Convective Heat Transfer Measurements in Cigarettes*

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NOMENCLATURE

- A_T = surface area of tobacco shred [cm^2].
 c = coefficient of specific heat [$\text{cal}/(\text{g } ^\circ\text{C})$].
 d_c = diameter of cigarette [cm].
 d_I = diameter of thick wall insulation [cm].
 d_p = diameter of particle [cm].
 d_w = thickness of thin wall insulation [cm].
 G = superficial mass velocity = $\rho u \Phi$ [$\text{g}/(\text{cm}^2 \text{ s})$].
 H = convective heat transfer coefficient between tobacco and gas [$\text{cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$].
 H_o = convective heat transfer coefficient between outside surface of insulation and ambient atmosphere [$\text{cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$].
 H_w = convective heat transfer coefficient between gas and inside surface of insulation [$\text{cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$].
 L = length of tobacco rod [cm].
 $\dot{m}$ = gas mass flow [$\text{g}/(\text{cm}^2 \text{ s})$].
 N = number of tobacco particles per unit volume [$\#/\text{cm}^3$].
 r = radial dimension [cm].
 t = time [s].
 T = temperature [$^\circ\text{C}$].
 u = gas velocity [cm/s].
 x = axial dimension [cm].

Greek Symbols

- α = thermal diffusivity [cm^2/s].
 λ = thermal conductivity [$\text{cal}/(\text{cm s } ^\circ\text{C})$].
 ρ = density [g/cm^3].
 Φ = porosity (volume of gas / volume of tobacco rod).

Subscripts

- No subscript implies the gas (e. g. T = gas temperature).
 T = tobacco.
 w = insulation.

1. INTRODUCTION

Convective heat transfer as used here refers to the transfer of heat by conduction between stationary tobacco particles in a cigarette and a flowing gas. If convective heat transfer is very rapid relative to other thermal processes in a burning cigarette (chemical reaction, radiative transfer, dilution) which change the temperature of either the tobacco or the gas, the local gas and tobacco temperatures are equilibrated. However, in a burning cigarette the rate of equilibration by heat transfer between gas and tobacco is of the same order as, for example, the rate of energy release in the coal by char oxidation. Consequently local temperature differences between gas and tobacco of more than 100°C exist in the smolder zone during a puff (1).

Convective heat transfer in a burning cigarette is of interest because it affects the temperature distribution and rates of heating of tobacco constituents which in turn control smoke generation. For example, the heat from the highly exothermic surface oxidation of char is convectively transferred to the gas. This heated gas flows from the zone of char oxidation toward the smoker and convectively transfers its heat to the cooler tobacco in the pyrolysis zone where most of the smoke is generated. High rates of convective heat transfer tend to concentrate heat in the tobacco rod near the zone of char oxidation where the major heat release occurs. Control of convective heat transfer constitutes a potential new tool for the cigarette designer, but the influence of convective heat transfer is difficult to access without the aid of a mathematical model. An unsteady, one-dimensional mathematical model of steady draw smoking which predicts overall burning characteristics of a cigarette has been developed (2). The model indicates that doubling the convective heat transfer coefficient can increase the rate of burning (the rate of movement of the paper char line) by 50%.

Convective heat transfer is proportional to the temperature difference between gas and solid. The proportionality factor, or convective heat transfer coefficient, is normally a function of the *Prandtl* and *Reynolds* numbers for forced convection. The problem of heat transfer in a cigarette has similarities to several well-studied problems in the chemical and mechanical engi-

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engineering literature, including heat transfer in packed beds and in thin-walled, large surface area heat exchangers. While this literature is instructive it does not provide directly useful numbers. A cigarette is a unique, low density packed bed having a particle size to bed diameter ratio that is too large to assure statistical uniformity.

It is worth noting that, because heat and mass transfer occur by analogous mechanisms in a packed bed, the heat transfer coefficient can be used to calculate a mass transfer coefficient. This latter number is important in modeling the rate of char oxidation in the coal of a cigarette (2, 11).

2. MEASUREMENT OF CONVECTIVE HEAT TRANSFER BY AN UNSTEADY-STATE TECHNIQUE

There are several techniques which have been used to measure convective heat transfer coefficients in packed beds. None of the techniques directly measure a heat transfer coefficient; they all require matching model solutions containing assumed coefficients with experimental temperature measurements in a packed bed. The technique selected here to measure the heat transfer coefficient in a cigarette is generally referred to as an unsteady-state technique. The experiments involved dried cigarettes at temperatures somewhat above room temperature ($\leq 50^\circ\text{C}$). Experiments at higher temperatures, although more appropriate, would be difficult to interpret because the tobacco would be gasifying. The unsteady-state technique begins with an insulated cigarette at room temperature. An abruptly started, steady flow of warm air enters one end of the cigarette. The resulting variation of air temperature versus time at the other end of the cigarette is an indirect measure of the heat transfer coefficient. If the heat transfer coefficient is zero the tobacco is not heated at all. As a result, the gas temperature at the end of the cigarette abruptly rises to the inlet temperature in the time it takes a fluid particle to travel the length of the cigarette. If the heat transfer coefficient is infinite the tobacco and gas temperatures are instantly equilibrated locally resulting in a tobacco shred being heated from room temperature to inlet gas temperature before the next downstream shred sees any elevated gas temperature. In this case, the gas temperature at the end of the cigarette abruptly rises to the inlet temperature in a time equal to the thermal mass of the cigarette divided by the rate at which heat enters the cigarette:

$$\tau = \frac{[\rho c \Phi + \rho_{TC} T (1 - \Phi)] L}{\rho c \Phi u}$$

It will be shown later that the actual gas temperature variation at the end of the cigarette lies between the above extremes.

3. DESCRIPTION OF THE EXPERIMENT

The model must be a real description of the experiment; therefore, the experiment is described first. The experimental apparatus shown in Fig. 1 was designed to record the gas temperature response at the end of a cigarette resulting from an abruptly started flow of warm air entering the cigarette. The major design difficulties are: [1] attaining a sharp step in inlet gas temperature at the start of the experiment, [2] connecting the cigarette into the apparatus, [3] sealing the sides of the cigarette to air flow, and [4] insulating the sides of the cigarette against heat losses.

The source of warm air is a 12 cm diameter by 15 cm deep packed bed of gravel with a typical particle dimension of 5 mm. Near the bottom of the gravel bed there are two heaters in series, each a matrix of resistance wire, powered by a variable transformer. To eliminate radial temperature gradients in the gravel bed, the sides are insulated with 2 cm of rigid, close-celled, low density, polyurethane foam. Provision is made to monitor, if desired, the pressure and temperature of the gas at the top of the gravel bed. A hollow cylinder of foam which fills most of the test section channels the warm air to the cigarette and holds the thermocouple which measures the inlet temperature at the bottom of the cigarette rod.

The cigarette is contained in another cylinder of rigid, closed-cell, polyurethane foam which also seals the cigarette paper against air flow through it, insulates the cigarette against peripheral heat losses, and holds the

Figure 1. Packed bed heater for unsteady-state heat transfer experiments. Insulated holder with cigarette and exit gas thermocouple is inserted at time zero.

sensitive thermocouple which measures exit gas temperature. The cigarette is mounted in the foam cylinder by lightly pressing it into a hole which is either cast when the foam is formed or drilled after the foam is formed. The end of the foam cylinder which attaches to the packed bed is machined to make a snug fit and then smeared with vacuum grease to insure an airtight connection with the top 3/8" of the phenolic tube which forms the test section. The cigarette in its holder is attached to the packed bed heater with no thermal mass between the cigarette and the warm air supply. Any such thermal mass absorbs heat in the gas flow at an unsteady rate and spreads out the desired step in inlet gas temperature, which is inconvenient to incorporate in the heat transfer model.

It is necessary for the cigarettes to be bone-dry. Even small amounts of water in a cigarette have significant effects on the exit gas temperature profile because of the large heat of vaporization of water (~ 540 cal/g). A cigarette dries in about two hours with the packed bed heater forcing warm, dry air through it. The cigarette and its foam holder are cooled prior to an experiment in an apparatus similar to the packed bed heater except the air temperature is maintained at room temperature.

The inlet and exit gas temperatures are measured by chromel/constantan thermocouples constructed with wire whose diameter (0.076 mm or 0.003") is small enough that the junctions are rapidly responsive to temperature changes during an experiment. Thermocouple signals are referenced by a room temperature junction, amplified, and fed into a multi-channel recording oscillograph. The combination of chromel and constantan for thermocouple junctions was selected because its voltage output per degree temperature change is the highest of those thermocouples listed and calibrated in the Omega Engineering Temperature Measurement Handbook (3). Thermocouple junctions are made by butt-welding the wires with an oxygen-acetylene torch in a helium atmosphere.

A sensitive measurement of heat transfer coefficient requires effective insulation against peripheral heat losses. Also, these heat losses imply radial temperature profiles in the cigarette which are not accounted for by the heat transfer models. Surrounding the cigarette with a vacuum would be the best steady-state insulation; however, the thermal mass of the necessary glass envelope would itself overwhelm unsteady heat transfer in the cigarette. In steady-state conductive heat transfer, best insulation occurs with the lowest thermal conductivity. However, in unsteady-state heat transfer, best insulation occurs with the lowest product of thermal conductivity times specific heat times density ($\rho c \lambda$).

An excellent unsteady-state (and steady-state) insulating material is rigid, closed-cell, polyurethane foam. In addition, it is impermeable to air flow, it can be made in the laboratory from a kit, and it is machinable. According to manufacturer's specifications the foam has a lower thermal conductivity [3.8×10^{-5} cal/(cm s °C)] than still dry air [6.2×10^{-5} cal/(cm s °C)], which

would make it a better steady-state insulator. However, the product ($\rho c \lambda$) for still dry air is less than that of foam making it a better unsteady-state insulator. With the foam insulation steady-state heat losses from cigarettes were about 25%.

A possible alternative way to attack the problem of heat losses from the sides of the cigarette is to use cigarettes with a diameter larger than 8 mm. Home-made cigarettes several centimeters in diameter or larger could be experimented with as long as the flow was one-dimensional along the axis of the specimen. However, a larger diameter cigarette would reduce the relative effect of channeling near the paper (significant in real cigarettes). Also, the packing would be more uniform which would reduce channeling in the interior of the cigarette. Channeling will be important in the explanation of the difference between measured heat transfer coefficients in a cigarette and calculated heat transfer coefficients based on the packed bed literature.

Another way to reduce heat losses into the peripheral insulation is to surround the test specimen with matched test specimens (matched with respect to weight and pressure drop). The reduced heat losses would be similar to using a test specimen of increased diameter. A multi-hole cigarette holder used here is a foam cylinder with a drilled hole in the center and six equally-spaced drilled holes on a 1.83 cm (0.720") diameter circle. Hole sizes are 0.804 cm (0.316") and the minimum wall thickness between any two cigarettes is 0.112 cm (0.044"). This is the minimum wall thickness which withstood the mechanical pressure of drilling and lightly pressing in the cigarettes. Steady-state heat losses from the central cigarette in the holder, which is the test specimen, are about 10% compared to 25% for the other holder.

4. MODELING THE UNSTEADY-STATE EXPERIMENT

Two ways were used to model the unsteady-state experiment. Both models describe the gas and tobacco in the same way; they differ in the completeness of their description of peripheral heat losses into the insulation. Because estimating the correct heat transfer coefficient requires repeated solutions of the model, a quickly solved although less accurate model is especially useful for initial estimates.

The models incorporate the following assumptions:

- Temperatures are less than 40 °C above room temperature, the tobacco is bone-dry and the air is dry. Therefore, there is no gasification from or condensation on the tobacco.
- Conduction of heat along the longitudinal axis of the cigarette is negligible in both the gas and tobacco. Experimental gas velocities are large enough to insure that the ratio of gas conduction to convection is much less than one. The tobacco shreds, which are oriented somewhat along the length of the cigarette, have a

small cross-sectional area compared to their surface area exposed to the gas flow. Thus a section of a shred is more readily heated by convective heat transfer than by conduction from an adjacent section.

c. The temperature across the thickness of tobacco shreds is uniform; that is, the characteristic time for a tobacco shred to be uniformly heated is much less than the time of passage of the gas temperature wave for the experimental conditions used.

d. The predominant gradients are in the longitudinal direction, therefore the models are one-dimensional (with respect to the cigarette). Peripheral heat loss is treated as a convective heat loss to the insulation that constitutes a sink in the equations.

e. The gas phase is quasi-steady. Time derivative terms are neglected in the gas phase equations because the gas phase adjusts so quickly to changes in the solid phase that the gas phase appears to be steady relative to the unsteady solid phase. For example, in the conservation of gas mass equation, the time rate of increase of mass within an elemental volume $\partial \rho$ is small compared to the mass flow sweeping $\frac{\partial}{\partial t}$ through the elemental volume. This equation thus reduces to $\dot{m} = \dot{q}u = \text{constant}$.

Based on these assumptions, both models contain an energy equation for the gas and an energy equation for the tobacco. These equations (for both models) are:

$$\frac{\partial T}{\partial x} = \frac{-HNA_T}{\dot{m}c\Phi} (T - T_T) - \frac{4H_w}{\dot{m}c\Phi d_c} (T - T_{w,o}),$$

$$\frac{\partial T_T}{\partial t} = \frac{HNA_T}{\rho_T c_T (1 - \Phi)} (T - T_T).$$

If temperatures are non-dimensionalized so that room temperature is zero and the inlet gas temperature is unity, the appropriate boundary/initial conditions are:

$$T(0, t) = 1, \text{ and}$$

$$T_T(x, 0) = 0.$$

What remains is to describe the temperature of the insulation next to the cigarette, $T_{w,o}$, as a function of space and time. The two approaches to describing $T_{w,o}$ are referred to as the thin wall and thick wall models.

The thin wall model assumes convective heat transfer from the gas to a thermally thin wall and convective heat loss from the outside of this thin wall to ambient. Actually, in the thick layer of insulation around a cigarette, heat is being transferred from the cigarette to the surrounding thin layer of insulation adjacent to the cigarette. This layer partly stores the heat which raises its temperature and partly transfers the heat to the next layer of insulation. Thus the thin wall model is qualitatively similar to the actual situation. In the thin wall model, the temperature is uniform across the

thickness of the wall, i.e. $T_w = T_{w,o}$. The additional energy equation describing the thin wall is:

$$\frac{\partial T_w}{\partial t} = \frac{H_w}{\rho_w c_w d_w} (T - T_w) - \frac{H_o}{\rho_w c_w d_w} T_w.$$

The boundary condition is that the wall is initially at ambient temperature: $T_w(x, 0) = 0$.

This wall energy equation requires three empirically fitted parameters: d_w , H_w and H_o . This fit comes from analyzing the exit gas temperature profile from a cigarette holder without the cigarette and by matching the steady-state heat loss when a cigarette is in the holder. Such an approximation to the actual heat losses is useful only if they are relatively small compared to the heat influx of the warm air.

For H_o much larger than H_w , the thin wall temperature remains close to ambient temperature; an ambient temperature wall provides no insulation. For $H_o = 0$, the thin wall is a small but finite heat sink which does not lose heat to ambient; in this case the wall is a good insulator. For H_o of the same order as H_w , which approximates the real case, the thin wall both stores heat and transfers it to ambient.

The thick wall model applies to a thick layer of insulation as in the actual experiment. Because axial temperature gradients within the insulation are small relative to radial gradients, the model considers only radial conduction of heat within the insulation. This greatly simplifies the numerical solution. In an unsteady-state experiment of 100 to 200 s duration, the insulation is too thick for the outside surface to be heated; therefore, this surface in the model is assumed to remain at ambient temperature. The energy equation describing the thick wall is:

$$\frac{\partial T_w}{\partial t} = \alpha_w \left\{ \frac{\partial^2 T_w}{\partial r^2} + \frac{1}{r} \frac{\partial T_w}{\partial r} \right\}.$$

The boundary condition between the gas and insulation is:

$$\lambda_w \frac{\partial T_{w,o}}{\partial r} = -H_w (T - T_{w,o}).$$

The boundary condition at the outside surface of the insulation is:

$$T_w(x = d_{1/2}, t) = 0.$$

The peripheral heat loss requires the specification of one fitted parameter, the heat transfer coefficient between the gas and the wall, H_w .

In Fig. 2 the exit gas temperature profiles employing the thin and thick wall insulation models are compared; insulation parameters that yield the same steady-state heat losses (about 20%) are used. The thick wall model profile is based on rigid polyurethane foam insulation: $\alpha_w = 3.22 \times 10^{-8} \text{ cm}^2/\text{s}$ and $\lambda_w = 3.79 \times 10^{-5} \text{ cal}/(\text{cm s } ^\circ\text{C})$. The heat transfer coefficient at the wall is $H_w = 0.0004 \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$. The thin wall model uses

Figure 2. Comparison of computed non-dimensional exit gas temperature histories for thin wall and thick wall insulation models.

the same value for H_w ; in order to obtain the correct steady-state heat loss, the parameter values $\rho_w c_w d_w = 0.001 \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$ and $H_0 = 9.5 \times 10^{-5} \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$ are used in the thin wall model.

The thin wall profile in Fig. 2 closely agrees with the thick wall profile for the first 25% of the exit gas temperature rise. One reason for the agreement was mentioned before; that is, a thin wall qualitatively corresponds to the first layer of insulation in a thick wall and the heat losses during the first part of an experiment are dominated by the layer of insulation next to the cigarette. Another reason for the agreement is seen in Fig. 3 which shows that the rate of heat transfer to the insulation is small compared to the rate of heat transfer to the tobacco for the first 25% or so of the exit gas temperature rise. Near the beginning of an experiment, only a fraction of the cigarette is heated, so only this section loses heat to the insulation. As the experiment progresses, the thermal wave moves further along the tobacco rod leaving more area behind it for heat transfer to the insulation, and so less heat reaches the front of the thermal wave.

So the first 25% of the exit gas temperature rise is predominantly a measure of heat transfer to the tobacco. Since this most sensitive region of the exit gas temperature rise is equally well described by the thin and thick wall models, the simpler thin wall model will be predominantly used.

5. EXPERIMENTAL RESULTS AND THEIR MODEL EVALUATION

Heat transfer in dry cigarettes was studied using the multi-hole and single-hole cigarette holders. The periph-

Figure 3. Timewise variation of heat flows during unsteady heat transfer measurement. Results computed from model with heat loss to a thermally thick wall.

eral heat losses are larger for the single-hole cigarette holder, but the flow rate through the center cigarette in the multi-hole cigarette holder is not directly controlled (it depends on an equalized flow distribution among seven cigarettes). The exit gas temperature profile is more sensitive to a small change in flow rate than it is to a small change in the heat transfer coefficient; therefore, a small imbalance in the flow distribution among the seven cigarettes can significantly reduce the accuracy of a heat transfer measurement. The convective heat transfer coefficients obtained from both cigarette holders are compared to indicate the sensitivity of the measured coefficients to the experimental technique.

The following parameters were varied: inlet gas temperature, cigarette length, flow rate, and type of cigarette (American or British). The inlet gas temperature ranged from 36 to 63 $^\circ\text{C}$ and did not affect the results. Two lengths of tobacco rod were used, 3.0 and 6.3 cm. The 6.3 cm tobacco rod is the total length of the tobacco rod in the brand of American filter-tipped cigarettes that was used. The 3.0 cm length was used to increase the rate of gas temperature rise at the end of the cigarette and to reduce peripheral heat losses. The exit gas temperature profile for longer cigarettes becomes increasingly diffuse due to heat transfer with the tobacco and diminishes due to peripheral heat losses. Thus, if the cigarette is too long, the exit gas temperature profile is insensitive to the heat transfer coefficient. On the other hand, there are certain disadvantages to the 3.0 cm length. First, non-uniformities along the length of the tobacco rod are averaged better with longer cigarettes. Second, any error in cutting the length of the test specimen and any tobacco shreds which fall from the

Figure 4. Matching of experimental and computed exit gas temperature histories for a 6.3 cm tobacco rod and a 15.1 cm³/s gas flow rate.

ends of the test specimen have a relatively greater effect for shorter tobacco rods.

The convective heat transfer coefficient is a function of the flow rate. Flow rates were varied from 7.6 to 28 cm³/s. With lower flow rates, peripheral heat losses have a larger effect on the exit gas temperature profile. For this reason, flow rates less than 7.6 cm³/s were not investigated.

Data were taken for the following test conditions:

[1] 3 cm American tobacco rods in the multi-hole cigarette holder at a flow rate of 15.0 cm³/s. The steady-state heat loss for these experiments was only 5%. However, because the data had more scatter than for other test conditions, it was not used in measuring a convective heat transfer coefficient. The scatter in the data was from cutting tobacco rods to 3 cm lengths and then trying to obtain matched sets (with respect to weight and pressure drop) of seven 3 cm rods.

[2] 6.3 cm American tobacco rods in the multi-hole cigarette holder at flow rates of 7.6 and 15.1 cm³/s. Experimental scatter was reduced by using 6.3 cm long tobacco rods in the multi-hole cigarette holder. Typical exit gas temperature profiles for three test specimens are shown in Fig. 4 (as dashed lines) for a flow rate of 15.1 cm³/s. For repeated experimental runs with the same cigarette there was usually less than a 2% variation in the exit gas temperature profile. Steady-state heat losses are less than 10% for the 15.1 cm³/s flow rate and less than 20% for the 7.6 cm³/s flow rate.

[3] 6.3 cm American tobacco rods in a single-hole cigarette holder at flow rates of 17.5 and 28.0 cm³/s. Heat losses for the single-hole cigarette holder are larger, steady-state heat losses with flow rates of 17.5 and 28.0 cm³/s are respectively about 30% and 25%. Variations between the profiles for repeated experiments

with the same test specimen are still generally less than 2% with respect to inlet gas temperature. All cigarettes are matched within $\pm 2\%$ for weight and pressure drop. The exit gas temperature profiles between ten different cigarettes varied more at the 28 cm³/s flow rate than at the lower flow rate.

[4] 3 cm British tobacco rods in a single-hole cigarette holder at a flow rate of 17.5 cm³/s. The thermal wave moves more slowly through the more densely packed British cigarettes; therefore, the shorter tobacco rods were used in order to keep steady-state heat losses at reasonable levels ($\sim 25\%$). Again, because of the shorter tobacco rods, the experimental reproducibility was not nearly as good. Note that cigarettes are not uniformly packed and non-uniformities in packing tend to be averaged out more when using full length tobacco rods.

The following parameters were used to describe the American cigarettes:

$$\begin{aligned} N &= 680 \text{ particles/cm}^3, \\ \rho &= 1.19 \times 10^{-3} \text{ g/cm}^3, \\ \rho_T &= 1.12 \text{ g/cm}^3, \\ c &= 0.25 \text{ cal/(g } ^\circ\text{C)}, \\ c_T &= 0.33 \text{ cal/(g } ^\circ\text{C)}, \\ A_T &= 0.066 \text{ cm}^2/\text{particle}, \\ d_c &= 0.8 \text{ cm}, \\ \Phi &= 0.8. \end{aligned}$$

To describe the British cigarettes three parameters were changed:

$$\begin{aligned} \Phi &= 0.71, \\ \rho_T &= 0.96 \text{ g/cm}^3, \\ N &= 815 \text{ particles/cm}^3. \end{aligned}$$

The porosity was measured by a gas displacement technique; the tobacco density, ρ_T , was then calculated to give the correct weight per length of tobacco rod. The number of tobacco shreds per unit volume, N , and the average size of the shreds was provided by Philip Morris, Inc., for typical American cigarettes. From the size data an average shred surface area, A_T , was computed. A precise knowledge of the parameters N and A_T is not required to measure heat transfer, since the parameter group $N \cdot A_T \cdot H$ [heat transfer per unit cigarette volume, cal/(cm³ s °C)] appears in both the gas and tobacco energy equations; therefore, $N \cdot A_T \cdot H$ can be measured and used just as easily as H alone. To compute explicit values of H for the British cigarettes the shreds were assumed to be the same size as for the American cigarettes (an approximation), and N for the British cigarettes was increased to reflect the greater weight of tobacco per unit volume of cigarette. Thus there is more uncertainty in the value of H for the British cigarettes than there is in the product $N \cdot A_T \cdot H$ for these cigarettes.

The data from the multi-hole cigarette holder were analyzed with the thin wall model, and the data from the single-hole cigarette holder were analyzed with both the thin and thick wall models. Note that the thick wall model is not applicable for the multi-hole cigarette

Figure 5. Measured convective heat transfer coefficients as a function of gas flow rate.

- a: American tobacco rods in multi-hole cigarette holder.
b: American tobacco rods in single-hole cigarette holder.
c: British tobacco rods in single-hole cigarette holder.

△ Indicates best fit value American cigarettes.

○ Indicates best fit value British cigarettes.

holder. The heat loss parameters used in the thin wall model for the multi-hole cigarette holder were:

$$\begin{aligned} Q_w C_w d_w &= 0.001 \text{ cal}/(\text{cm}^2 \text{ } ^\circ\text{C}), \\ H_w &= 0.0004 \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C}), \\ H_o &= 5 \times 10^{-5} \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C}). \end{aligned}$$

The heat loss parameters used in the thin wall model for the single-hole cigarette holder were:

$$\begin{aligned} Q_w C_w d_w &= 0.001 \text{ cal}/(\text{cm}^2 \text{ } ^\circ\text{C}), \\ H_w &= 0.0008 \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C}), \\ H_o &= 0.0003 \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C}). \end{aligned}$$

Note these heat transfer coefficients are larger than those for the multi-hole cigarette holder. The heat loss parameters used in the thick wall model for the single-hole cigarette holder are:

$$\begin{aligned} H_w &= 0.0004 \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C}), \\ \alpha_w &= 0.011 \text{ cm}^2/\text{s}, \\ \lambda_w &= 0.00014 \text{ cal}/(\text{cm s } ^\circ\text{C}). \end{aligned}$$

The measured heat transfer coefficients between tobacco and gas in the single-hole holder were the same using both the thin and thick wall models.

Shown in Fig. 4 are experimental exit gas temperature profiles on which are superimposed points computed with the thin wall model. The best value of H appears to be $0.0008 \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$, but $\pm 25\%$ changes in H [$H = 0.0006$ and $0.001 \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$] only slightly alter the model profiles. Notice that all profiles cross at about mid-temperature with the largest heat transfer coefficient yielding the profile with the greatest slope at the cross-over. Due to the dominance of heat losses (which are only approximately modeled) at the high

temperature end of the profile and the cross-over of profiles with different values of H at mid-temperature, the first part of the exit gas temperature profile is the most sensitive measure of H .

The measured heat transfer coefficients for American and British tobacco rods are shown in Fig. 5. Through this data on the log-log plot a line with a slope of unity has been drawn. This evaluation of the data shows the convective heat transfer coefficient to vary essentially linearly with the flow rate. The British tobacco rods at the one flow at which they were tested have nearly the same heat transfer coefficient as the American tobacco rods. So it appears that more tightly packed tobacco rods, which are characteristic of British compared to American cigarettes, do not measurably change the convective heat transfer coefficient. However, the product $NA_T H$ for the British cigarettes is about 50% higher than for the American cigarettes due to the higher packing density of the British cigarettes.

An indication of the absolute accuracy of the measurements of H is difficult to make. For the worst case, a range of heat transfer coefficients of $\pm 33\%$ about the best fit for H was within the limits of experimental results. However, the actual uncertainty is probably less.

6. COMPARISON WITH CALCULATIONS FROM THE LITERATURE

The conclusion that the convective heat transfer coefficient is linearly proportional to the flow rate does not appear consistent with other packed bed heat transfer measurements. Packed bed heat transfer data are correlated by *Whitaker* (4) with H (in terms of the *Nusselt* number) being proportional to the flow rate (in terms of the *Reynolds* number) raised to the $1/2$ power for laminar flow and raised to the $2/3$ power for turbulent flow. The flow in a cigarette for the heat transfer experiments ranges from purely laminar flow into possibly the transition region between laminar and turbulent flow.

Kunii and *Suzuki* (5) developed a theoretical model for heat transfer in packed beds with channeling in which the *Nusselt* number is proportional to the flow rate. Channeling is characteristic of low *Peclet* number flows [$Pe = (d_p c G / \lambda) < 10$]. In a cigarette, due to the slight orientation of tobacco shreds in the flow direction and the very sparse packing density channeling could persist for $Pe > 10$. Not only is the correlation of H versus flow rate different for channeling than for more evenly distributed flow in a packed bed, but the apparent heat transfer coefficient is also less. *Kunii* and *Suzuki* plotted the data of more than ten researchers to demonstrate that for $Pe = 1$, H is an order of magnitude less when channeling occurs.

At a flow rate of $17.5 \text{ cm}^3/\text{s}$ the data on Fig. 5 indicate that $H \approx 8 \times 10^{-4} \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$. Using the packed bed equation in [*Bird, Stewart and Lightfoot* (6)], $j_H = 0.91 \text{ Re}^{-0.51} \Psi$ ($\text{Re} < 50$), in which $j_H = H/Gc \text{ Pr}^{2/3}$ is

the Colburn j -factor for heat transfer and Ψ is the shape factor, one computes that $H = 3.8 \times 10^{-3} \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$ using the shape factor for flakes. When extrapolating the plot of j_H versus Re ($10 \leq Re \leq 1000$) in [Smith (7)] to $Re = 5$, $H = 2.5 \times 10^{-2} \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$ (this number is unusually high which is not a consequence of the extrapolation). Using the packed bed equation in [Gamson (8)], $j_H = 18.3 Re^{-1} (1 - \Phi)^{0.2}$ ($Re < 10$), $H = 5.7 \times 10^{-3} \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$. Thus, heat transfer coefficients in a tobacco rod appear to be about an order of magnitude less than those for other packed beds, which is further evidence of channeling as mentioned above.

Chang and Johnson (9) measured convective heat transfer coefficients for a flat section of tobacco leaf during constant rate drying experiments. For air flows of 132 to 178 cm/s at air temperatures of 94 to 160 $^\circ\text{C}$, they measured heat transfer coefficients in the range 3.8×10^{-4} to $5.9 \times 10^{-4} \text{ cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$. These gas velocities in a cigarette correspond to volume flow rates (at room temperature) of 53 and 71 cm³/s; for these flow rates Fig. 5 indicates values of H which are about five times higher than reported by Chang and Johnson. However, heat transfer in a bed of shredded tobacco is expected to be faster than for a flat leaf of tobacco because of the convoluted nature of the flow paths.

7. CONCLUSIONS

The convective heat transfer coefficients measured in dry American and British cigarettes near room temperature at flow rates from 7.6 to 28 cm³/s increase linearly from 0.0003 to 0.0012 $\text{cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$. The measured coefficients are evidence that the gas flow through the relatively porous (70–80%) tobacco rods follows channels rather than uniformly distributing itself over the cross section. Convective heat transfer coefficients reported in the chemical engineering literature for typical, large scale packed beds in which the flow is more evenly distributed are an order of magnitude higher and not linearly proportional to the flow rate.

Channeling of the gas flow affects more than convective heat transfer. For example, channeling will reduce the filtering capabilities of the tobacco rod to gases and particles in the smoke aerosol. Also, channeling will promote unstable burning during a puff by permitting air to penetrate the base of the coal at selected locations. These locations where more air enters have higher burning rates and temperatures which will affect the smoke composition.

SUMMARY

An unsteady-state technique has been used to measure convective heat transfer coefficients in the tobacco rods of American and British cigarettes. The coefficients measured in dry cigarettes near room temperature at flow rates from 7.6 to 28 cm³/s increase linearly from

0.0003 to 0.0012 $\text{cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$. Because of channeling of the gas flow in the relatively porous (70–80%) tobacco rods, these heat transfer coefficients are an order of magnitude lower than those for large scale packed beds, in which the gas flow is more evenly distributed.

ZUSAMMENFASSUNG

Die im Tabakstrang amerikanischer und englischer Cigaretten zu beobachtenden Wärmeübergangskoeffizienten wurden unter Verwendung einer nichtstationären Technik gemessen. Die in trockenen Cigaretten nahezu bei Raumtemperatur und bei Strömungsgeschwindigkeiten von 7,6 bis 28 cm³/s gemessenen Koeffizienten nehmen linear von 0,0003 auf 0,0012 $\text{cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$ zu. Durch die Kanalisierung des Gasstromes in dem relativ porösen (70–80%) Tabakstrang liegen die Wärmeübergangskoeffizienten um eine Größenordnung niedriger als jene von locker gepackten Medien, in denen der Gasstrom gleichmäßiger verteilt ist.

RÉSUMÉ

On a employé une technique de l'état non stationnaire afin de mesurer les coefficients de transfert de chaleur par convection dans les boudins de tabac de cigarettes américaines et anglaises. Les coefficients mesurés dans des cigarettes sèches environ à température ambiante et à des débits de 7,6 à 28 cm³/s augmentent de façon linéaire de 0,0003 à 0,0012 $\text{cal}/(\text{cm}^2 \text{ s } ^\circ\text{C})$. A cause de la canalisation du flux de gaz dans les boudins de tabac relativement poreux (70–80%), ces coefficients de transfert de chaleur sont d'un ordre de grandeur inférieurs à ceux de milieux moins tassés, dans lesquels le gaz se répartit de façon plus uniforme.

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