Pyrodynamics. 1966. Vol. 4, pp. 191–203. © 1966, Gordon & Breach, Science Publishers Ltd.
Printed in Northern Ireland

A CALORIMETER FOR DETERMINING RADIATION AND CONVECTION IN SMALL-SCALE COMBUSTIONS†

R. J. McCarter and A. Broido
Pacific Southwest Forest and Range Experiment Station
Forest Service, U.S. Department of Agriculture, Berkeley, California 94701

Abstract—A "bench-top" calorimeter was constructed to determine the radiation–convection partition of energy released in the burning of fuel. Samples of 1 to 2 g, no larger than 4 cm in horizontal dimension, and with flame height less than 15 cm, can be accommodated. The apparatus functions by (1) absorbing radiant energy from the burning fuel in an insulated thin aluminum shell whose temperature rise is recorded by a series of imbedded thermocouples, and (2) measuring convective energy by recording the electric power required to heat an air stream so that it just balances the flow rate and temperature of gases convected from the combustion zone. The apparatus can be used with both liquid and solid fuels and for both glowing and flaming modes of combustion.

INTRODUCTION

The spread of fire in the burning of large fuel complexes, such as forests or cities, results from the transfer of the combustion energy produced by the burning fuel to the nearby unignited materials. Most attempts at modeling such fire propagation start with assumptions, largely intuitive, about the spatial and temporal distribution of energy release and the principal mechanisms of energy transfer. The National Academy of Sciences–National Research Council Committee on Fire Research proposals for a concerted research effort to improve our fundamental understanding of fire[1] have spurred laboratory studies using wood cribs or pine needle beds to investigate the major parameters affecting the combustion process.[2,3] Although more economical and manageable than full-scale field experiments, such fires are still costly and time-consuming. Thus, experiments on an even smaller scale are useful if many variables or many materials are to be tested.

This situation arose after Broido and Martin[4] demonstrated that small traces of inorganic impurities could drastically alter the combustion behavior of cellulose and change a primarily flaming process to a primarily glowing process. The consequent increase in the number of

† Paper 66-14, 1966 Spring Meeting, Western States Section, The Combustion Institute, Denver Research Institute, April 1966.

required tests indicated the need for an apparatus permitting combustion measurements on a "bench-top" scale, *i.e.*, involving the free burning of one- or two-gram samples in air. With the sample supported so that negligible heat would be lost by conduction, the apparatus was to measure the radiative and convective energies released by the fuel without confusing or appreciably interchanging these energies.

DESIGN

The principles of the evolved apparatus were (1) the absorption of emitted radiation by a surrounding chamber, the measurement of this energy by recording the temperature rise of the chamber mass, and (2) the determination of convected energy by measuring the electric

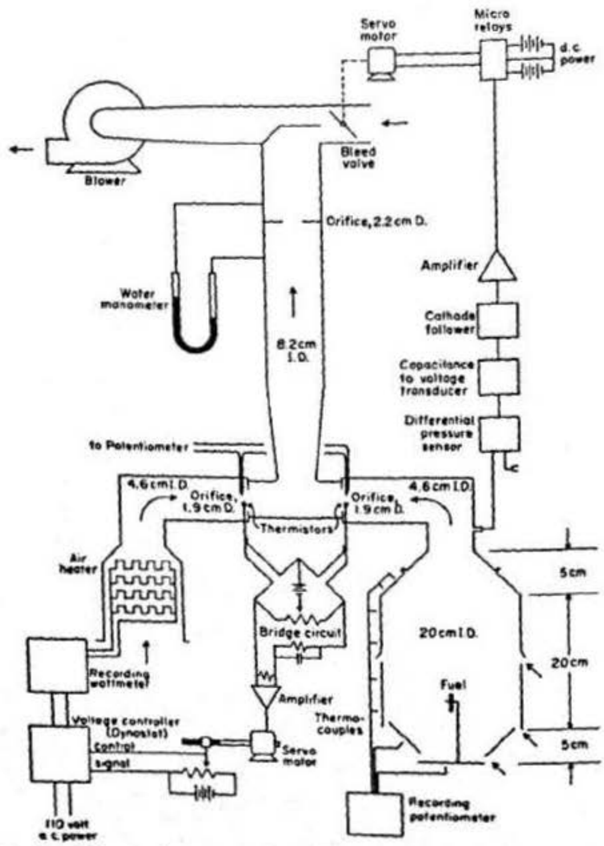


Figure 1.—Schematic Diagram of Calorimeter Controls and Air Flow.

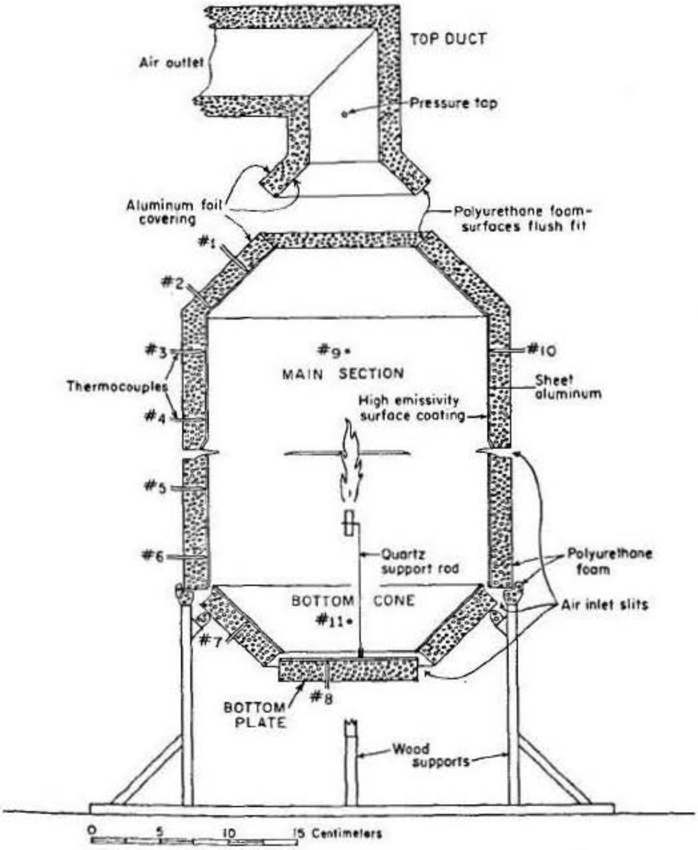


Figure 2.—Cross Section of Combustion Chamber.

power required to heat an air stream duplicating the flow rate and temperature of gases convected from the combustion zone. The equipment, shown schematically in Figure 1, included (1) a combustion chamber with thermocouples imbedded in the shell, (2) a second chamber housing the heater which duplicated electrically the energy output of the convection column, (3) a blower with control system to draft appropriate air flows through the two chambers, and (4) a temperature-sensing control system for matching heater power to the convected energy.

Combustion Chamber. Since a small fuel sample burning with flames presents a vertically elongated radiating source, the combustion chamber (Figure 2) was shaped as a cylinder with truncated cone ends, with height greater than diameter in order to obtain a more uniform absorption of radiation. Shadow-graphs of the convection column from a series of preliminary tests were used to proportion the top opening of the chamber.

As a compromise between inaccuracies from temperature measurements and from incurred heat losses, the shell was made of sheet aluminum of 0.8 mm thickness to provide a heat capacity yielding roughly a 20°C temperature rise in a typical combustion. Slots, 3 mm in width and arranged for negligible escape of radiation and sufficient air inflow to simulate natural convection, were provided half-way around the middle of the chamber wall and entirely around both ends of the bottom conical section. The latter slots thus divided the chamber into three sections, the main section comprising 80% of the shell weight, the bottom cone 16%, and the bottom plate 4%. Eleven chromel-alumel thermocouples, imbedded in the shell by drilling and peening, served to record vertical and radial temperature profiles. Two recording potentiometers were normally connected to thermocouples #4 and #7, centrally located in the major shell sections, but a manual switching arrangement permitted intermittent checks of the output of the other thermocouples.

To maximize radiant absorption, the internal surfaces of the shell were coated with a 0.01 mm thickness of stove blackening, providing a spectral reflectivity less than 13% out to 13 μ.† To minimize heat losses from the shell, an insulating layer of 1.6-cm-thick polyurethane foam (thermal conductivity 4.8×10^{-5} cal/sec-cm-°C), covered with 0.03 mm aluminum foil, was placed over the walls of the chamber. Further thermal insulation was inserted between the chamber and the wood support structure.

Fuel samples are supported by a slender quartz rod extending 10 cm upward from the bottom plate. The top opening of the chamber, with diameter somewhat larger than a typical convection plume at this height, connects with a flush fit to the duct through which the convection gases are withdrawn. This duct, fabricated with polyurethane foam walls, was lined with aluminum foil to minimize thermal absorption, with a 90° turn to provide a surface which would reflect radiation back into the

† More recently available materials which provide a threefold reduction in infrared reflectivity, but require a somewhat thicker coating, include Thos. Parsons & Sons Optical Black Enamel (available in the United States from The Eppley Laboratory, Inc.) and 3-M Black Velvet (a more durable optical coating particularly useful for field instrumentation).

chamber. Since the duct opening is less than 3 percent of the projected spherical area of the chamber, radiation losses are negligible.

Heater. For minimum heat capacity and thermal lag, the heater chamber was also fabricated from polyurethane foam covered with aluminum foil. Air enters the open bottom of the housing and exits through ducting duplicating that of the combustion chamber. The heater was formed from four thin nichrome wire coils, connected in parallel, and wound into a helix with asbestos paper separation between coils and between the heater and the housing. Measurements under conditions of use showed radiant heat losses to be less than 0.1% of heater output.

Airflow Control System. A standard household rotary vacuum cleaner is used as the exhaust mover, with draft to the system regulated by a bleed valve. To insure inward air flow to the combustion chamber and outward flow of all heated air through the duct, natural convective conditions are simulated by adjusting the bleed valve to maintain a slight sub-atmospheric pressure (0.04 mm Hg) at the top of the chamber. The control system consists of a Decker 306 differential pressure sensor with pressure tap through the top duct wall, a Decker Delta Unit 902-1 capacitance-to-voltage analog transducer, and a Decker Monitor 902-2 cathode follower with its output amplified to actuate polarized relays that control power to a servo which positions the bleed valve. A square-edge orifice (2.22 cm D) in the 8 cm D main duct, with radius taps and a differential water manometer, serves to monitor the gas flow. The pressure difference at this orifice is normally about 0.7 mm Hg, which, over the gas temperature range of use, corresponds to an air flow of about 170 to 200 liters/min. A tee-junction joins the main duct to the smaller ducts from the combustion chamber and from the heater housing. Duplicate 1.9 cm D orifices were installed on each side of this tee. Since these orifices provide the major resistance to flow in each leg, and since the lesser flow resistances are also balanced, equal volumetric flow rates are obtained in the two legs. (This was checked by inserting a blank flange successively in each side of the tee connection and measuring the air flow as a function of bleed setting.) Equality of air flow can also be checked using the thermistors in the orifice throats (see Heater Control System) as velocity sensors.

Heater Control System. For air temperature sensing, a thermistor was installed in each orifice throat at the tee-junction. A matched pair of thermistors (Nucleonics Products Co., type CS bare, $\sim$2 KΩ at

25°C with α of -3.6% per °C and time constant 6 to 8 seconds in still air) were selected from stock on the basis of similarity of response in a stream of heated air. After several cyclings through the intended temperature range to remove subsequent drift, the selected pair gave a resistance ratio that varied only about 2% over the temperature range of 20 to 95°C. This variation corresponds to an error of less than 1% in the indicated temperature balance.

The thermistors were connected into a bridge circuit actuating the control sequence indicated in Figure 1. The bridge circuit output, with capacitor and resistances introduced in the circuit to provide derivative control action, is fed through an AC amplifier to supply rotor current to a servo motor that adjusts the voltage signal to a solid-state power regulator (Dynastat Model 8120, time constant 5 milliseconds). Power input to the heater, as recorded by an Esterline-Angus recording wattmeter, Model AW, then provides the electrical equivalent to the convection energy emerging from the combustion chamber.

Any heat accumulation, lag, or loss in the outlet duct of the combustion chamber is balanced out by its equivalent counterpart in the heater duct. Differences in gas composition resulting from products of combustion do not significantly affect the convection measurement. For example, the carbon dioxide and water released by the complete combustion of 1 g of cellulose in one minute is estimated to change the specific heat of air in the outlet duct by less than 0.4%.

As a means of monitoring the accuracy of heater control, a pair of opposed chromel–alumel thermocouples were installed in the throats of the balancing orifices, and their output differences were routinely recorded.

CALIBRATION

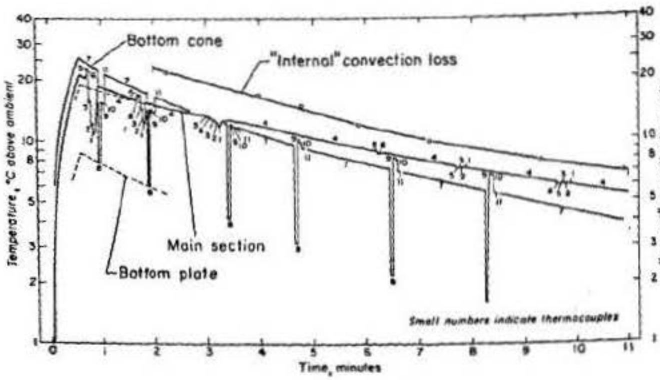
Convected Energy. The proper functioning of the system for convected energy measurement was checked by a counterpart electrical input. A nichrome wire heater element, surrounded by concentric radiative shields, was placed in the center of the combustion chamber. Power input to this heater was measured by a wattmeter and compared to that indicated by the recording wattmeter. Agreement within the readout ability of the wattmeter scales was obtained under steady-state conditions over the range of use; *i.e.*, in the order of 2% for readings of 100 watts. By alternately moving a flame source into and out of the chamber, the time constant for response of the convection measuring system was determined to be 8 seconds.

Radiant Energy. The major problem in calibrating the apparatus for radiant energy measurements was that of determining the heat capacity of the shell assembly. The insulation, attachments, and coating of the shell added considerable heat capacity to the value of 97 cal/°C which may be computed for the 451 grams of aluminum in the walls. Thus, it was decided to determine the effective heat capacity experimentally under conditions approximating those for which the apparatus was designed. The calibrations were obtained using a tubular quartz infrared lamp (GE no. 500T3, 115–125 V) suspended by its electrical leads within the calorimeter chamber.

With this lamp set at full power, the energy required to heat the shell 20°C could be delivered in a few seconds. Such a procedure minimizes errors resulting from losses from the shell proper, but it maximizes uncertainties in estimating mean shell temperature—since temperature non-uniformities are at a maximum. Longer exposures involve increasingly inaccurate corrections for heat losses from the shell. In fact, if the entire shell assembly is considered to be the receiver, the measured heat capacity increases with exposure time, as more of the insulation is involved. Under steady-state conditions, attained in times of the order of an hour, the effective heat capacity is maximized. Under such conditions the non-uniformities in shell temperatures are minimized but, because of the long times, errors due to heat losses from the assembly are maximized.

Depending upon the duration of the calibration procedure used, the effective heat capacity of the shell assembly was found to range from 145 to nearly 200 cal/°C. The lowest values were obtained using two similar procedures. In the first of these, air flow was set at its normal rate, power of about 400 watts (as monitored by a wattmeter) was applied for a short interval, and the temperature rise of the shell was recorded by its thermocouple array. The heat capacity of the shell was calculated after correcting for the heat retained by the lamp, the "internal" convection loss from the shell during the exposure, and the "external" losses through the shell assembly.

The first correction was estimated from the rated efficiency of the lamp and checked by measuring the mass of the lamp parts and the temperature increase under comparable conditions. The second correction was obtained by readings on the recording wattmeter during the experiment. The small correction for "external" heat loss came from the steady-state heat transfer coefficient, 3 cal/min-°C, obtained by taking the difference between the power input to the lamp and that to the balancing air heater ("internal" convective loss) at a succession of steady-state temperature levels in the shell. The temperature recordings of a typical test are presented in Figure 3, as are the computations leading to the estimated heat capacity.



Average shell temperature increase:

Main section = 18.7 × 0.80 = 15.0°C
Bottom cone = 25.6 × 0.16 = 4.1
Bottom plate = 8.9 × 0.04 = 0.4
Ave. = 19.5°C

Gross input to lamp (388 watts for 36 sec) =		3,340 cal
Retained by lamp (260°C × 1.5 cal/°C) =	390	
Convection loss (12 watts for 36 sec) =	102	
Conduction loss (ave. temp. of 9.3°C) =	28	
Total losses		520
Net input to shell		2,820 cal

$$\text{Est. heat capacity} = \frac{2{,}820\ \text{cal}}{19.5^\circ\text{C}} = 145\ \text{cal}/^\circ\text{C}$$

Figure 3.—Calorimeter Recordings and Heat Capacity from Infrared Lamp Input.

A similar estimate of the shell heat capacity was obtained by experiments with all openings, including the top, sealed with insulation and masking tape. As before, power was applied to the infrared lamp for a short interval and the temperature rise of the shell recorded. For such experiments, the heat retained by the lamp was estimated as before. There are, of course, no "internal" convection losses. With the shell sealed, the value for steady-state "external" heat losses was about double that obtained under normal conditions. Apparently in normal operation a sizeable fraction of the heat conducted through the insulated walls subsequently re-enters the shell through the open slots.

Later portions of curves such as those recorded in Figure 3 provide higher values for the effective heat capacity. For example, Table 1 presents the computation made by integrating the "internal" convection curve for the arbitrarily-selected 9-minute interval beginning at 2 minutes. This integrated value, corrected for external losses and heat input from the cooling lamp, may be divided by the decrease in the average shell temperature to provide a heat capacity estimate.

TABLE 1.—HEAT CAPACITY OF SHELL FROM TEMPERATURE DECAY.

1. Time interval: from 2 min to 11 min (see Figure 3)	
2. Average shell temperature decrease:	
(a) Main section = 10.0°C × 0.80 = 8.0°C	
(b) Bottom cone = 12.7°C × 0.16 = 2.0	
(c) Bottom plate = 4.8°C × 0.04 = 0.2	
Average = 10.2°C	
3. Heat loss from shell:	
(a) "Internal" convection loss (112 watt-min)	= 1600 cal
(b) Estimated "external" loss (ave. temperature above ambient = 9.3°C) =	250 cal
(c) Estimated convection from lamp (105°C × 1.5 cal/°C)	= 160 cal
Net loss ($a + b - c$)	= 1690 cal
4. Estimated heat capacity = 1690 cal/10.2°C	= 166 cal/°C

The highest values of effective heat capacity, of course, came from the steady-state experiments. With normal air flow, temperature growth and decay curves gave a computed time constant of about 9 minutes for the shell assembly. The external loss coefficient, as given above, was 3 cal/min-°C. A large number of "internal" convection measurements led to an average value of 19 cal/min-°C for the coefficient of convective transfer to the inducted air flow. The product of the time constant and the sum of the two loss coefficients is the estimate of the heat capacity. Similar results are obtained for the closed shell case with only "external" losses and the correspondingly longer time constant.

For typical burn times of 1–5 minutes, the effective heat capacity for the assembly was estimated to be 160 ± 10 cal/°C. Since an uncertainty in shell heat capacity affects only the measurement of radiated energy, it has a correspondingly lower effect on the estimate of total energy release. Thus, for an experiment in which $\frac{1}{5}$ of the energy is radiated, a 10 cal/°C error in heat capacity corresponds to only a 2% error in total energy release.

Although the conventional time constant for the shell assembly is much longer than that for the convection measuring system, under the appropriate circumstances the radiation measurements can more rapidly indicate changes in burning rate. For uniform irradiation of the front (inside) surface of the shell, the differentiated time constant[5]

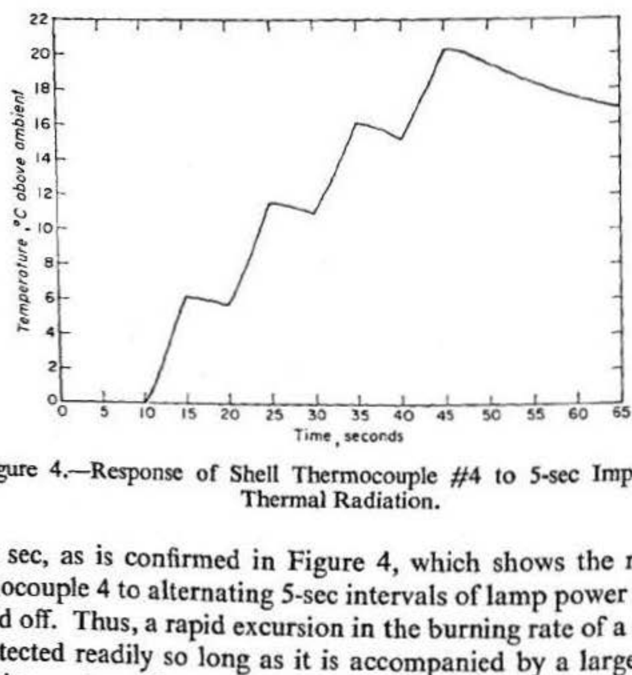


Figure 4.—Response of Shell Thermocouple #4 to 5-sec Impulses of Thermal Radiation.

is <1 sec, as is confirmed in Figure 4, which shows the response of thermocouple 4 to alternating 5-sec intervals of lamp power (500 watts) on and off. Thus, a rapid excursion in the burning rate of a sample can be detected readily so long as it is accompanied by a large change in radiation rate. However, since non-uniform exposure introduces significant lateral heat flow, measurement at one thermocouple location cannot distinguish between a temporal and a spatial excursion.

One obvious way in which an extreme non-uniformity in shell temperature can be introduced would be for the convection plume to impinge on the top edges of the calorimeter shell. To determine the vertical heat transfer rate in such circumstances, the infrared lamp was used to heat the upper part of the shell quickly and evenly while the rest of the shell was shielded by asbestos paper. The decay of the vertical temperature gradient was then recorded by the thermocouple array and a factor, 10.5 cal/min-°C difference between thermocouples no. 1 and no. 4, was derived for vertical heat transfer *versus* temperature gradient assuming uniform distribution of shell heat capacity.

OPERATION

All calorimeter runs have been made with the apparatus initially at ambient temperature, the flow controller adjusted to 1 cm of H_2O differential pressure at the main orifice, and the temperature controller zeroed and operating. The bottom plate of the combustion chamber is momentarily lowered to permit ignition of the fuel sample. Since the burning rate of the sample must remain within limits prescribed by the calorimeter, preliminary trials are used to determine appropriate sample size and shape, and direction of flame envelopment, *i.e.*, upward, lateral, or downward. Cellulosic fuels over a range of density, with and without retardant treatment, have been satisfactorily burned. Liquid fuels (hydrocarbons and alcohols) have also been burned in fuel pans, 0.2 to 1 cm^3 in volume, fabricated from 1-mil aluminum foil with epoxy resin joints and supported by thin quartz or wood rods.

Results, in the form of percent energy radiated and convected, are calculated by converting the heat absorbed by the shell and the electric energy to the heater to common units, for parts or for all of the burning period. Depending upon the burning characteristics of the fuel, various minor corrections may be introduced for extraneous thermal transfers and heat losses as warranted. For most experiments to date, the magnitude of these corrections has been less than 2% of the total energy.

In cases where combustion clearly proceeds to completion, the post-combustion temperature curve for the shell can be extrapolated back to the appropriate mid-point of the run, as in standard calorimetry practice, to obtain the estimation of radiated heat. As a check on this procedure, especially whenever only part of the burning period is of interest, corrections for heat losses may be made by integrating the shell temperature record over the appropriate time interval, as was done in the calibration experiments.

As noted earlier, heat transfer from gas to shell occurs when the convection plume spreads sufficiently to impinge on the top edges of the calorimeter shell. In some cases, flame height did not permit raising the sample support to eliminate this extraneous transfer. Such impingement is, of course, noted by the vertical temperature profile afforded by

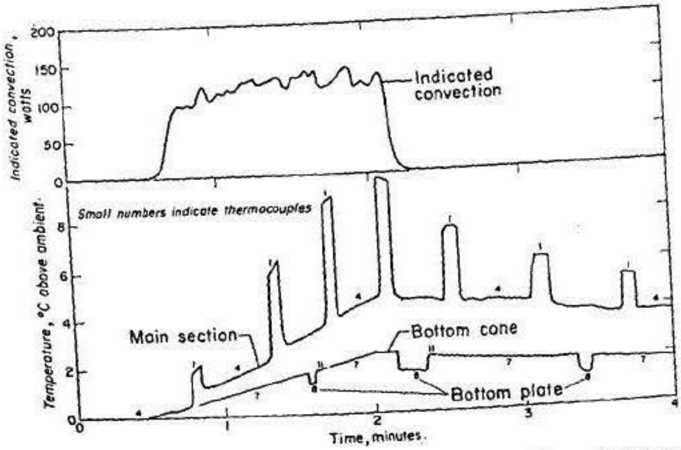


Figure 5.—Calorimeter Recordings for the Combustion of 0.6 cm³ of *n*-Propanol.

TABLE 2.—CALCULATION OF RADIANT AND CONVECTIVE ENERGY FROM THE BURNING OF A SMALL SAMPLE OF *n*-PROPANOL.

1. Fuel burned, 0.6 cm³ of *n*-propanol, in 1.3-cm-diam. pan with about 8 cm flame height
2. Duration of burn = 1.47 min
3. Average shell temperature increase, Δt (from Figure 5):
   (a) Main Section = 4.93°C × 0.80 = 3.94°C
   (b) Bottom Cone = 2.46 × 0.16 = 0.39
   (c) Bottom Plate = 1.97 × 0.04 = 0.08
   Average = 4.4°C
4. Recorded radiant energy, $R = 4.4°\text{C} \times 160\ \text{cal/°C} = 700$ cal
5. Recorded convective energy, C (from Figure 5) = 176 watt-min $\times\ 14.34 \frac{\text{cal}}{\text{watt-min}} = 2{,}530$ c
6. Corrections:
   (a) Conduction heat loss from shell (3 cal/min-°C) = 11 cal
   (b) Convective heat loss from shell (19 cal/min-°C) = 65 cal
   (c) Shell heating by plume impingement (from T.C. #'s 1 and 4)
      (1) In top of shell at end of exposure = 83 cal
      (2) Transferred down shell during exposure = 47 cal
7. Corrected estimate of radiant energy,
   $R' = R + a + b - c(2) =$ 730 cal
8. Corrected estimate of convective energy,
   $C' = C - b + c(1) + c(2) =$ 2,600 cal
9. Estimated total energy = 3,330 cal
10. Calculated heat of combustion of 0.484 g *n*-propanol c 2% H_2O = 3,444 cal

the thermocouple array, so that a correction can be made for the surplus heat residing in the top of the shell at run completion, and for conductive transfer down the shell during the course of a run.

The burning of a sample of *n*-propanol may be used to illustrate the heat balance computations for the case of complete combustion but with relatively large transfer corrections. The combustion was judged complete since there was no residue and since no trace of carbon monoxide could be found in the convection gases as monitored over the duration of burning by a detector unit sensitive to 0.01 vol. % of CO. Convection plume impingement on the shell is demonstrated by the high ratio of readings at thermocouple #1 to those at thermocouple #4, as may be seen in Figure 5, which gives the calorimeter recordings for the run. The energy calculations are given in Table 2.

Experience to date indicates that, in such cases of complete combustion, balances to within 5% of the total may be expected between heat of combustion and measured energies, with some small uncertainty attached to percent moisture and the combustion heat value. Calorimetry results are to be described in another report.

Acknowledgments—The authors wish to acknowledge the helpful contribution of J. R. Murray and A. W. McMasters, U.S. Forest Service, who assisted in the devising and assembly of instrumentation.

REFERENCES

1. National Academy of Sciences–National Research Council. *Fire Research Abstracts and Reviews*, **1**, 1–8 (1958). $\pi\rho$ 00009.
2. Fons, W. L., Clements, H. C., Pong, W. Y., and Richards, S. S. *U.S. Dept. Agr. Forest Serv. Pacific Southwest Forest Range Exp. Sta. Summ. Progr. Rept. I on Project Fire Model* (1960). $\pi\rho$ 01276.
3. Anderson, H. E. and Rothermel, R. C. *Tenth Symposium (International) on Combustion*, pp. 1009–1019. Combustion Institute, Pittsburgh, Pa., 1965. $\pi\rho$ 04460.
4. Broido, A. and Martin, S. B. *Fire Research Abstracts and Reviews*, **3**, 193–201 (1961). $\pi\rho$ 01278.
5. Broido, A. and Willoughby, A. B. *J. Opt. Soc. Amer.*, **48**, 344–350 (1958). $\pi\rho$ 01290.