burner is regulated and varies only gradually with time, and h_{eff} can be considered constant because it depends only weakly upon D, being dominated by radiation in most cases.

Figures 5 - 7 illustrate how well the predicted fractional weight loss histories fit typical data from our experiments. To make these predictions, equation (19) was used to determine a value for T_c, then equation (20) used to calculate the fractional weight loss as a function of time. In these figures the experimental data are plotted as the "ragged" curve, which exhibits the random errors in weight measurements. The empirical curve with the form of equation (20) and the value of T_c chosen to fit the specific sample data best (in the least squares sense) is plotted as the third curve in each figure. It is our observation that the best-fit values of T_c describe the measurement data quite well, and the model-predicted values of T_c perform adequately over the range of data taken in these experiments.
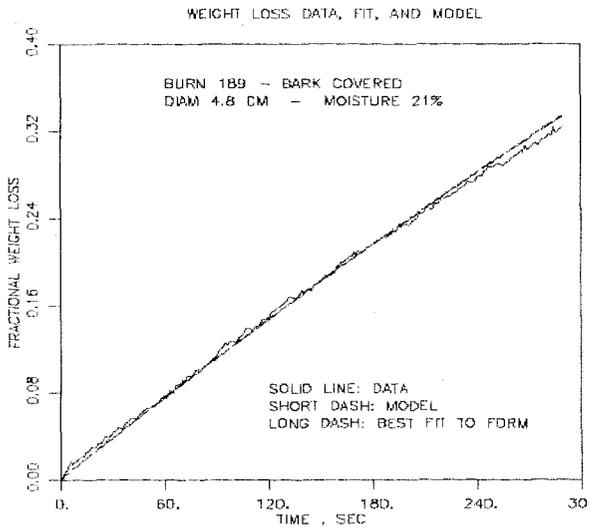

Figure 5. Fractional weight loss history of bark covered fuel specimen 189. The data fit well the form of equation (20), using either the value of T_c that best fits these data or the value calculated from equation (19).
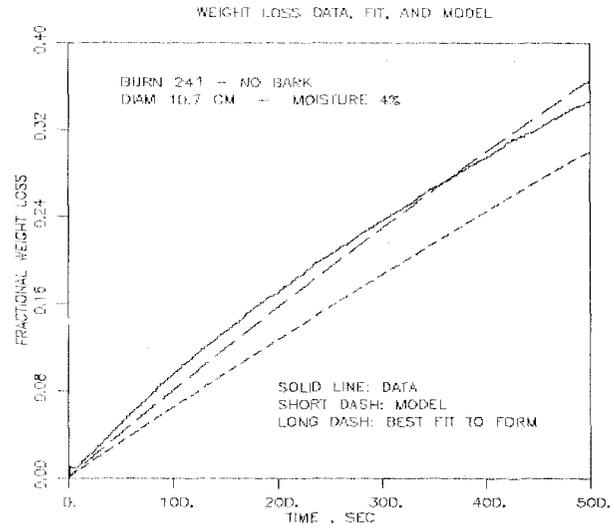
Figure 6. Fractional weight loss history of fuel specimen 241, which had no bark. Burn rate was underpredicted by perhaps 20%.
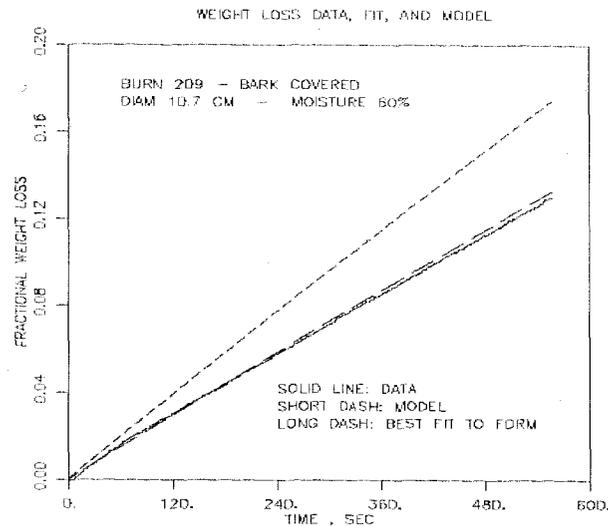

Figure 7. Fractional weight loss history of bark covered fuel specimen 209. Burn rate for this specimen was overestimated by about 30%.

Figure 5 is for a fuel specimen with 21% moisture content, which is in the middle of our experimental data range. The agreement between predicted and observed sample weight loss rates is typical of the "good" fits found in this moisture range. Figure 6 is for a very dry fuel specimen with 4% moisture content, and the model underpredicts the weight loss at 500 s by about 20%. The specimen of Figure 7 has 60% moisture content, near the maximum for our experiments. In this case the model overpredicts the weight loss by perhaps as much as 30% at about 600 s. We believe this level of performance to be adequate for the intended uses of these models.

Summary

Research is under way to develop a model for predicting intensity histories of wildland fires involving the persistent burning of large woody fuels and for predicting the ultimate quantity of each fuel component that will be burned. Such a model is intended for use in assessing the severity of potential fire control problems in accumulations of large woody fuels, and as an aid to prediction or correlation of the effects of fires in such fuels. As both intensity history and ultimate fuel consumption depend upon the type, quantity, condition, and arrangement of the fuels, this model will of necessity deal with several interconnected aspects of the overall process. Here we report on the formulation and calibration of two component parts of this model.

One component predicts the time delay between the immersion of a moist woody cylinder into a constant fire environment until the cylinder ignites. The ignition event was identified in laboratory experiments using a propane burner as a surrogate for a wildland fire, as the attachment of flame over the entire surface of a cylindrical natural fuel specimen. The other component model predicts the weight loss history of a large moist woody fuel cylinder immersed in a steady fire environment. In both components, fire environment is quantified by the equilibrium temperature of a small inert object placed in the flame volume at the location of the fuel specimen.

It is intended that these components be combined with others to create a model that can be used for the purposes described above. In this application, we take account of the fact that fire intensity (hence fire environment temperature, T_F) is neither constant in time nor uniform over the fire area. These factors will be addressed in a forthcoming paper describing calibration and testing of the model using laboratory burns of cribs made with large fuel components and using field data from prescribed burns of harvest debris.

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