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ABSTRACT

The goal of the research presented in this paper was to find a simple means for monitoring the heat output of a wood stove. To accomplish this, general engineering models of heat transfer are used to develop a model that predicts the heat output of a stove from measurements of surface temperature. Using the surface area and the measured surface temperature as inputs, the model predicts the heat output of the stove by radiation and natural convection. As a means of verification, surface temperature data from four wood stoves monitored in a calorimeter room are used to make heat output predictions. These predicted heat outputs are then compared with the actual heat outputs measured by the calorimeter room. The predictions involve several potential monitoring schemes: 1) separate temperature measurements for each surface of the stove, 2) an average temperature measurement for all stove surfaces, and 3) a single surface temperature measurement. The accuracies of the predictions are characterized by their geometric bias and scatter, as well as their predictions of total energy delivered. The scatter is a measure of the trackability of the model, analogous to the arithmetic standard deviation. Predictions made from average temperature measurements are found to be as accurate as those based on individual temperature measurements, whereas single-temperature measurements cause an additional 5% uncertainty in predictions. For both the average temperature and individual temperature predictions, the bias is between 2% and 24%, with 16% as the typical scatter. The trend in the bias is underprediction, possible causes of which are discussed at length.

NOMENCLATURE

A_a	is the surface area of the ambient surroundings [m^2 (ft^2)],
A_s	is the surface area [m^2 (ft^2)],
c_p	is the specific heat of the fluid [J/kg K (Btu/lbm F)],
g	is the acceleration of gravity $[m/s^2\ (ft/s^2)]$,
Gr	is the Grashof number [dimensionless],
$\overline{h}_c$	is the unit surface conductance $[W/m^2K\ (Btu/h\ \ ft^2F)]$,
k	is the conductivity of the fluid $[W/mK\ (Btu/h\ \ ft\ \ F)]$,,
K	is an empirical constant [dimensionless],
K'	is a dimensional constant that includes all of the constants in equation 7, [15.9 W/$m^2K^{0.92}$ (2.94 Btu/h$ft^2R^{0.92}$)],,
L	is the characteristic dimension of the surface [m (ft)],
Nu	is the Nusselt number [dimensionless],
Pr	is the Prandtl number [dimensionless],
Q_r	is the radiant heat flow from the surface [W (Btu/h)],
Q	is the total heat flow from the surface [W (Btu/h)],
Ra	is the Rayleigh number (GrPr) [dimensionless],
T_a	is the absolute temperature of the ambient surroundings [K (R)],
T_s	is the absolute temperature of the surface [K (R)],
β	is the coefficient of thermal expansion of the fluid [1/K (1/R)],
ϵ_a	is the emittance of the ambient surrounding surfaces [dimensionless],
ϵ_s	is the emittance of the surface [dimensionless],
ρ	is the density of the fluid $[kg/m^3\ (lbm/ft^3)]$,
σ	is the Stefan-Boltzmann constant, 5.67 x 10^{-8} $[W/m^2K^4]$ (0.1714 x 10^{-8} $[Btu/h\ \ ft^2R^4]$),
μ	is the viscosity of the fluid [kg/m s (lbm/ft s)].

INTRODUCTION

As a means of quantifying the success of energy conserving building designs and retrofits, as well as to validate computer energy simulation models, researchers have attempted to monitor the overall energy performance of residences[1,2]. However, because over 40% of single family residences use wood-burning appliances[3,4], a major obstacle to such efforts has been the difficulty of accurately determining the energy contribution of wood heating[5-7]. Even when a reliable efficiency rating for a wood-burning appliance is available from the manufacturer, monitoring the amount of wood burned is unacceptable. The heat content of wood varies widely with moisture content and stove efficiencies vary with operating conditions. In addition, significant occupant cooperation is required to accurately monitor the wood consumption.

Although there is a large body of research relating to wood-burning appliances, this work has focused almost exclusively on improvements in wood appliance performance, reduction of emissions, and techniques for determining thermal efficiency[8-11]. The problem of developing a simple technique for measuring the heat output of a wood-burning appliance in the field has not received as much attention[12]. The goal of the research presented in this paper was to find a simple means of monitoring the heat output of a wood stove.

In an initial investigation of the data analyzed in this report (reference 12), a simple correlation was found between the heat output of a wood stove and its surface temperature or radiant heat flux. From that study it was concluded that monitoring a wood stove with a single data acquisition channel was both possible and practical. It demonstrated that a single-parameter correlation based on either surface temperature or radiant heat flux measurements could predict the full-cycle (start-up to cool-down) heat output of a wood stove to within 20% of the actual output. The major drawback of that study, however, was that it did not satisfactorily address the issue of extending the correlations to stoves that had not been tested. The research presented in this report is an attempt to find a general model for predicting the heat output of a wood stove using only the stove surface area and emissivity to characterize the stove.

THEORY AND APPLICATION

The model developed here can be used to predict the heat output of wood stoves that transfer their heat to the surroundings by radiation and natural convection. That is, it is presently not applicable to wood stoves fitted with forced-convection blowers.

For a wood stove without a forced-convection blower, as for any other body that exchanges heat with its surroundings by radiation and natural convection, it is possible to determine the energy-exchange rate directly from the surface temperature and the temperature of the surroundings. The model described below is a simplification of general engineering models of radiation and natural convection for the particular case of a wood stove.

Radiation

As is well known, the quantity of energy leaving a surface as radiant heat depends on the absolute temperature and the nature of the surface. In this analysis of wood stoves, several assumptions are made about the nature of the surfaces involved. First, all surfaces are treated as gray bodies; that is, the rate of radiant heat emission is related to that of a perfect radiator, or blackbody, by a single constant coefficient. Each surface is thus characterized by its emittance, ϵ, the ratio of its emission to that of a blackbody. It is also assumed that the surface of the stove does not radiate to itself, but rather, exchanges radiation only with the surrounding room. Incorporating these approximations into the radiative heat exchange equation, the net rate of heat transfer from a stove surface can be expressed as:

$$Q_r = \frac{A_s \, \sigma \, (T_s^4 - T_a^4)}{\dfrac{1 - \epsilon_s}{\epsilon_s} + 1 + \dfrac{A_s (1 - \epsilon_a)}{A_a \, \epsilon_a}} \tag{1}$$

Further assuming that the area of the surroundings (A_a) is much larger than the area of the stove (A_s), equation 1 reduces to:

$$Q_r = \epsilon_s \, A_s \, \sigma \, (T_s^4 - T_a^4) \tag{2}$$

In equation 2, the radiation from a stove surface is characterized with just one parameter, the emittance of the surface. In fact, the emittance of the surfaces of a typical wood-burning stove is known to vary within a small range, between 0.85 and 0.95.

Free Convection

Free convective heat transfer occurs whenever a body is immersed in a fluid at a temperature different from that of the body. In the case of a wood stove, cool air is drawn to the stove surface as heated air is carried away by buoyant forces. The important characteristic of this type of heat transfer is that the mass transfer rate (i.e., fluid flow) is a function of the temperature difference. Because of this relationship, the rate of heat removal from the stove surface does not vary linearly with the temperature difference between the stove and the surrounding fluid.

The most common engineering approach to the problem of predicting heat transfer by free convection has been to use similarity parameters to obtain a simplified expression for the unit surface conductance (i.e., convective heat transfer coefficient). Similarity parameters are dimensionless ratios used to characterize the underlying mechanisms behind a physical process. For free convection, the parameter for establishing dynamic similarity is the ratio of buoyant to viscous forces, or the Grashof number. As a convenient means for determining the unit surface conductance, $\overline{h}_c$, for different applications, empirical relationships between the Grashof number and other pertinent similarity parameters have been developed. [13,14] These relationships include two additional similarity parameters, the Nusselt number and the Prandtl number. The three parameters, the Grashof number (Gr), Nusselt number (Nu), and Prandtl number (Pr) are defined as:

$$Gr = \frac{\rho^2 g \ \beta (T_s - T_a) L^3}{\mu^2}$$

$$Nu = \frac{\overline{h}_c \ L}{k} \tag{3}$$

$$Pr = \frac{c_p \ \mu}{k}$$

For the case of wood stoves, the pertinent empirical relationships are those developed for vertical and horizontal surfaces. These empirical relationships have been determined experimentally for both laminar and turbulent flow regimes. In the laminar regime, the relationship between the three dimensionless parameters is

$$Nu = K (Gr \ Pr)^{\frac{1}{4}} \tag{4}$$

whereas in the turbulent regime the relationship is:

$$Nu = K(Gr\ Pr)^{\frac{1}{3}} \tag{5}$$

As with most flow phenomena, the transition between these two flow regimes is not clear-cut. For vertical plates, the turbulent flow relationship is recommended for Rayleigh numbers above 10^9, where the Rayleigh number is the product of the Grashof and Prandtl numbers. For heated horizontal plates facing upward the turbulent flow relationship is recommended [15] for Rayleigh numbers above 2 x10^7. Table 1 presents computations of the Rayleigh number for some typical wood stove surface temperatures.

TABLE 1

Rayleigh Numbers for a Range of Surface Temperatures of Wood-Burning Stoves

Stove Surface Temperature [°C (°F)]	Mean Air Temperature[a] [°C (°F)]	Rayleigh Number[b] [Dimensionless]
400 (752)	212.5 (414.5)	1.52 x 10^9
300 (572)	162.5 (324.5)	1.65 x 10^9
200 (392)	112.5 (234.5)	1.66 x 10^9
100 (212)	62.5 (144.5)	1.18 x 10^9
75 (167)	50.0 (122.0)	0.87 x 10^9
50 (122)	37.5 (99.5)	0.47 x 10^9

[a] The average of the stove and air temperatures, $(T_s+T_a)/2$, where the ambient air temperature is assumed to be 25°C (77°F).

[b] Based on a typical surface length of 0.6 m (2 ft), and evaluating the fluid properties at the mean air temperature, except for β, which is evaluated at air temperature, 25°C (77°F).

Examination of Table 1 shows that the Rayleigh numbers for free convective flows around wood stoves remain relatively constant over the entire operating temperature range, only dropping below 10^9 at the lowest operating temperatures (about 75°C(167°F)). These Rayleigh numbers are in the transition region between laminar and turbulent flow for vertical surfaces and well into the turbulent regime for heated horizontal surfaces facing upward. Given that the

Rayleigh number is so stable, and that it is in the transition region for vertical surfaces, either the turbulent or laminar relationship, or even a single-point relationship between the similarity parameters, could be used to determine the unit surface conductance for the vertical surfaces of a wood stove. Based on the fact that the unit surface conductance is independent of surface length in the turbulent relationship, and on the fact that the horizontal surface on top of the stove is clearly in the turbulent flow regime, the turbulent relationship was chosen to model free convection around both vertical and horizontal surfaces of wood stoves (Note: As explained below, heat transfer from the bottom of the stove, which is probably in the laminar flow regime, is not included in the predictions of total heat output).

The empirical relationships between the similarity parameters remain to be solved for the unit surface conductance of vertical and horizontal plates. The two relationships (equations 4 and 5) are essentially identical, differing only slightly in the value of K, 0.13 for vertical plates and 0.14 for horizontal plates (reference 14). Because wood stoves have mostly vertical surfaces, the vertical plate relationship was chosen to predict convective heat transfer. Although this approximation causes a 7% underprediction of the convective heat transfer from the stove top, as shown below, the underprediction of the total heat output of the stove is less than 1%. Inserting the definitions in equation 3 into equation 5, and solving for the unit surface conductance, yields:

$$\bar{h}_c = K \frac{k}{L}\left(\frac{c_p \rho^2 g \beta (T_s - T_a) L^3}{\mu k}\right)^{\frac{1}{3}} \tag{6}$$

The length, L, drops out of the equation and leaves:

$$\bar{h}_c = K \left(\frac{k^2 \rho^2 c_p g \beta}{\mu}\right)^{\frac{1}{3}} (T_s - T_a)^{\frac{1}{3}} \tag{7}$$

In principle, equation 7 could be used directly to compute the unit surface conductance of a stove, which depends only on the temperature difference between the stove and the surrounding air, some physical properties of air, and an empirical constant. However, the physical properties of air in equation 7 vary significantly with temperature. Because the surface temperatures of a wood stove vary over a wide range, 50-400°C (120-750°F), the temperature dependence of the air properties should be incorporated into equation 7 when it is applied to wood stoves.

The problem of variable fluid properties is discussed at length in Reference [16]. Although the treatment in this reference is strictly valid for laminar free convection, the results should apply as well to turbulent free convection. The authors show that the fluid properties (except β) should be evaluated at an empirically determined reference temperature, but that evaluation of the properties at the film temperature, $(T_{\text{ambient}}+T_{\text{surface}})/2$, provides essentially the same results. It is also shown that β should be evaluated at the ambient air temperature.

Applying these results to equation 7, the film temperature will be used for evaluating all properties except β. The temperature dependence of the density, ρ, is determined directly from the ideal gas equation. The coefficient of thermal expansion, β, is evaluated at the ambient air temperature in the house, and can thus be treated as a constant. The specific heat, c_p, is essentially independent of temperature, and, in the temperature range of interest here, the temperature dependence of the conductivity (k) and viscosity (μ) can be well represented by power-law fits with absolute temperature (data obtained from appendix III in reference 14). Power-law fits performed for the temperature range of interest showed that the viscosity scales with the absolute temperature to the 0.72 power, and the conductivity scales with the absolute temperature to the 0.75 power. Incorporating these relationships into equation 7, a numerical expression for the unit surface conductance as a function of the film temperature $((T_s+T_a)/2)$ and the stove-air temperature difference is obtained:

$$\bar{h}_c = K' \frac{(T_s - T_a)^{\frac{1}{3}}}{\left(\frac{T_s + T_a}{2}\right)^{0.41}} \qquad (8)$$

The variation in unit surface conductance with stove surface temperature is presented in Table 2. To examine the importance of the temperature variation in air properties, the unit surface conductance computed with equation 8, which includes the temperature variation, and the unit surface conductance computed with equation 7 using air air properties at 93.3°C (200°F) (corresponding to a stove surface temperature of 162°C (324°F)) are compared in Table 2.

TABLE 2

Unit Surface Conductance for a Range of Stove Surface Temperatures

Stove Surface Temperature [°C (°F)]	Mean Air Temperature [°C (°F)]	$\bar{h}_c^a$ [W/m^2 K]	$\bar{h}_c^b$ [W/m^2 K]
400 (752)	212.5 (414.5)	9.10	10.34
300 (572)	162.5 (324.5)	8.58	9.22
200 (392)	112.5 (234.5)	7.76	7.94
100 (212)	62.5 (144.5)	6.19	5.98
75 (167)	50.0 (122.0)	5.50	5.23
50 (122)	37.5 (99.5)	4.43	4.15

[a] Computed with Equation 8 (includes temperature variation of properties).

[b] Computed with Equation 7, using air properties at 93.3°C (200°F), corresponding to a stove surface temperature of 162°C (324°F), and computing β at 25°C (77°F).

At a stove temperature of 400°C (752°F), the error associated with using equation 7 with constant properties is 13%, implying a 13% overprediction of the convective heat output of the stove. As the advantages of using equation 7 are slight, and the errors can be large, equation 8 was used to model free convective heat transfer from a wood stove.

Wood Stove Model

The simplified models developed for radiative and free convective heat transfer from wood stove surfaces were combined into a single model for predicting the total heat output of a wood stove. Combining equations 2 and 8 gives a model that predicts the heat output of each stove surface from the surface temperature and the temperature of the surrounding air and walls, using just the stove surface area and emissivity as input parameters:

$$Q = A_s \left[\epsilon_s \, \sigma (T_s^4 - T_a^4) + K' \, \frac{(T_s - T_a)^{\frac{4}{3}}}{\left(\frac{T_s + T_a}{2} \right)^{0.41}} \right] \tag{9}$$

Applying this model to monitoring wood stoves in houses can be approached several ways: 1) the temperature of each stove surface could be separately monitored, 2) an average surface temperature for the stove could be monitored, or 3) a single surface temperature considered representative of the stove could be monitored. These strategies are listed in order of decreasing complexity, the latter two requiring only a single channel for monitoring the output of a stove.

RESULTS

The four criteria chosen for evaluating the success of the three proposed monitoring strategies are: 1) that they be able to quantify the total amount of energy that a wood stove contributes to the living space, 2) that they be able to track the heat output of the stove throughout most of its burn cycle, 3) that they be applicable to a large number of stoves, and 4) that they be simple and inexpensive to implement. The first requirement is the most important; the second is important for monitoring designed to track short-term heating load or indoor temperature variations; the third and fourth requirements are crucial for any large-scale field monitoring.

Measured surface temperatures and heat outputs of four wood stoves were used to compare the monitoring strategies. The measurements were made by operating each of the stoves in a calorimeter room (reference 12) and monitoring the surface temperatures and heat output at two-minute intervals during the entire burn cycle. The calorimeter room, shown in figure 1, was designed to have quick thermal response, thereby allowing the fluctuations in heat output to be accurately monitored. Predictions of the heat output of each stove were made from surface temperature measurements of five stove surfaces (excluding the stove bottom) and the stove-pipe. The first monitoring strategy was examined by computing the heat output of each stove surface with equation 9 and adding the individual heat flows to get the total heat output. The second monitoring strategy was simulated by computing an unweighted average surface temperature from the six measured temperatures (five surfaces plus the stove pipe) and using this temperature, along with the total surface area of the stove and stove pipe, to compute the heat output with equation 9. The third monitoring strategy was examined by using the temperature of a single surface along with the total surface area to predict the heat output.

The accuracy of the predictions are expressed as the *bias* and *scatter* of the predictions relative to the measured heat outputs. The bias can be interpreted as the expected long-term trend in model predictions (i.e., does the model consistently overpredict or underpredict the heat output?); the scatter is a measure of how well the model tracks the normal fluctuations in heat output that occur during a burn cycle. Because prediction errors are expected to scale with heat output, the computation of bias and scatter is based on a geometric (or log-normal) distribution. The prediction results for the first two monitoring strategies are presented in Table 3. For each test, the bias, defined as predicted heat output divided by measured heat output, and the scatter, the geometric equivalent of the arithmetic standard deviation, are determined for the period in which the stove provides 90-95% of the total heat output, thereby ignoring the very low power outputs at the end of each test.

TABLE 3
Comparisons of Bias and Scatter of Heat Output Predictions for Four Wood-Burning Stoves

Stove/Test	Test Length [h]	Average Heat Output [W]	Individual Prediction		Average Prediction	
			Bias	Scatter [%]	Bias	Scatter [%]
Stove A						
1	13	6200	.97	14	.97	11
2	15	2800	.95	9	.96	9
Stove B						
1	8	2800	.81	12	.87	10
2	15	1300	.90	15	.90	16
3	8	2900	.78	27	.79	24
4	8	3000	.70	36	.71	33
Stove C						
1	10	5100	.71	15	.83	15
2	12	1200	.82	12	1.03	12
Stove D						
1	12	5300	1.00	20	1.10	18
2	12	5100	.82	11	.94	10
3	11	3300	.88	11	1.00	10
4	10	3200	.87	19	.86	16
5	10	3900	1.00	14	1.00	11
Average	11	3500	.86	17	.92	15
Std. Dev.	2.4	1500	.10	7.6	.11	6.9

The results presented in Table 3 indicate that, despite a tendency to underpredict the heat output of every stove, both monitoring strategies predict wood stove heat output quite well. The predictions made from individual temperatures are approximately 14% low on average, whereas those made from temperature averages are approximately 8% low. On the other hand, the scatter for all predictions is quite good, indicating that both strategies are able to track fluctuations in heat output.

One surprising result in Table 3 is that predictions made from an average stove temperature are as good as predictions made from the individual temperatures for each surface. A possible explanation for this result is that the unweighted temperature average happens to bias the predictions in the correct direction. A more careful examination of the individual temperature measurements and surface areas tends to confirm this explanation. For all stoves, but especially for the two smallest stoves, the stove pipe is the largest single surface element, and, on average, has the lowest temperature. The implication is that predictions made with average temperature and total surface area should be larger than the summation of individual predictions. This fact, combined with the fact that the model tends to underpredict, suggests that the predictions from average temperatures should be closer to the measured heat output.

One disadvantage of using the geometric (or log-normal) bias to characterize the performance of a model is that this bias does not provide a measure of absolute error. Small absolute errors at small heat outputs can have a large effect on the log-normal bias. In other words, the log-normal bias may not be the best measure of how well the model predicts the total amount of energy that a wood stove contributes to the living space. For this reason, the integrated heat output (or total quantity of energy delivered) determined from the model predictions is also compared with the integrated measured heat output. This comparison provides a measure of the absolute error in the total heat delivered, which is the first criteria chosen for evaluating the success of a monitoring strategy. The results summarized in Table 4 are arrived at by dividing the predicted total energy delivered by the measured value.

TABLE 4
Comparison of Predicted and Measured Integrated Heat Output for Four Wood-Burning Stoves

Stove/Test	Test Length [h]	Total Heat Output [kWh]	Integrated Heat Output Ratio	
			Individual Prediction	Average Prediction
Stove A				
1	13	82	1.06	1.04
2	15	42	.97	.97
Stove B				
1	8	22	.89	.94
2	15	19	.93	.94
3	8	24	.88	.89
4	8	24	.85	.88
Stove C				
1	10	51	.71	.82
2	12	14	.83	1.03
Stove D				
1	12	64	1.06	1.17
2	12	61	.83	.96
3	11	36	.90	1.02
4	10	32	.96	.93
5	10	39	1.04	.97
Average	11	39	.92	.97
Std. Dev.	2.4	20	.10	.09

The results is Table 4 indicate that the individual temperature strategy underpredicts the total energy delivered by 8% on average, and that the average temperature strategy underpredicts by 3% on average. These results indicate that both strategies are better at estimating delivered energy than the log-normal biases would suggest.

Some comparisons of predicted and measured heat outputs are plotted in figures 2 through 4. In all three figures, the measured heat output and the heat outputs predicted from individual and average surface temperatures are plotted from data taken every two minutes throughout the course of the test. Figures 2 and 3, for the largest of the stoves, Stove A, show a close correspondence between the model predictions and the measured heat output, and also show that the individual temperature and the average temperature predictions are essentially equivalent. Figure 4, for the smallest stove, Stove C, presents an example of the worst predictions, where the predictions are approximately 30% lower than the measured output on average. Although the predictions appear to track the heat output quite well, they are consistently low, and the predictions from average temperatures are significantly better than those from individual temperatures.

For the third monitoring strategy (measuring the temperature of a single surface) to be an accurate alternative, a single surface whose heat output is representative of the heat output of the entire stove must be identified. One way to test for this possibility is to use each individual surface temperature along with the total stove surface area to predict the total heat output. The ratios of these heat output predictions to the measured heat outputs (i.e., the bias of the predictions) are summarized in Table 5.

TABLE 5

Bias of Heat Output Predictions made from Temperature Measurements on a Single Surface

Stove	Bias					
	Stove-pipe	Top	Right side	Left side	Front	Back
Stove A	0.38	1.35	0.83	1.20	1.01	1.08
Stove B	0.52	1.39	0.51	0.94	0.76	1.08
Stove C	0.27	1.47	0.65	1.22	0.76	1.58
Stove D	0.62	1.33	0.94	1.00	0.54	1.16
Average Std. Dev.	0.45 34%	1.39 4.5%	0.73 26%	1.09 13%	0.77 25%	1.23 19%

Although a bias of unity is desirable, a more important criteria for the selection of a single sensor measurement is the consistency of the bias. If a surface is consistently cooler or hotter than the rest of the stove, as long as the bias does not change from stove to stove, predictions made from that surface temperature could be corrected with the known bias. On the other hand, if a surface has an average bias of unity with a large standard deviation, the correct bias for an untested stove could not be determined a priori. The surface that meets these qualifications is the top of the stove, which consistently predicts the heat output to be 39% higher than measured heat output, with a standard deviation for the bias of less than 5%. Assuming that these stoves (and these measurements) are representative, measurements of the surface temperature of the top of a stove can be used to predict the total heat output with an additional uncertainty of only 5%.

DISCUSSION

The heat transfer model and monitoring strategies developed here appear to be capable of predicting, within 25% of measured values, the heat output of a wood-burning stove. Testing the accuracy of the predictions, however, revealed several uncertainties that suggest the need for further research.

One finding that warrants discussion, for example, is the consistent underprediction of the total heat output. Three explanations for this problem that come to mind are: 1) the heat transferred from the bottom of the stove, which was not included in any of the heat output predictions, 2) the installation of the sensors on the stoves, which may have caused conduction along the sensor wires, thereby lowering the measured temperatures, and 3) the locations of the sensors, which may or may not be representative of the average temperature on each surface.

The heat output from the bottom of the stove was not included in the predictions because temperature measurements were not made on that surface. In addition, the heat output from the stove bottom would be difficult to monitor in the field, because the distance between the stove and the floor, and therefore both the convective and radiative heat transfer, varies considerably between stoves. Because the temperature of the floor under the stove is significantly higher than that of the walls of the room, its temperature, as well as the temperature of the stove bottom would have to be separately monitored.

As an approximate means for including the heat transfer from the bottom of the stove, the heat output biases from Table 3 and the integrated heat output ratios from Table 4 were multiplied by the ratio of the stove area including the bottom to the stove area without the bottom. The resulting biases are summarized in Table 6.

TABLE 6

Bias of Heat Output Predictions Made by Including the Surface Area of the Bottom of the Stoves

Stove/Test	Individual Prediction		Average Prediction	
	Bias	Output Ratio	Bias	Output Ratio
Stove A				
1	1.16	1.26	1.16	1.24
2	1.13	1.16	1.14	1.16
Stove B				
1	.93	1.02	1.00	1.08
2	1.04	1.07	1.04	1.08
3	.90	1.01	.91	1.02
4	.81	.98	.82	1.01
Stove C				
1	.80	.80	.93	.92
2	.92	0.93	1.16	1.16
Stove D				
1	1.15	1.22	1.26	1.34
2	.94	.95	1.08	1.10
3	1.01	1.03	1.17	1.15
4	1.00	1.10	.99	1.07
5	1.15	1.19	1.15	1.11
Average	1.00	1.06	1.06	1.11
Std. Dev.	13%	12%	10%	12%

The average biases in Table 6 indicate that the inclusion of the surface area of the bottom of the stove generally makes the average heat output predictions higher than the measured values. Although these results cannot be considered conclusive, they do indicate that the accuracy of the predicted heat outputs can be improved by assuming that the stove bottom contributes some fraction of the total heat output. These results also indicate, as expected, that the stove bottom provides a smaller heat flux than the remainder of the stove. It should be noted that, at least in the laminar regime, the convective heat transfer coefficient for a heated surface facing downwards (such as the bottom of the stove) is approximately one half that of a surface facing upwards (reference 14).

With regard to the question of heat conduction along the wires leading to the thermocouple sensors, because the sensor is part of a heat flow path from the stove surface to the air (the wire acting as a heat transfer fin), it will measure a temperature lower than that of the stove surface. Although this problem can be minimized by insuring good thermal contact between the sensor wire and the stove, the sensor mounting procedure used for the calorimeter room tests was not tested for this effect.

The third possible reason for the underprediction, that the sensors do not measure temperatures representative of the surfaces they are monitoring, stems from the fact that temperature sensors on some surfaces were located at the center whereas others were located near edges or corners. The effects of sensor location can be seen in Table 5, where, although the stoves are symmetric, the predictions made with the sensors on the sides of the stoves are very different from each other. This result indicates that differences in sensor location have a significant effect on the heat output predictions in Table 3, where, if the left-side sensor measures a representative temperature, the right-side sensor would cause underprediction.

The two stoves with the worst heat output predictions in Table 3, stoves B and C, appear to have unusually asymmetric heat outputs. For both stoves the left-side sensor was approximately in the center of the surface, whereas the right-side sensor was at the corner of the surface. In Table 5, the right sides of both stoves appear to be providing only one-half the heat output of the left side. Although not enough data are available for stove C, (the left-side sensor was disconnected during run 2), the bias in the data for stove B is quite clear in all four tests. As a means of quantifying the size of this effect, predictions of heat output were made using the left-side (center) temperature for both surfaces. The results of this substitution are presented in Table 7.

TABLE 7

Bias and Scatter for Heat Output Predictions Using Left-side (Center) Temperature for Both Left and Right Surfaces

Stove/Test	Individual Prediction		Average Prediction	
	Bias	Scatter [%]	Bias	Scatter [%]
Stove B				
1	.86	12	.93	11
2	.97	16	.98	18
3	.82	28	.83	25
4	.79	32	.80	30

Comparison of the predictions in Table 7 and Table 3 shows an improvement of more than six percentage points in the bias of the heat-output predictions. This shift represents a significant improvement in prediction accuracy, and confirms that sensor location can have an important effect on heat output predictions.

In summary, although there is not enough evidence to choose one of the above possibilities as the source of the underprediction, they all move the predictions in the correct direction. Further experimentation could be used pinpoint the sources of error more precisely.

CONCLUSIONS

The major conclusion to be drawn from the research presented herein is that the heat output of a wood stove can be predicted by measuring its surface temperature. It has been shown that, given the surface area of the stove, the simplified heat transfer model developed here can be used to predict the heat output to within 25% of the values measured by a calorimeter room. Two strategies for measuring surface temperatures, measurements of individual surface temperatures and measurement of an average surface temperature, were shown to be capable of providing accurate predictions of the total energy delivered to the room, as well as track the instantaneous heat output of the stove. The analyses showed that even the simplest of the measurement strategies, monitoring the temperature of a single surface, could possibly provide accurate predictions of the heat output of the stove. Because this simple measurement strategy requires only one channel of a data acquisition system, it makes widespread monitoring both feasible and relatively inexpensive.

Based on the analyses of the errors in the predictions, it can also be concluded that the location of the sensors on the stove can significantly change the predictions. It was shown that measuring the temperature near the corner of a surface can reduce the heat output prediction by 50% relative to measuring at the center of the surface.

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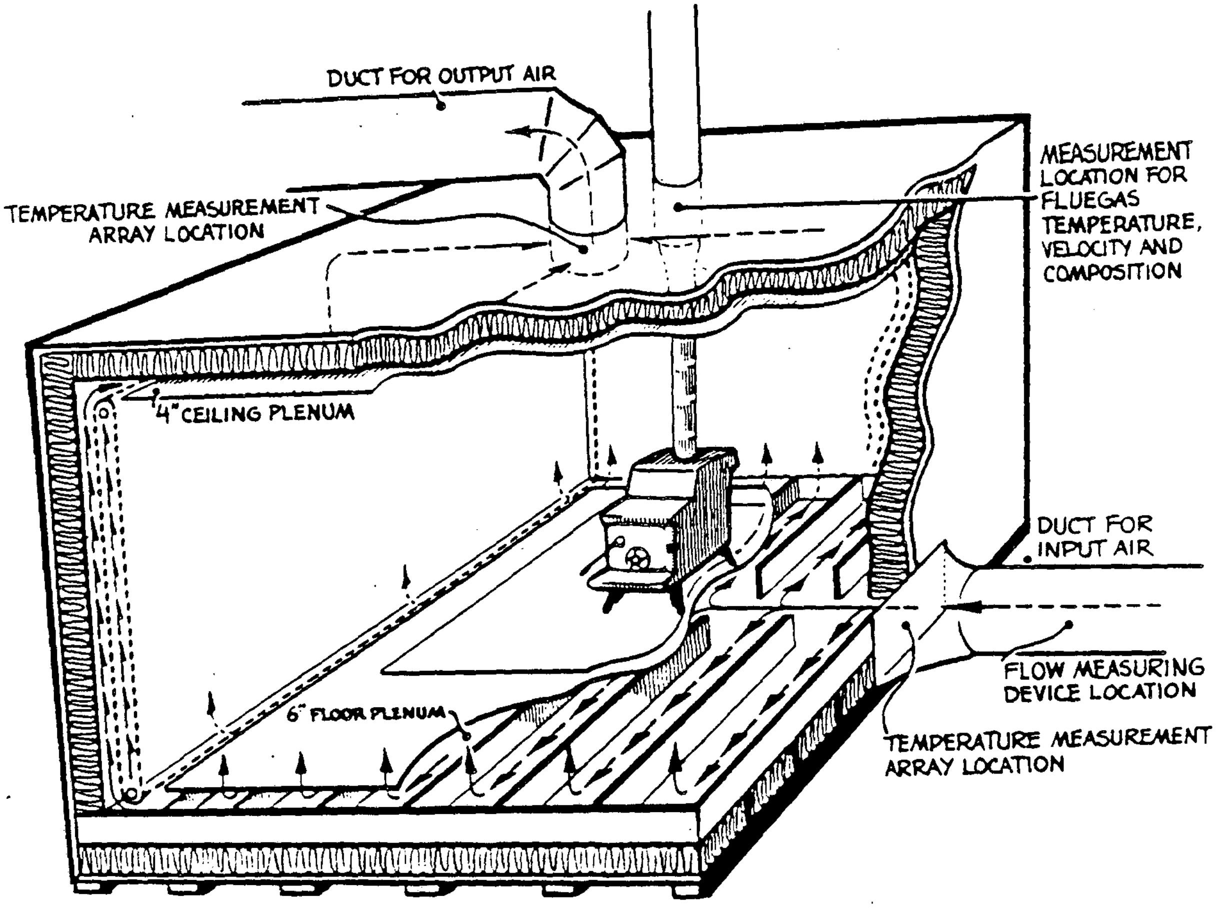
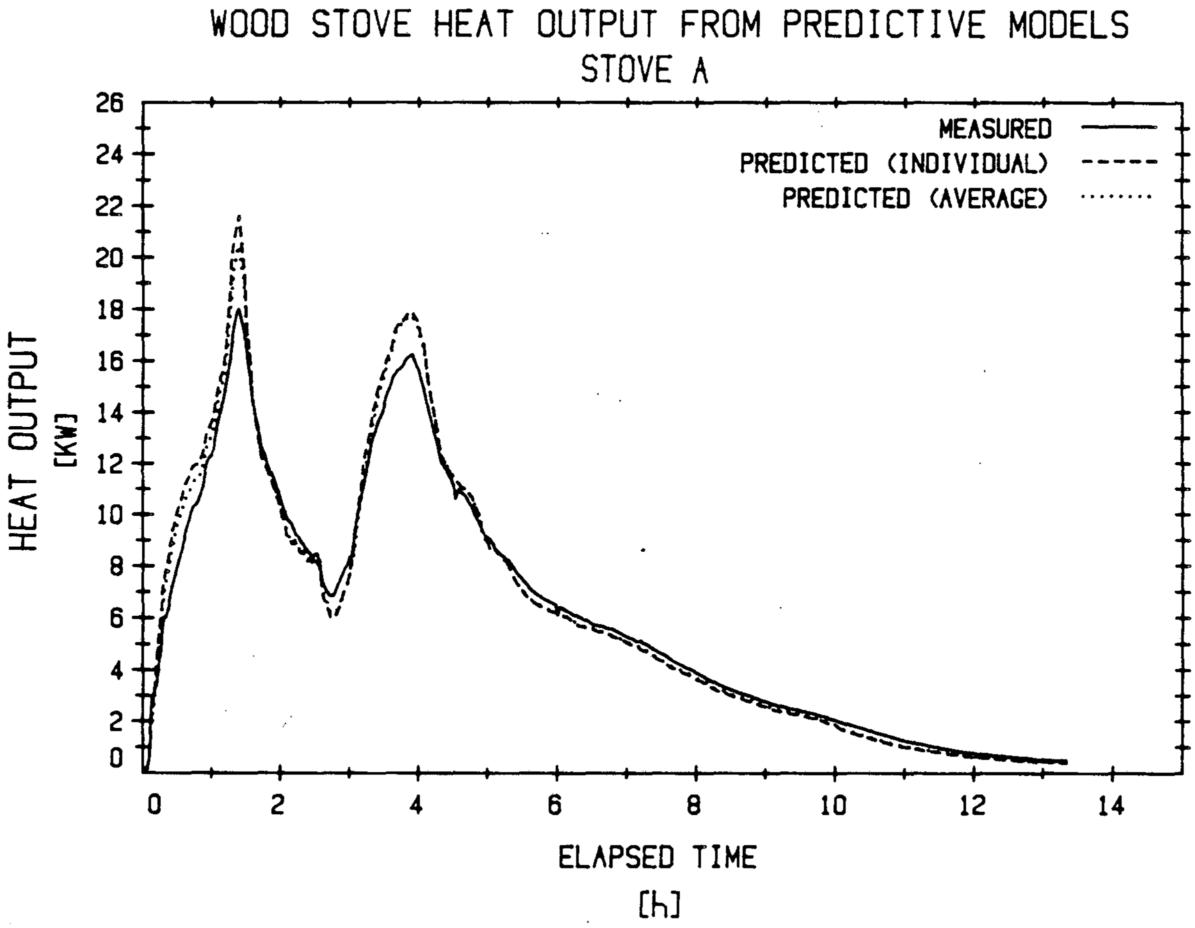


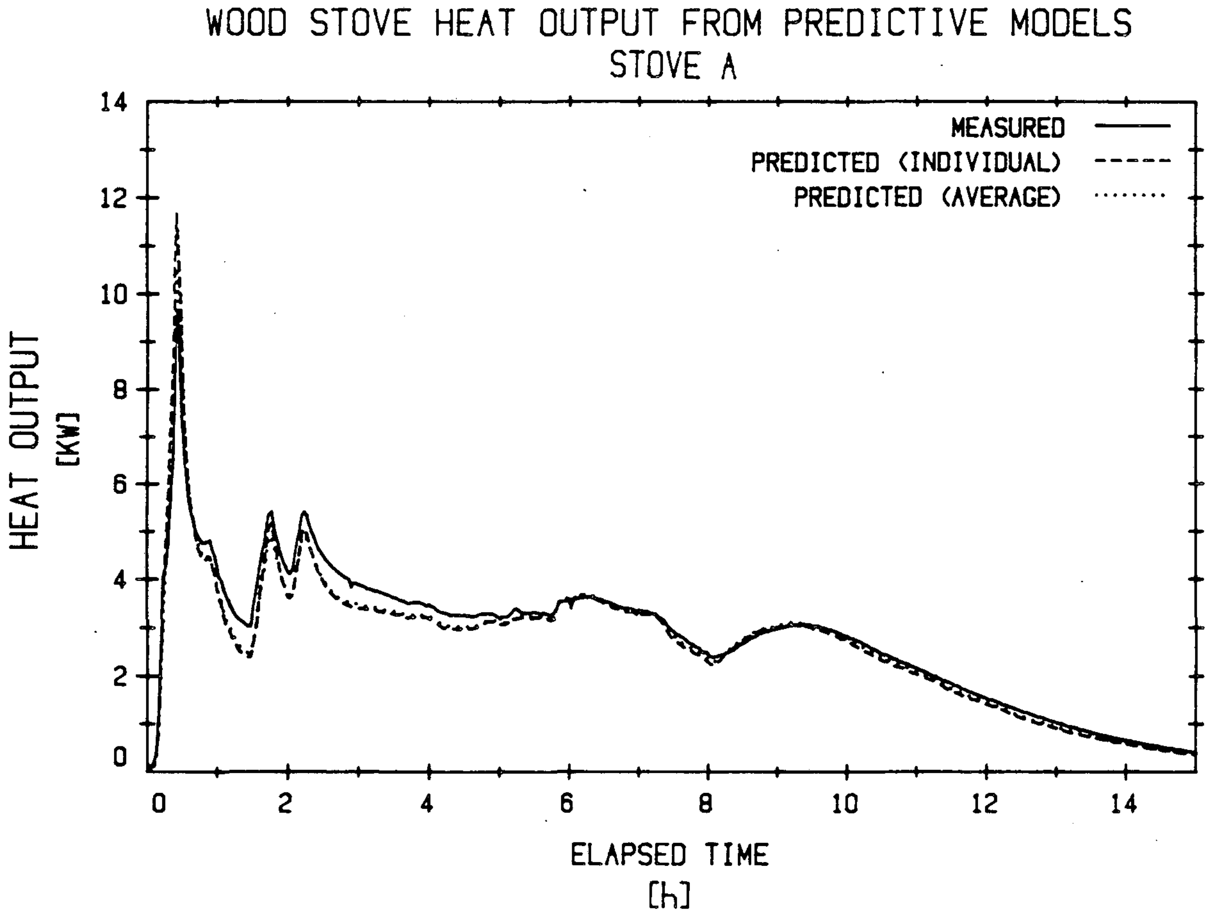
XBL 844-1592

Figure 1. Calorimeter room used to measure heat output of wood stoves.



XBL 847-3057

Figure 2. Measured and predicted heat output of Stove A (Test #1); predictions from individual temperature measurements and average temperature measurements.



XBL 847-3058

Figure 3. Measured and predicted heat output of Stove A (Test #2); predictions from individual temperature measurements and average temperature measurements.

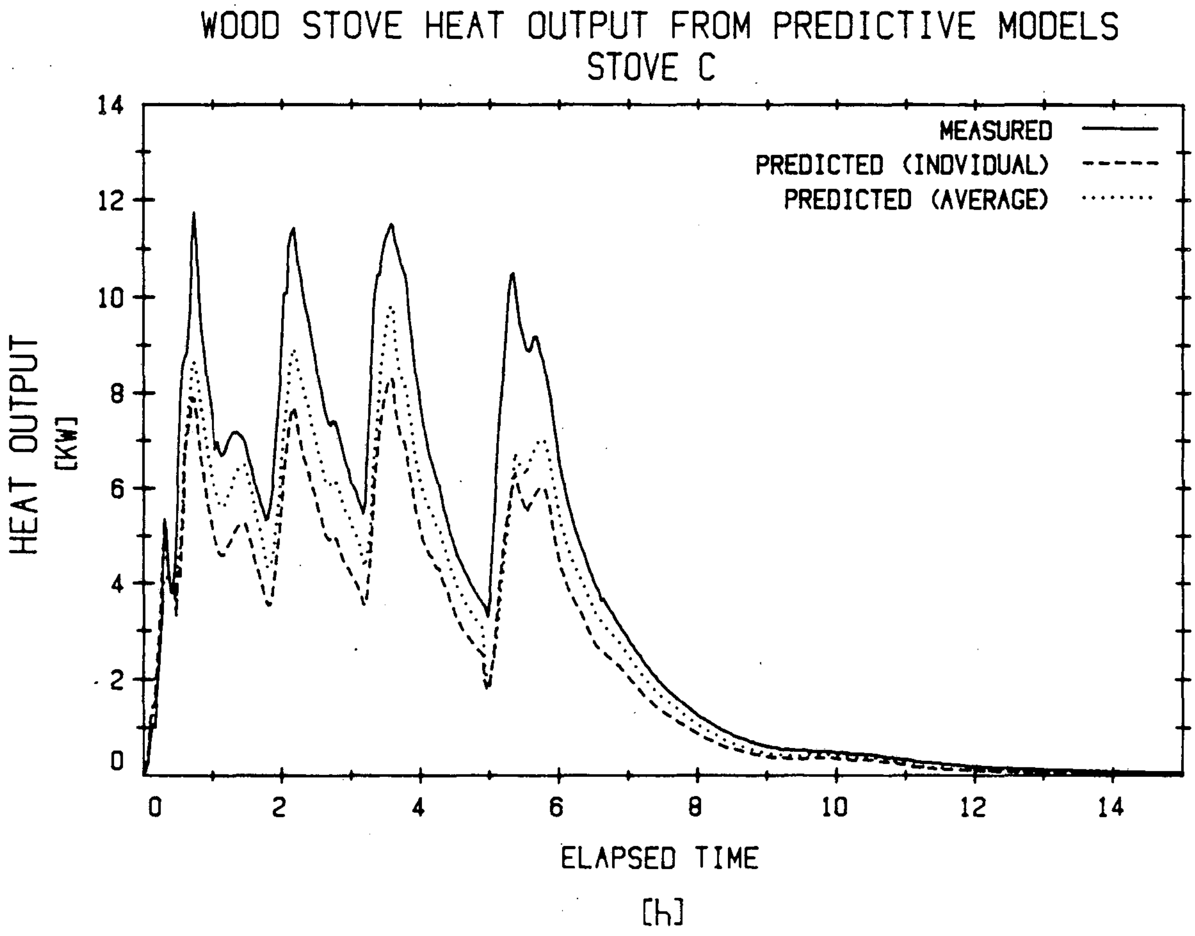


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Figure 4. Measured and predicted heat output of Stove C (Test #1); predictions from individual temperature measurements and average temperature measurements.

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