DUCT GAS COOLING MODEL

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Introduction

The work reported here was motivated by a fire safety plan proposed by H. A. Lee for glove boxes, hoods, or other enclosures which contain radioactive or other toxic materials. Toxic particulates in the ventilation gases drawn off from such enclosures to avoid outflow of toxic substances are typically removed from the flow stream by filtration. Accidental fires in glove boxes can produce not only gas temperatures damaging to filter materials but also burning debris capable of damaging filters by contact or ignition. The safety plan of H. A. Lee was to remove burning debris from hood or glove box gases by a primary filter near the enclosure and to remove toxic particulates at a location far enough downstream to escape damage.

The objective in the task reported here was to

- Calculate the temperatures produced in the duct as a function of inlet temperature, position, flow velocity, duct geometry, duct materials, gas composition, and other relevant parameters.
- Provide a method for convenient retrieval and utilization of the information for duct system design.

The first of these objectives has been largely met by the construction of a computer program (operational on the CDC CYBER 70 Model 74 computer) to calculate one dimensional gas flow with calculated heat losses, although a few refinements remain to be added. This program (GAS/COOL) has been used to simulate British Atomic Energy Establishment experiments¹ in electrically heated air flow through 8 inch diameter ducts. Agreement with this experiment was good. Agreement with a simplified hand calculation for air flow through 4-inch diameter ducts was fair. Calculated temperature drops in a 4-inch diameter stainless steel duct from a flame hood with an actual fire were somewhat smaller than those observed experimentally. Three possible contributors to this difference might be (1) transient effects in the experiment, (2) neglect of vibrational (but not rotational) energy in the computed thermal conductivity of duct gas mixtures, and (3) use of a higher ambient temperature (100°F) in calculations than was present in the experiment (64°F).

Computational Procedure and Theory

The calculational technique selected was to incorporate into BNW Program GAS², which describes one dimensional stream-tube flow of a mixture of compressible real gases, a calculated heat loss rate based on data correlations for boundary layer and convective heat flow.

The physics representation of steady state gas flow is summarized from the Program Gas document as follows:

<u>Physics Representation</u>	
Mass Balance: $\nabla \cdot \rho \underline{v} = Q_1 \cdot$	(1)
Energy Balance: $\nabla \cdot \rho \underline{v} (h + v^2/2) = Q_2 \cdot$	(2)
Momentum Balance: $\nabla \cdot (\rho \underline{v} \underline{v} + p) = \underline{Q}_3 \cdot$	(3)
Entropy Balance: $\nabla \cdot \rho \underline{v} s = Q_4 \cdot$	(4)
Equations of State: $p = r \rho T \cdot$	(5)
$c_p(T) = c_v(T) + r \cdot$	(6)
$c_v(T) = r \sum_m (N_m/N) [3/2 + R_m/2 + \sum_n \{ (T_{mn}/2T) \operatorname{csch} (T_{mn}/2T) \}^2]$	(7)

Here ρ is fluid density (mass/volume), $\underline{v}$ is fluid velocity, h is fluid enthalpy/mass, p is fluid pressure, s is fluid entropy/mass, T is fluid absolute temperature, $r = p/\rho T$ is the gas constant, $c_p(T) = (\partial h/\partial T)_p$ is specific heat/mass at constant pressure, $c_v(T) = T(\partial s/\partial T)_v$ is specific heat/mass at constant volume, and N_m/N is the fraction of molecules having rotational degrees of excitation freedom R_m and internal vibration characteristic temperatures T_{mn} . The flexible source-sinks $Q_1, Q_2, \underline{Q}_3, Q_4$ provide for piecewise variant transfer of mass, energy, momentum, and entropy respectively.

Program GAS actually uses stream tube surface integral relations obtained by integrating the balance equations over steam-tube volume elements. If we integrate the balance equations over a straight stream-tube volume and utilize the divergence theorem, we obtain

$$(\rho VA)_2 = (\rho VA)_1 + \int_1^2 Q_1 d\tau \quad \text{Mass} \quad (8)$$

$$\rho VA(h + V^2/2)_2 = \rho VA(h + V^2/2)_1 + \int Q_2 d\tau \quad \text{Energy (9)}$$

$$(\rho V^2 A + PA)_2 = (\rho V^2 A + PA)_1 + \int Q_3^{(V)} d\tau \quad \text{Momentum (10)}$$

$$(\rho VAs)_2 = (\rho VAs)_1 + \int Q_4 d\tau \quad \text{Entropy (11)}$$

where A is area normal to the flow, the subscripts 2 and 1 represent the downstream and upstream spatial flow cross sections, and the volume integrals

$$\int Q_1 d\tau, \int Q_2 d\tau, \int Q_3^{(V)} d\tau, \int Q_4 d\tau$$

represent the production per unit time of mass, energy, flow-direction momentum component, and energy, respectively, within the volume element. Program GAS deduces flow and thermodynamic variables at subsequent flow cross sections from the values of three out of the four preceding integrated balance law equations. The choice for a straight, uniform cross section flow segment is the set mass, energy, and momentum.

For the gas duct temperature calculations, we take the mass source $\int Q_1 d\tau$ and the flow-direction momentum component source $\int Q_3^{(V)} d\tau$ to be zero. That is, no mass is injected into the flow, and frictional and other flow direction forces are negligible. A collection of subroutines was developed to compute the (negative) energy source as a result of conductive, convective, and radiative heat losses.

Heat Loss Model

The heat loss from the well-mixed, high Reynolds number flow of the main stream gas is envisioned as occurring by the following processes. (1) Heat is transferred from the gas stream to the duct metal through a stagnant gas boundary layer by a combination convection-conduction process. (2) Heat is transferred through the duct material (or through subsequent layers of duct material) by pure conduction. (3) Heat is lost from the outside duct surfaces to the ambient air by radiation, natural convection, and possibly forced convection.

Specializing to cylindrical ducts allows the use of well-verified data correlations for the processes. The transfer of heat from flowing gas to inside duct surface is described by

$$\dot{q} = \pi D_1 \ell h_1 (t_b - t_1) \quad (12)$$

where h_1 (heat transfer per unit area per unit temperature difference) is taken from the Dittus and Boelter correlation:³

$$\frac{h_1 D_1}{k_b} = 0.023 \left(\frac{D_1 G}{\mu_b} \right)^{0.8} \left(\frac{C_p \mu}{k} \right)_b^{0.4} \quad (13)$$

Here D_1 is inner duct diameter, the subscript b indicates properties of the bulk fluid, ℓ is length of cylindrical duct, and q is rate of heat flow through the cylindrical region of length ℓ . Temperatures t_b and t_1 are temperatures of the bulk fluid (in the duct) and of the inner surface of the duct, respectively. The symbols k , μ , and C_p represent thermal conductivity, viscosity, and heat capacity at constant pressure (energy/(mass degree temperature rise)), respectively, for the bulk fluid. The symbol G represents mass flux in the duct (mass/(area x time)).

The rate of heat flow $\dot{q}$ through any of a sequence of cylindrical annulus conductors is given by

$$\dot{q} = \frac{2\pi \ell k_i (t_{i+1} - t_i)}{\ln(r_{i+1}/r_i)} \quad (14)$$

where ℓ is length, r_{i+1} and r_i are outer and inner radii of the i th cylindrical conducting annulus, and k_i is its thermal conductivity. Temperatures t_{i+1} and t_i are at outer and inner radii of the i th cylindrical annulus.

The heat transfer rate from the outer duct surface has contributions from radiation, natural convection, and forced convection.

$$\dot{q} = \dot{q}_{\text{rad}} + \dot{q}_{\text{nc}} + \dot{q}_{\text{fc}} \quad (15)$$

Natural convection and forced convection heat losses here refer to heat losses by flow driven by the hot duct itself (natural convection) or by other means like wind (forced convection). The radiative contribution to net transfer of heat to surroundings we take as

$$\dot{q}_{\text{rad}} = \sigma 2\pi r_{n+1} \ell \epsilon (t_{n+1}^4 - t_a^4) \quad (16)$$

where σ is the Stefan-Boltzmann radiation constant, r_{n+1} is outer radius of the duct, ϵ is effective emissivity of the duct, and t_{n+1} and t_a are

outer duct surface temperature and ambient temperature, respectively.

The natural convection contribution to the energy flow rate from the outer cylindrical duct surface is given by⁴:

$$\dot{q}_{nc} = \pi D_{n+1} \ell h_{n+1} (t_{n+1} - t_a) , \quad (17)$$

where h_{n+1} (energy transfer per unit area per unit time per degree temperature difference) satisfies the correlation⁴

$$\frac{h_{n+1} D_{n+1}}{k_a} = \lambda \left(\frac{D_{n+1}^3 \rho_a^2 g \beta \Delta t}{\mu_a^2} \left(\frac{C_p \mu}{k} \right)_a \right)^{0.25} , \quad (18)$$

where λ is a slowly varying function of the other factor appearing on the right hand side and has a value of approximately .53 for much of the domain of interest. The subscript a in the preceding expressions (17) and (18) refers to the ambient fluid around the cylindrical duct, which we take to be air. The ambient fluid properties k , C_p , μ , ρ , and β are thermal conductivity, heat capacity at constant pressure, viscosity, density, and coefficient of volumetric expansion. The temperature difference Δt in (18) is the absolute value of $t_{n+1} - t_a$, the outer surface temperature minus the ambient temperature. The g is the acceleration of gravity. The preceding relation is a relation between three dimensionless numbers, the Nusselt, Grashof, and Prandtl numbers.

$$N_{Nu} = \frac{hD}{k} \quad \text{Nusselt number} \quad (19)$$

$$N_{Gr} = \frac{D^3 \rho^2 g \beta \Delta t}{\mu^2} \quad \text{Grashof number} \quad (20)$$

$$N_{Pr} = \frac{C_p \mu}{k} \quad \text{Prandtl number} \quad (21)$$

The natural convective heat loss correlation (18) can be equivalently written

$$N_{Nu} = \lambda (N_{Gr} N_{Pr})^{0.25} . \quad (22)$$

In coding this correlation, a table of Nusselt numbers was built into the program as a function of the product of the Grashof and Prandtl numbers.

The Nusselt number was obtained by a programmed table look up and interpolation. In differentiating the Nusselt number with respect to temperature for purposes of solving resulting heat flow equations iteratively, we used the preceding relation (19) or (22) with λ constant. This natural convection heat flow correlation used for outer cylindrical surfaces has been well verified by a diversity of measurements by several authors.⁴

The forced convection contribution takes into account heat loss from the outer surface from ambient fluid flow other than that driven by heat flow from the duct. This can be wind, air conditioning system air flow, or convective flow driven by heat sources or sinks other than the duct itself. The forced convection heat loss rate from length ℓ of duct is given by

$$\dot{q}_{sc} = \pi D_{n+1} \ell h_{fc} (t_{n+1} - t_a) \quad (23)$$

where h_{fc} satisfies the correlation⁵

$$\frac{h_{fc} D_{n+1}}{k_a} = B \left(\frac{D_{n+1} \rho_a V_a}{\mu_a} \right)^n \quad (24)$$

where B and n are slowly varying functions of the Reynolds number, which is the enclosed expression on the right hand side. In the coding of correlation (24), we actually do a programmed table look up of the Nusselt number (left hand side) as a function of the ambient air Reynolds number.

In steady state heat flow from a duct, energy conservation implies that the radial heat flow $\dot{q}$ for length ℓ is the same at each of the cylindrical surfaces, and hence the same in equations (12), (14), and (15). Equations (12) and (14) imply that

$$\dot{q} = (t_b - t_{n+1}) \left(\frac{1}{\pi D_1 \ell h_1} + \sum_{i=1}^n \frac{\ln(r_{i+1}/r_i)}{2\pi \ell k_i} \right)^{-1} \quad (25)$$

while from equations (15), (16), (17), and (23)

$$\dot{q} = \sigma \pi D_{n+1} \ell \epsilon (t_{n+1}^4 - t_a^4) + \pi D_{n+1} \ell (h_{n+1} + h_{fc}) (t_{n+1} - t_a) \quad (26)$$

Eliminating $\dot{q}$ between equations (25) and (26) gives the equation

$$D_{n+1}(h_{n+1} + h_{fc})(t_{n+1} - t_a) + ((D_1 h_1)^{-1} + \sum_{i=1}^n \frac{\ln(r_{i+1}/r_i)}{2k_i})^{-1} \quad (27)$$

$$(t_{n+1} - t_b) + \sigma D_{n+1} \epsilon (t_{n+1}^4 - t_a^4) = 0$$

Equation (27) is solved iteratively in Program GAS/COOL for t_{n+1} by a modified Newton-Raphson method for each stream-tube segment. The resulting heat flow rate from this segment of length ℓ is then calculated from equation (25). The flow stream segments should be input sufficiently short that the bulk temperature may be approximated as constant within the segment for purposes of calculating the heat loss rate. The outer surface convective heat transfer coefficient h_{n+1} is iteratively modified in the solution, since it is a function of temperature difference Δt .

The heat transfer coefficient h_1 for heat flow through the stagnant boundary layer inside the duct is a function of the flowing gas properties heat capacity at constant pressure C_p , viscosity μ_b , and thermal conductivity k_b . The heat capacity C_p is calculated by Program GAS on a quantum statistical mechanics model. See equations (6) and (7). The viscosity is calculated by a Program GAS subroutine for the flow stream gas mixture on a first order kinetic theory transport model using a Lennard-Jones interaction potential. See handbook of Physics, E.V. Condon and Hugh Odishaw, McGraw-Hill, 1958, Equation 4.27 and vicinity, Section 5-56. We judge the resulting viscosity to be totally satisfactory for our purposes. A subroutine was added to Program GAS/COOL to calculate the thermal conductivity on a similar model. See Condon and Odishaw, Equation 4.39 and vicinity. The thermal conductivity model could be upgraded to include the contribution of the vibrational degrees of freedom to the thermal conductivity. A molecule which migrates from one region to another in a gas can carry internal vibrational and rotational energy in addition to its translational kinetic energy. Translational and rotational but not vibrational contributions have been included. The maximum error in k_b we estimate to be of the order of 28% at high temperatures ($\sim 3000^\circ\text{C}$) and negligible at temperatures less than 1000°C . The corresponding error in h_1 would be on the order of 15% at high

temperature and much less at temperatures less than 1000°C. The resulting error in heat flow rate depends on what share of the "impedance" to heat flow comes from the inner boundary layer. For h_1 and h_{n+1} of the same order, the error in heat loss rate due to neglect of vibrational degrees of freedom in calculating thermal conductivity of the duct gas will be on the order of 5 to 10% at high temperature and much less at temperatures less than 1000°C.

Because the ambient fluid will in cases of interest be air at approximately room temperature, the properties of the ambient fluid were left as input numbers.

Results of Computations

A collection of computer runs with Program GAS/COOL were made with varying input assumptions. One set of runs describes a 4-inch diameter system treated with more extensive approximations in a hand calculation in the United Kingdom Atomic Energy Authority report. A second set of computer runs describes the steady state condition in a British 8-inch diameter duct experiment in which air was electrically heated to 1000 °C and pumped into a duct at controlled flow rate. The third series was an attempt to produce the results obtained at Hanford by H. A. Lee using an actual fire in a hood with gases exhausted through a 4-inch diameter stainless steel duct at a velocity measured with a Pitot tube.

The computer runs were set up to use a consistent system of units in both input and output. We elected to use a foot-slug-second-degree Rankine system (FSSR) rather than say an MKSK system (meter-kilogram-second-degree Kelvin). The input data will be given in the actual input number and units and sometimes other familiar units.

British 4-Inch Diameter Air Duct Hand Calculation¹

The British calculation¹ assumed a mass flow rate equivalent to 50 cubic feet per minute of STP (standard temperature and pressure, $T = 491.688^\circ\text{R} = 273.16^\circ\text{K} = 0^\circ\text{C}$ and $P = 2116. \text{ lbf/ft}^2 = 1 \text{ atmosphere}$) air. Their calculation assumed the thermal conduction of the duct material to be effectively infinitely fast compared with other heat transfer rates. The radiative heat transfer was totally neglected. Inlet temperature was taken as 1000 °C (2291°R). Our corresponding calculations had the same mass flow rate, but with finite conductivity stainless steel walls. We set radiant emissivity to .675 in one run and to zero in another. The emissivity of .675 is high for stainless steel and corresponds to a roughened,

somewhat dirty surface. An emissivity of .675 is approximately correct to slightly low for oxidized steel.

Input data for this case in our computer run were:

$$\begin{aligned}
 t_a &= 559.688^\circ\text{R} = 100^\circ\text{F} && \text{Ambient air temperature} \\
 \rho_a &= 2.204 \times 10^{-3} \text{ slug/ft}^3 && \text{Ambient air density} \\
 C_p &= 6. \times 10^3 \text{ ft. lbs. slug}^{-1} (\text{deg Rankine})^{-1} && \text{Ambient air heat capacity} \\
 \mu_a &= 3.97 \times 10^{-7} \text{ slug ft.}^{-1} \text{ sec}^{-1} && \text{Ambient air viscosity} \\
 k_a &= 3.339 \times 10^{-3} ((\text{ft. lb./sec})/\text{ft}^2)/(\text{degree Rankine/ft.}) && \text{Ambient air thermal conductivity} \\
 r_1 &= .1667 \text{ ft.} = 2 \text{ in.} && \text{Inner duct radius} \\
 r_2 &= .1767 \text{ ft.} = 2.12 \text{ in.} && \text{Outer duct radius} \\
 k_1 &= 1.746 ((\text{ft. lb./sec})/\text{ft}^2)/(\text{degree Rankine/ft.}) && \text{Duct thermal conductivity} \\
 \epsilon &= \begin{cases} .675 & \text{Duct surface thermal emmissivities} \\ .0 & \text{(two cases)} \end{cases} \\
 \rho VA &= 2.067 \times 10^{-3} \text{ slug/sec.} && \text{Mass flow rate} \\
 &&& \text{(Corresponds to 50 ft}^3\text{/minute of STP air)} \\
 t_{b1} &= 2290.7^\circ\text{R} = 1000^\circ\text{C} = 1832^\circ\text{F} && \text{Inlet bulk gas temperature}
 \end{aligned}$$

The resulting duct gas temperature and linear flow velocity are plotted in Figures 1 and 2 as a function of spatial mesh point. Each spatial mesh point represents 4-feet in these figures.

The comparison with the British hand calculated results is summarized in Table 1.

Table 1.

Distance for Specified Temperature Reduction for 50 cfm (STP) Flow in 4-Inch Diameter Duct .	Distance for temperature reduction to 450 °C from 1000 °C .
British hand calculation	30 feet
GAS/COOL - emissivity .675	27.4 feet
GAS/COOL - emissivity 0.	44.3 feet

From comparison of the emissivity zero and emissivity .675 results, it is evident that radiation must be included as a heat loss mechanism. It is not clear why it is the emissivity .675 result which agrees with the approximate hand calculation, when the hand calculation neglected radiative

losses. The comparison with experimental data in the next series should be taken more seriously than this comparison with an approximate calculation.

British 8-Inch Diameter Electrically Heated Air Duct Experiment¹

Gases at 1000 °C were passed into an 8-inch diameter duct and the temperatures at set distances were measured as a function of time. The temperatures for comparison with GAS/COOL results are the asymptotic temperatures, which were approached in about 40 seconds at 5 feet, 200 seconds at 32 feet, and had not quite been reached in 300 seconds at 61 feet from entrance to the duct.

The GAS/COOL input data which differed from those of the 4-inch diameter duct calculation were:

$$\begin{aligned} r_1 &= .33333 \text{ ft.} = 4 \text{ inch} && \text{Duct inner radius} \\ r_2 &= .33854 \text{ ft.} = 4.0625 \text{ in.} && \text{Duct outer radius} \\ \rho VA &= 1.944 \times 10^{-2} \text{ slug/sec} && \text{Mass flow rate} \\ \epsilon &= \begin{cases} .9 \\ .8 \\ 0.0 \end{cases} && \text{Emissivity (3 cases)} \end{aligned}$$

The gas temperature calculated by GAS/COOL is plotted as a function of velocity in Figures 5, 6, and 7 for emissivities .9, .8, and 0., respectively. Each mesh point represents 2 feet in these figures. For comparison, we have read from the British temperature vs. time plots the asymptotic temperature of the gas at 32 feet and the near asymptotic temperature (300 seconds after flow start) at 61 feet. The comparisons of the calculated temperatures and experimentally observed temperatures at 300 seconds are summarized in Table 2.

Table 2. Experimental and Calculated Temperatures for Electrically Heated Air in 8-Inch Diameter Ducts

	<u>Temperature at 32 feet</u>	<u>Temperature at 61 feet</u>
Experiment	780 °C, 1438 °F, 1898 °R	575 °C, 1067 °F, 1527 °R
GAS/COOL - Emissivity .9	733 °C, 1351 °F, 1810 °R	568 °C, 1054 °F, 1514 °R
GAS/COOL - Emissivity .8	738 °C, 1316 °F, 1821 °R	576 °C, 1069 °F, 1528 °R
GAS/COOL - Emissivity 0.	871 °C, 1599 °F, 2059 °R	770 °C, 1418 °F, 1878 °R

Agreement with the experimentally observed temperatures is fairly good for the emissivity .8 case. An emissivity of .7 should probably have been used for oxidized cold steel ducts in the British experiments. This would be consistent with values reported by Adams for emissivities:

Rolled sheet iron at 70°F	.66
Oxidized iron at 212°F	.74
Steel plate, rough, 100° to 700°F	.94 to .97

The calculated temperatures at 32 feet would agree very well with the experimentally observed ones with the use of an emissivity .7. The good agreement between calculated temperature and experimentally observed temperature at 61 feet obtained with assumed emissivity .8 would still persist with emissivity .7.

Hanford Flame Hood Experiment With 4-Inch Diameter Stainless Steel Duct

In experiments performed at the USAEC Hanford reservation, combustible materials were burned in an outdoor flame hood with exhaust gases removed through a 4-inch diameter stainless steel duct. Flow rate through the duct was controlled by varying the speed of the centrifugal blower pulling it. The flow rate was deduced from Pitot tube readings at a location approximately 8 feet from the flame hood. The temperatures in the approximately 30 feet of duct were measured at a location just downstream from a primary filter which removed debris from the exhaust gases and also at a location just upstream from the secondary filter and centrifugal pump. The volume flow rate reported was approximately 50 cubic feet per minute at this location. Owing to the nature of the fuel used (representative of glove box combustibles), constant temperature could not be achieved, but a mean temperature on the order of 650 °F was maintained for on the order of 20 minutes with typical fluctuations of approximately 50 °F during this time. The reported ambient temperature was 64 °F.

A computer run was set up to iteratively adjust the mass flow rate at duct entrance to give the reported volumetric flow rate at a location within the duct. An ambient temperature of 100 °F was used in the computer run rather than the 64 °F value reported for the experiment. This will result in an understatement of the cooling rate for the experiment but

should result in values more typical of those likely to occur in indoor glove box or hot cell fires. Data for this case differing from that of previous cases are summarized below:

t_{bl} = 1109.68 °R	Duct inlet temperature
r_1 = .16667 ft. = 2. in.	Duct inner radius
r_2 = .1767 ft. = 212 in.	Duct outer radius
ρVA = 1.288×10^{-3} slug/sec	Mass flow rate
(Corresponds to 50 cfm at a point within the duct)	

The four cases treated include the follow

Case 1:	Emissivity	.675
	Wind Velocity	0.0
	Spatial mesh length	4. ft.
	Air mixture	
Case 2:	Emissivity	.675
	Wind velocity	29.33 ft./sec (20 mph)
	Spatial mesh length	4. ft.
	Air mixture	
Case 3:	Emissivity	.5
	Wind velocity	0.0
	Spatial mesh length	4. ft.
	Air mixture	
Case 4:	Emissivity	.675
	Wind velocity	0.0
	Spatial mesh length	4. ft.
	Dodecane combustion gas rather than air -	
	Number fractions:	
	H ₂ O	.184
	N ₂	.713
	CO ₂	.095
	Argon	.008

The resulting temperature as a function of mesh point and hence of length from GAS/COOL calculations is displayed in Figures 8 to 12. For single number comparison of cases we present the temperatures at 24, 32, and 50 feet in Table 3.

Table 3. Temperatures in Computational Simulation of the Hanford Flame Hood Duct Gas Experiment. Ambient temperature 100 °F, inlet temperature 650 °F assumed. See preceding text for detailed assumptions.

	<u>24 ft.</u>	<u>32 ft.</u>	<u>50 ft.</u>
Case 1	329. °F	225. °F	197. °F
$\epsilon = .675$, WV = 0, Air			
Case 2	260. °F	207. °F	144 °F
$\epsilon = .675$, WV = 29.33 ft/sec			
Case 3	338. °F	284. °F	205. °F
$\epsilon = .5$, WV = 0			
Case 4	333. °F	278. °F	200. °F
$\epsilon = .675$, WV = 0, Combustion product gas			

From Table 3 we see much less sensitivity of calculated temperatures to assumed emissivity for the 650 °F entry temperature than was the case for the 1000 °C entry temperature cases where radiation played a more important role. We summarize in Table 4 the emissivity information from Heat Transmission by McAdams.

Table 4. Stainless Steel Emissivities (from William H. McAdams, Heat Transmission, McGraw-Hill, New York, 1954)

<u>Type</u>	<u>Temperature</u>	<u>Emissivity</u>
Type 316, cleaned	75 °F	.28
316, repeated heating	450 - 1600 °F	.57 - .66
304, 42 hr. at 980 F	420 - 980 °F	.62 - .73
310, furnace service	420 - 980 °F	.90 - .97
Allegheny #4, polished	212 °F	.13

The value .5 for emissivity in the Hanford experiment seems reasonable.

The effect of change in gas composition from air to a composition somewhat representative of dodecane (or any other light hydrocarbons) combustion did not produce a dramatic change, but the contribution to the viscosity and thermal conductivity from non-spherical molecules like water is not adequately represented by the Lennard-Jones interaction model.

Table 3 results do show that wind can cause a significant change in the duct gas cooling.

In the Hanford experiment, the duct gas temperature at a point approximately 20 feet from the duct entrance with 650°F duct entry (after filter) temperature was $180^{\circ} \pm 15^{\circ}\text{F}$. This is lower than the calculated temperature. A major part of this difference may stem from the assumption in the calculations of an ambient temperature of 100°F, rather than the 64°F present in the experiment. Transient effects in the experiment may also contribute to the discrepancy. Inclusion of vibrational contribution to duct gas thermal conductivity would be a small effect but in the direction of better agreement.

Evaluation and Recommendations

The reasonable agreement with the British 8-inch duct experiment and the fair agreement with the Hanford 4-inch flame hood duct experiment indicates the adequacy of the correlations used for horizontal cylindrical ducts whose position is far enough from obstructions to avoid interference with the convective flow pattern. The improvement of the thermal conductivity calculation for the duct gases will move calculated results toward the experimental ones.

A possible extension of the computational code would be the inclusion of vertical duct sections (easy) and of other cross section ducts (more approximately). It is not known how valuable these features would be.

Another major step would be the systematization of the results for convenient use in hot cell and glove box duct design. This may be possible as an alternative to running the computer program developed here as the design tool.

Footnotes and References

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Figure 2. UKAEA 4" Duct
Flow rate 50 cfm of STP air

Emissivity .675

$T_{in} = 1000\text{ }^{\circ}\text{C} = 2292\text{ }^{\circ}\text{R} = 1832\text{ }^{\circ}\text{F}$

$\Delta L = 4\text{ ft/mesh point (horizontal axis)}$

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Figure 4. UKAEA 4" Duct
Flow rate 50 cfm of STP air
Wind velocity 0.
 $T_{in} = 1000\text{ }^{\circ}\text{C} = 2292\text{ }^{\circ}\text{R} = 1832\text{ }^{\circ}\text{F}$
Emissivity 0.
 $\Delta L = 4\text{ ft/mesh point (horizontal axis)}$

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Figure 5.

UKAEA 8" Duct Experiment Simulation

Emissivity .9

Wind velocity 0.

Flow rate 465 cfm of STP air

Inlet temperature

$T_{in} = 1000\text{ }^{\circ}\text{C} = 2291\text{ }^{\circ}\text{R} = 1832\text{ }^{\circ}\text{F}$

$\Delta L = 4\text{ ft/mesh point (horizontal axis)}$

BNW PROGRAM GAS-R1

Figure 6.

UKAEA 8" Duct Experiment Simulation

Emissivity .8

Wind velocity 0.

Flow rate 465 cfm of STP air

Inlet temperature

$T_{in} = 1000\text{ }^{\circ}\text{C} = 2291\text{ }^{\circ}\text{R} = 1832\text{ }^{\circ}\text{F}$

$\Delta L = 4\text{ ft/mesh point (horizontal axis)}$

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Figure 8. Hanford 4" Duct

Emissivity .675

Flow rate 50 cfm at 8. ft down duct

$T_{in} = 650\text{ }^{\circ}\text{F} = 1110\text{ }^{\circ}\text{R}$

$T_{ambient} = 100\text{ }^{\circ}\text{F}$

Wind velocity 0.

$\Delta L = 2\text{ ft/mesh point (horizontal axis)}$

BNW PROGRAM GAS-R1

Figure 9. Hanford 4" Duct

Emissivity .675

Wind velocity 29.33 ft/sec = 20 mph

Flow rate 50 cfm at 8. ft down duct

$T_{in} = 650\text{ }^{\circ}\text{F} = 1110\text{ }^{\circ}\text{R}$

$T_{ambient} = 100\text{ }^{\circ}\text{F}$

$\Delta L = 2\text{ ft/mesh point (horizontal axis)}$

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Figure 10. Hanford 4" Duct
 Emissivity .5
 Wind velocity 0.
 Flow rate 50 cfm at 8 ft down duct
 $T_{in} = 650\text{ }^{\circ}\text{F} = 1110\text{ }^{\circ}\text{R}$
 $T_{ambient} = 100\text{ }^{\circ}\text{F}$
 $\Delta L = 2\text{ ft/mesh point (horizontal axis)}$

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Figure 12. Hanford 4" Duct

Emissivity .675

Flow rate 50 cfm at 8 ft down duct

$T_{in} = 650\text{ }^{\circ}\text{F} = 1110\text{ }^{\circ}\text{R}$

$T_{ambient} = 100\text{ }^{\circ}\text{F}$

Wind velocity 0.

$\Delta L = 2\text{ ft/mesh point (horizontal axis)}$

BNW PROGRAM GAS-R1

Figure 13. Hanford 4" Duct

Emissivity .675

Wind velocity 29.33 ft/sec = 20 mph

Flow rate 50 cfm at .8 ft down duct

$T_{in} = 650\text{ }^{\circ}\text{F} = 1110\text{ }^{\circ}\text{R}$

$T_{ambient} = 100\text{ }^{\circ}\text{F}$

$\Delta L = 2\text{ ft/mesh point (horizontal axis)}$

BNW PROGRAM GAS-R1

Figure 14. Hanford 4" Duct

Emissivity .5

Wind velocity 0.

Flow rate 50 cfm at 8 ft down duct

$T_{in} = 650\text{ }^{\circ}\text{F} = 1110\text{ }^{\circ}\text{R}$

$T_{ambient} = 100\text{ }^{\circ}\text{F}$

$\Delta L = 2\text{ ft/mesh point (horizontal axis)}$

BNW PROGRAM GAS-R1

Figure 15. Hanford 4" Duct
 Emissivity .675
 Wind velocity 0.
 Flow rate 50 cfm at 8 ft down duct
 Combustion product gas flow
 $T_{in} = 650\text{ }^{\circ}\text{F} = 1110\text{ }^{\circ}\text{R}$
 $T_{ambient} = 100\text{ }^{\circ}\text{F}$
 $\Delta L = 2\text{ ft/mesh point (horizontal axis)}$

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