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A review of empirical flame impingement heat transfer correlations

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Impinging flames are used in industrial heating and melting, safety research, and aerospace applications. Multiple modes of heat transfer are commonly important in those processes. However, the detailed heat transfer mechanisms are not well understood. The available semianalytical heat transfer solutions have only limited applicability. Therefore, researchers and designers have either made measurements or used empirical correlations to determine the heat flux rates. Here, the empirical correlations are reviewed. The correlations are first arranged by the flow geometry. Four geometries have been studied. These include flames impinging: (1) normal to cylinders in cross-flow; (2) normal to hemispherically nosed cylinders; (3) normal to plane surfaces; and (4) along plane surfaces. The correlations are then arranged by the size of the region on the target under consideration. Correlations that apply to a small region on the target, such as the stagnation point, are referred to as the local heat flux. Correlations that apply to a large region of the target, such as the entire body of a cylindrical target, are referred to as the average heat flux. Next, the correlations are arranged by the type of heat transfer. Correlations for natural and forced convection, thermochemical heat release (TCHR), radiation, forced convection with TCHR, and forced convection with radiation have been reported for some combinations of geometry and operating conditions. Finally, the correlations are arranged by the flow type, either laminar or turbulent. Correlations do not exist for many combinations of geometry and operating conditions; therefore, recommendations are given for further research.

Keywords: flame impingement; heat transfer; empirical correlations; forced convection; natural convection; radiation

Introduction

Flame impingement heating is important in many processes. Low-intensity flames are used to heat metal bars prior to rolling into sheets. They are also used in safety research. In one application, they are used to quantify the heating rate caused by buoyant fires impinging on walls and ceilings. In others, they are used to simulate extensive fires, impinging on structural elements. Such high-velocity flames may be caused by ruptured fuel pipes. High-intensity flame impingement arises in many applications. They are commonly used to melt scrap metal and to shape glass. They have also been used in recent years to produce synthetic diamond coatings by chemical vapor deposition. They have been used in thermal spallation, where a supersonic velocity flame bores holes through rock. The rock fragmentation is caused by large, transient, thermal stresses. This is attributable to high heat fluxes impinging on cold surfaces. High-intensity flame impingement has also been used to study the extremely high heat fluxes encountered by space vehicles, re-entering the Earth's atmosphere. High fluxes are caused by the hypersonic impact gas velocities, which ionize the highly shocked atmospheric gases.

The primary objective of this review is to provide designers and researchers with available empirical heat transfer correlations for flame impingement on target surfaces. Another objective is to provide a comprehensive reference for this subject and to identify areas requiring further research.

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Heat transfer mechanisms

Convection, thermochemical heat release, radiation, and condensation are the heat transfer mechanisms identified in the flame impingement studies reviewed here. The relative importance of each heat transfer mechanisms depends on the experimental conditions. For example, when targets have been located inside a hot furnace, radiation from the environment; that is, the furnace walls, has been very important. However, for targets not located in any enclosure, radiation from the ambient environment is negligible. In many circumstances, multiple mechanisms have been combined into one heat transfer correlation. This has commonly been done for forced convection coupled with thermo-

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chemical heat release (TCHR), primarily because of the difficulty in experimentally separating these mechanisms. In some studies, heat transfer mechanisms have been completely ignored, without sufficient justification. These are discussed by Baukal et al. (1996). The following section is a general discussion of these mechanisms. A detailed presentation of specific correlations, the main subject of this work, follows in a later section.

Convection

Forced convection. This is the predominant mechanism in low-intensity flames, with negligible chemical dissociation. Laminar flames were investigated in several studies (e.g., Jackson and Kilham 1956). Turbulent flames were tested in many studies (e.g., Giedt et al. 1960). There, the turbulence level directly enhances the importance of forced convection.

Natural convection. You (1985) correlated the heat transfer for a buoyant flame in terms of the Rayleigh number. The correlations are given later in Equation 27. Buoyancy was important in their study because of the low burner exit velocities.

Thermochemical heat release (TCHR)

This mechanism refers to the energy release caused by exothermic chemical reactions. Hot, dissociated combustion products impinge on a cool surface. The radical species diffuse along concentration gradients toward lower temperature regions. Then, they exothermically recombine with other species to form more stable molecules. The new molecular components are thermodynamically preferred at the lower temperatures. For example, there may be significant amounts of unburned fuel, in the form of CO and H_2, produced by O_2/CH_4 flames. These flames usually have much higher temperatures than air/CH_4 flames. This is caused by the removal of the N_2, which acts as a diluent to moderate the flame temperatures. The unburned fuel components are thermodynamically preferred at the high temperatures. However, at temperatures below about 1600K, these radical species cool and react to form CO_2 and H_2O. They simultaneously release energy. Thermochemical heat release is commonly very important in high-temperature flames, because the level of dissociation increases with temperature. Three types of TCHR have been identified: equilibrium, catalytic, and mixed. These are discussed in Baukal et al. (1996).

Radiation heat transfer

Three types of thermal radiation are involved in the flame impingement studies reviewed here. They include nonluminous, luminous, and emission from surfaces.

Nonluminous radiation. Gaseous combustion products, such as carbon dioxide and water vapor, produce nonluminous radiation. Radiation from CO_2 and H_2O has been extensively studied because of its important in combustion. Such radiation depends on the gas temperature, partial pressure, and concentration of each gas species and the optical path length through the gas. In some studies, nonluminous radiation amounted to a significant fraction of the total heat flux to the target. Kilham's (1949) data indicated that flame radiation accounted for between 5 and 16% of the total heat flux. Jackson and Kilham (1956) found that the measured flame radiation accounted for up to 5% of the total heat flux. Ivernel and Vernotte (1979) calculated nonluminous flame radiation effects. They accounted for up to 34% of the total heat flux. The measured CO_2 and H_2O concentrations in the furnace, together with the Hottel curves

Notation

c_p	specific heat
d	diameter
g	gravity
h^C	chemical enthalpy
h^S	sensible enthalpy $= \int c_p\, dt$
h^T	total enthalpy $= h^C + h^S$
k	thermal conductivity
l	length
L	dimensionless distance between burner and stagnation body $= l_j/d_n$
Le	Lewis number
m	mole fraction
Ma	Mach number
Nu	Nusselt number
Pr	Prandtl number
q''	heat flux
q_f	burner firing rate, kW
Q_H	H atom heat of recombination
r	radial distance from burner centerline
R	dimensionless radial distance from the burner centerline $= r/d_n$
Ra	Rayleigh number
Re	Reynolds number
t	temperature
Tu	turbulence intensity $= \dfrac{\sqrt{v'^2}}{\bar{v}}$
v	velocity
v'	fluctuating velocity
x	axial distance from the burner toward the stagnation point
X	dimensionless axial distance from the burner toward the stagnation point $= x/d_n$

Greek

$\tilde{\beta}$	coefficient of thermal expansion
μ	absolute or dynamic viscosity
ρ	density
ϕ	equivalence ratio = (stoichiometric oxygen/fuel volume ratio)/(actual oxygen/fuel volume ratio)
Ω	oxygen enrichment ratio = (oxygen volume in the oxidizer/total oxidizer volume)

Subscripts

b	body or target
conv	convection
e	edge of boundary layer
∞	ambient
j	jet
max	maximum at a given location
n	nozzle
r	based on the radial distance r
rad	radiation
rec	evaluated at the recovery temperature, Equation 1f
ref	evaluated at the reference temperature, Equation 1e
s	stagnation point
TCHR	thermochemical heat release
v	position in the jet where the flow is fully developed
w	wall

(Hottel and Sarofim 1967), were used to determine the flame emissivity.

In some studies, nonluminous radiation was negligible. Giedt et al. (1960) measured and calculated the nonluminous radiation to be less than 2% of the total heat flux. Woodruff and Giedt (1966) measured the nonluminous radiation using the same configuration as Giedt et al. The effect was negligible. Shorin and Pechurkin (1968) also experimentally determined that radiation was negligible. Hoogendoorn et al. (1978) found that radiation did not exceed 5% of the total heat flux. No measurements or calculations were given. Davies (1979) calculated that nonluminous radiation account for only 2% of the total heat flux.

Luminous radiation. This occurs when soot particles are produced. These radiate approximately as a blackbody. This mechanism is very important for such liquid and solid fuels as oil and coal. It is not commonly significant for gaseous fuels such as methane. You (1985) measured radiation from pure diffusion ($\phi = \infty$) natural gas flames. Much of this radiation is expected to have been luminous because of the nature of buoyant, pure diffusion flames. Radiation accounted for 13–26% of the total flux at the stagnation point. Hustad and co-workers (1991) calculated the radiation flux to the stagnation point of a cylinder in cross-flow to be from 7–14% and 20–40% of the total flux for CH_4 and C_3H_8 pure diffusion flames, respectively. Luminous radiation may have been important in other studies, where the flames were very fuel rich (e.g., Buhr et al. 1973). However, no measurements or calculations were reported.

Surface radiant emission. Beér and Chigier (1968) studied stoichiometric air/coke oven gas flames impinging on the hearth of a furnace. The measured radiation was at least 10% of the total heat flux. This radiation was actually the combination of the furnace wall radiant emission and the radiation from the flame. However, the radiation from that type of flame is usually negligible, as previously discussed.

Vizioz and Lowes (1971) and Smith and Lowes (1974) studied flame impingement onto a cooled plane target located inside a hot furnace. Radiation and convection were often determined to be of comparable magnitude. Vizioz and Lowes measured radiation as being 4–100% of the total heat flux. Smith and Lowes calculated total radiation, using Hottel's zone method (Hottel and Sarofim 1967). It accounted for 30–43% and 10–17% of the total heat transfer to water-cooled and air-cooled flat plates, respectively.

Matsuo et al. (1978) studies flame impingement on a metal part, inside a hot furnace. Coke oven gas was combusted with air that had been preheated to 200°C. The top of the part was exposed to the impinging flame. The rest was exposed to the radiation from the furnace walls. Furnace radiation was the dominant heat transfer mechanism: (1) when the distance between the burner and the slab was large; (2) at high slab surface temperatures; and (3) at large R. Ivernel and Vernotte (1979) calculated that radiation from the furnace walls accounted for up to 42% of the total heat flux.

You (1985) studied a buoyant unenclosed flame impinging on a flat plate. The convective and total heat flux rates were measured using gauges plated with gold and black foils, respectively. By subtracting the convection effect from the total heat flux, radiation was calculated to be as much as 35% of the convective flux. Thermal radiation dropped off rapidly as R increased, for a fixed L.

Water vapor condensation

For the studies reviewed here, that of Hargrave et al. (1987) was the only one that considered the effect of condensation. They attempted to eliminate it by careful experimental design. Flowing ethylene glycol was used to cool the inside wall of a cylinder and a heminosed cylinder target. Glycol has a higher boiling point than that of water. Therefore, the target surface temperature could be maintained at a high enough level to prevent water vapor in the combustion products from condensing on the target. Water-cooled targets have been used in many studies (e.g., Anderson and Stresino 1963). The surface temperature was maintained below the boiling point of the water. This prevented cooling water boiling inside the target. However, the water vapor in the combustion products may have condensed on the cool target surface. For air/fuel flames, the water content in the combustion products is relatively small. However, in O_2/CH_4 flames, for example, these products are two-thirds water. Condensation may have been important in those studies. This effect has not been quantified in any study.

Thermophysical properties

The transport and thermodynamic properties used in the correlations to be presented below are discussed in this section. These properties have been evaluated, using several rules. These properties include, for example, the specific heat, thermal conductivity, enthalpy, and the viscosity of the combustion products. The nomenclature is shown in Figure 1. One common rule has been to evaluate the properties, using the gas temperature at the edge of the stagnation zone and the boundary layer, t_e:

$$p_e = p(t_e) \tag{1a}$$

where p is the property being evaluated. Another common rule has been to evaluate the properties at the wall temperature of the target t_w

$$p_w = p(t_w) \tag{1b}$$

Some properties have been evaluated at the film temperature. This is the mean temperature between the edge of the stagnation zone and the wall temperature. The property is evaluated using

$$\bar{p} = p[(t_e + t_w)/2] \tag{1c}$$

Some studies (e.g., Hargrave et al. 1987) have used a weighted average, over the temperature range between the edge of the stagnation zone and the wall temperatures, as

$$\bar{\bar{p}} = \int_{t_w}^{t_e} p \, dt \Big/ (t_e - t_w) \tag{1d}$$

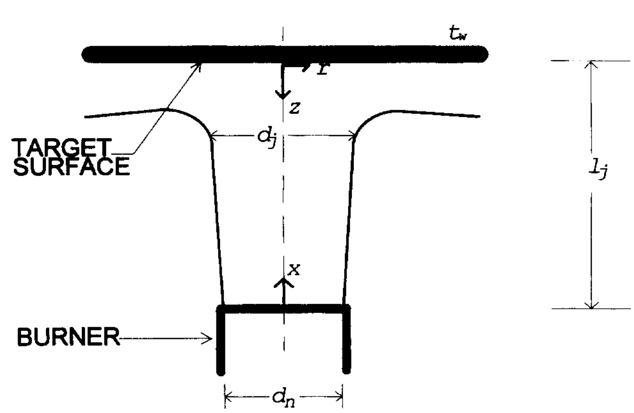

Figure 1 Nomenclature for flame impinging normal to a plane surface

Hoogendoorn et al. (1978) and Popiel et al. (1980) used the reference temperature method recommended by Eckert (1956):

$$p_{\text{ref}} = p[t_e + 0.5(t_w - t_e) + 0.22(t_{\text{rec}} - t_e)] \tag{1e}$$

where

$$t_{\text{rec}} = t_e + v_e^2 \text{Pr}_e^{0.5} / 2c_{p_e} \tag{1f}$$

For low-speed flows, the recovery temperature is about the same as t_e. Therefore, p_{ref} is nearly the same as $\bar{p}$ in Equation 1c. Beér and Chigier (1968) evaluated the properties at the maximum temperature measured at a given location in the flame jet:

$$p_{\max} = p(t_{\max}) \tag{1g}$$

Definitions of dimensionless quantities to be introduced below, such as Re, Nu, and Pr, will require specification of a characteristic length scale and characteristic transport properties. The transport properties will, in turn, require specification of the rule by which they are evaluated; e.g., use of a specific characteristic temperature of the material. When only transport properties; i.e., material temperature, are required, this is indicated by a single subscript; e.g., Pr_e (i.e., Pr evaluated at t_e). When both characteristic length and property temperature are required, these are indicated by the first and second of two subscripts, respectively. For example, $\text{Re}_{n,e}$ is shorthand notation for $\text{Re}_{n,e} = \rho_e v_e d_n / \mu_e$. For this example, the characteristic dimension is d_n. That dimension is an important parameter in correlating the heat transfer. There are many possible lengths that may be used. The most common one has been the burner nozzle diameter d_n. Another has been the axial distance from the nozzle edge to the surface stagnation point l_j. For cylindrical and heminosed cylindrical targets (see Figures 2 and 3), the diameter of the body d_b, has been used. For plane targets, the distance along the surface r has also been used. Smith and Lowes (1974) used the width of the cooling channel in the target l_c. Kataoka et al. (1984) used the radius where the velocity was half the axial ($r = 0$) velocity $r_{1/2v}$.

Empirical heat transfer correlations

These have been generally given in two forms. The first is the Nusselt number, which is a dimensionless heat transfer coefficient. In the studies considered here, it has appeared as $\text{Nu} \approx a\text{Pr}^b\text{Re}^c$, where a, b, and c are constants. The second form has been directly in terms of the heat flux, q''. These two forms are related as follows: $q'' = (k/d)\{\text{Nu}\}\Delta t$, where k is the thermal conductivity of the fluid, and d is a characteristic dimension.

For consistency, most of the correlations given here are written in terms of q''. This makes it easier to directly compare equations. Also, the driving force, or the potential for heat transfer, is explicitly given in the second form. In some correlations, this potential is the temperature difference. In others, it is the enthalpy difference. In the first form, the Nu formulation, the potential is not given explicitly. For the correlations converted from the first form to the second form (i.e., Nu to q''), the Nu relationship is shown inside curved brackets { }.

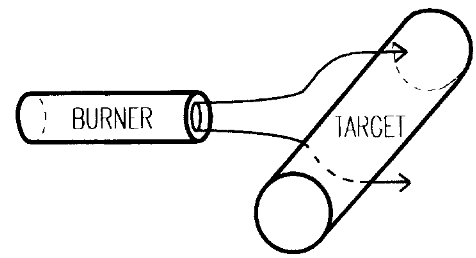

Figure 2 Flame impinging normal to a cylinder in cross-flow

This section is arranged to aid the designer or researcher to find the appropriate correlation for a specific set of conditions. The primary consideration is to ensure that the heat transfer correlation applies to the chosen geometric configuration. The next consideration is the location on the target where the desired flux is to be calculated. This has typically been at a specific location, such as the stagnation point. This is referred to as a local heat flux. Correlations have also been determined for the average heat flux over a given portion of the target. Next, the correlations are arranged in terms of the heat transfer mechanism(s). These include convection, TCHR, radiation, and combinations of these. Finally, the correlations are arranged by the flow type (laminar or turbulent). In some studies (e.g., Fairweather et al. 1984), correlations were given for both laminar and turbulent flows. However, in other studies, the flow type was not specified.

Flames impinging normal to a cylinder

This refers to flames impinging perpendicular to a cylinder, as in Figure 2, a cylinder in cross-flow. The experimental conditions for previous studies, using this geometry, are given in Table 1. The table also shows which heat transfer mechanisms were correlated in those studies.

Local convection heat transfer

Laminar and turbulent flows. Hustad and co-workers (1991) measured the radiation and the total heat fluxes for both laminar and turbulent flows. The radiation flux was also calculated. The convection heat transfer was calculated by subtracting the calculated radiation from the measured heat flux. Several correlations from the literature were tested for the convective portion of the heat transfer. The best match with the experimental convection at the stagnation point was the correlation from Zukauskas and

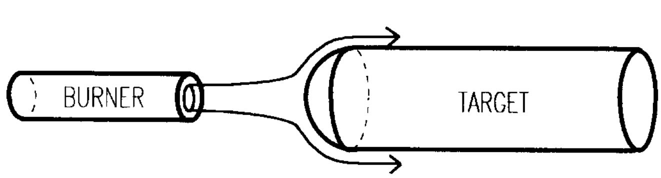

Figure 3 Flame impinging normal to a hemi-nosed cylinder in parallel flow

Table 1 Experimental conditions for flames impinging normal to cylinders

Location	Heat transfer mechanisms: Forced conv.	TCHR	Radiation	Oxidizer	Re_n	ϕ	Fuel(s)	Reference
Local	√			air	2000–8000	0.8–1.2	CH_4	Hargrave et al. 1987
	√	?	√	air	200–3600	∞	CH_4, C_3H_8	Hustad & co-workers 1991
Average	√			air	not given	1.25	C_3H_8	Fells & Harker 1968
	√			air	laminar[a]	1.0, 1.19	CO	Kilham 1949
	√			air	200–3600	∞	CH_4, C_3H_8	Hustad & co-workers 1991
	√			air	not given	0.6–1.3	natural gas	Davies 1979
	√	√		air/O_2		0.5–1.3		
	√	√		O_2		0.5–1.16		
	√			air	laminar[a]	1.0	H_2	Jackson & Kilham 1956
	√	√		O_2		1.0	H_2	
	√	√		O_2		1.0, 2.0	CO	
	√	√		air/O_2	not given	lean-rich[a]	town gas	Davies 1965
	√	√		O_2		1.0		
Maximum	√	?	√	air	26,000[b]	∞	CH_4	Hustad et al. 1992
					54,000, 330,000[b]		C_3H_8	
					66,000[b]		C_4H_{10}	

[a]According to author (only qualitative).
[b]Calculated from data in reference.

Ziugzda (1985), assuming Tu = 0.02:

$$q''_{s,\mathrm{conv}} = \frac{k_e}{d_b}\left\{0.41\mathrm{Re}_{b,e}^{0.6}\mathrm{Pr}_e^{0.35}\mathrm{Tu}^{0.15}\left(\frac{\mathrm{Pr}_e}{\mathrm{Pr}_w}\right)^{0.25}\right\}(t_e - t_w) \tag{2}$$

For a pure fuel jet, there may have been significant quantities of uncombusted species in the flame at the impingement point. Therefore, TCHR may have been important, but was not discussed by Hustad and co-workers. If TCHR was present, it would have been included, with forced convection, in Equation 2.

Turbulent flows. Hargrave et al. (1987) studied the effects of turbulence. The following correlation was determined:

$$q''_{s,\mathrm{conv}} = \frac{\bar{\bar{k}}}{d_b}\left\{\overline{\overline{\mathrm{Re}}}_b^{0.5}\left[1.071 + 4.669\left(\frac{\mathrm{Tu}\overline{\overline{\mathrm{Re}}}_b^{0.5}}{100}\right) - 7.388\left(\frac{\mathrm{Tu}\overline{\overline{\mathrm{Re}}}_b^{0.5}}{100}\right)^2\right]\right\}(t_e - t_w) \tag{3}$$

This correlation underpredicted the data by up to 14%.

Average convection heat transfer

Laminar flows. In related studies, Kilham (1949) and Jackson and Kilham (1956) measured the total and radiative heat fluxes. The convective heat transfer was calculated by subtracting the radiation from the total flux. Kilham studied air/CO flames. The convection was correlated by:

$$q''_{b,\mathrm{conv}} = \frac{\bar{k}}{d_b}\{\overline{\mathrm{Pr}}^{0.3}(0.35 + 0.47\overline{\mathrm{Re}}_b^{0.52})\}(t_e - t_w) \tag{4a}$$

Jackson and Kilham studied air/H_2, O_2/H_2, and O_2/CO flames. The convection was correlated by

$$q''_{b,\mathrm{conv}} = \frac{\bar{k}}{d_b}\{\overline{\mathrm{Pr}}^{0.3}(0.35 + 0.50\overline{\mathrm{Re}}_b^{0.52})\}(t_e - t_w) \tag{4b}$$

Both equations were valid for $60 \le \mathrm{Re}_b \le 230$. The constant multiplying Re_b varied slightly in the two studies. McAdams's (1954) correlation for this geometry, but for nonreacting jet impingement, had a multiplier of 0.56. In the O_2/H_2 and O_2/CO flames used in Jackson and Kilham's study, there may have been significant quantities of dissociated species. Therefore, TCHR may have been important, but was not discussed. If TCHR was present, it would have been included in Equation 4b. However, a larger difference in Equations 4a, b would have been expected if TCHR was significant.

Laminar and turbulent flows. Hustad and co-workers (1991) measured the total heat flux. The radiation heat flux was assumed to be uniformly distributed around the cylinder. The forced convection was calculated by subtracting the calculated radiation from the measured total heat flux. At a turbulent intensity of 2%, the best fit of the data was given by

$$q''_{b,\mathrm{conv}} = \frac{k_e}{d_b}\left\{0.23\mathrm{Re}_{b,e}^{0.6}\mathrm{Pr}_e^{0.35}\mathrm{Tu}^{0.15}\left(\frac{\mathrm{Pr}_e}{\mathrm{Pr}_w}\right)^{0.25}\right\}(t_e - t_w) \tag{5}$$

Flow type unspecified. Fells and Harker (1968) empirically determined:

$$q''_{b,\mathrm{conv}} = \frac{\bar{\bar{k}}}{d_b}\left\{(0.573 - 0.0179\, l_j)\overline{\overline{\mathrm{Re}}}_b^{0.5}\right\}(t_e - t_w) \tag{6}$$

for C_3H_8 flames. This correlated the experimental data within 10%. Davies (1979) calculated the forced convection heat transfer using an equation developed by Conolly and Davies (1972), for a water-cooled tube in cross-flow:

$$q''_{b,\mathrm{conv}} = (\bar{k}/d_b)\{1.32\overline{\mathrm{Pr}}^{0.4}\overline{\mathrm{Re}}_b^{0.5}\}(t_e - t_w) \tag{7}$$

Average convection heat transfer with thermochemical heat release

Flow type unspecified. Davies (1965) correlated heat flux data using an equation from McAdams (1954), assuming Pr = 0.7

$$q''_{b,\mathrm{conv+TCHR}} = (\bar{\bar{k}}/d_b)\left\{0.615\overline{\overline{\mathrm{Re}}}_b^{0.466}\right\}(t_e - t_w) \tag{8}$$

Average radiation heat transfer

Laminar and turbulent flows. Hustad and co-workers (1991) studies pure diffusion flames. The radiation from the CH_4 flames was given as

$$q''_{b,\mathrm{rad}} = 5 + 3.9(l_j - x_{\mathrm{liftoff}}) \quad (\mathrm{kW/m^2}) \tag{9a}$$

The radiation for the C_3H_8 flames was given as

$$q''_{b,\mathrm{rad}} = 10 + 7.8(l_j - x_{\mathrm{liftoff}}) \quad (\mathrm{kW/m^2}) \tag{9b}$$

These were based on an analytical model of the flame as a radiating cylinder. The liftoff distance, x_{liftoff}, is the length between the nozzle and the start of the flame. The length $l_j - x_{\mathrm{liftoff}}$ was termed the radiation height. For a conventional burner, x_{liftoff} is zero, because the flame is attached to the nozzle exit. The distances, used in Equations (9a, b), were in meters. In those studies, the fuel gas velocities at the nozzle outlet were Ma ≥ 0.3. This equates to a minimum outlet velocity of 134 and 82 m/s for CH_4 and C_3H_8, respectively. The burning velocity in air for CH_4 and C_3H_8 is 0.37 and 0.41 m/s, respectively (Lewis and von Elbe 1987). Therefore, the flames were not attached to the burner, because the exit velocities were higher than the flame speed. This produced a so-called unattached or lifted flame. The flame starts where the local gas velocity is the same as the flame speed.

Maximum convection and radiation heat transfer

Turbulent flows. Hustad et al. (1992) correlated the experimental data by

$$q''_{\mathrm{max,conv+rad}} = 20q_f^{0.3} \quad (\mathrm{kW/m^2}) \tag{10}$$

based on a thermal input of $q_f = 60$–1600 kW. The location of the maximum heat flux was at the center of the flame. For pure fuel jets, there may have been significant quantities of unburned fuel at the target impingement point. Therefore, TCHR may have been important. It would have been included in the above correlation.

Flames impinging normal to a hemi-nosed cylinder

This pertains to cylinders whose axis is parallel to the flame, as shown in Figure 3. One end of the cylinder has a hemispherical nose. The flame impinges on that nose. Experimental conditions for these studies are given in Table 2.

Local convection heat transfer

Laminar and turbulent flows. Fairweather et al. (1984) correlated experimental data for laminar and turbulent flames. The correlations were modifications of empirical equations. These were derived for heat transfer from the stagnation point of heated spheres to turbulent air flows. Galloway and Sage (1968) recommended the following equations for nonreacting flows:

$$\mathrm{Nu}_{s,e} = 1.255\mathrm{Re}_{b,e}^{0.5}(\mathrm{Re}_{b,e}\mathrm{Tu})^{0.0214} \quad \text{for} \quad \mathrm{Re}_{b,e}\mathrm{Tu} < 7000 \tag{11a}$$

$$\mathrm{Nu}_{s,e} = 1.128\mathrm{Re}_{b,e}^{0.5}(\mathrm{Re}_{b,e}\mathrm{Tu})^{0.2838} \quad \text{for} \quad \mathrm{Re}_{b,e}\mathrm{Tu} > 7000 \tag{11b}$$

Fairweather used modifications of these equations to get

$$q''_{s,\mathrm{conv}} = 1.743(\mathrm{Re}_{b,e}\mathrm{Tu})^{0.0214}\frac{\rho_e v_e c_{p_e}}{\mathrm{Re}_{b,e}^{0.5}}(t_e - t_w) \tag{12}$$

Gostkowski and Costello (1970) recommended the following equation, also for nonreacting flows:

$$\mathrm{Nu}_{s,e} = 2 + \left\{1.849\left(\frac{v_e}{v_w}\right)^{0.16} + [0.2122\mathrm{Tu}(\mathrm{Tu} - 0.1072) + 0.001317]\mathrm{Re}_{b,e}^{1/2}\mathrm{Pr}_e^{1/6}\right\}\mathrm{Re}_{b,e}^{1/2}\mathrm{Pr}_e^{1/3} \tag{13}$$

Fairweather recommended the following modification of Equation 13, for laminar and turbulent flames

$$q''_{s,\mathrm{conv}} = \rho_e v_e c_{p_e}\left[\frac{2}{\mathrm{Pr}_e\mathrm{Re}_{b,e}} + \frac{1.1849}{\mathrm{Pr}_e^{2/3}\mathrm{Re}_{b,e}^{1/2}}\left(\frac{v_e}{v_w}\right)^{0.16} + [0.2122\mathrm{Tu}(\mathrm{Tu} - 0.1072) + 0.001317]\mathrm{Pr}_e^{-1/2}\right] \times (t_e - t_w) \tag{14}$$

Both Equations 12 and 14 tended to overpredict the experimental data.

Table 2 Experimental conditions for flames impinging normal to hemi-nosed cylinders

	Heat transfer mechanisms							
Location	Forced conv.	TCHR	Radiation	Oxidizer	Re_n	ϕ	Fuel	Reference
Local	√			air	laminar & turbulent[a]	0.943, 1.171	CH_4	Fairweather et al. 1984
	√			air	2000–12,000	0.8–1.2	CH_4	Hargrave et al. 1987
	√	√		O_2	not given	0.95	natural gas	Ivernel and Vernotte 1979

[a]According to author (only qualitative).

Turbulent flows. Hargrave et al. (1987) recommended the following empirical correlation:

$$q''_{s,\text{conv}} = \frac{\bar{\bar{k}}}{d_b}\left\{\overline{\overline{\text{Re}}}_b^{0.5}\left[0.993 + 5.465\left(\frac{\text{Tu}\overline{\overline{\text{Re}}}_b^{0.5}}{100}\right) - 2.375\left(\frac{\text{Tu}\overline{\overline{\text{Re}}}_b^{0.5}}{100}\right)^2\right]\right\}(t_e - t_w) \tag{15}$$

This underpredicted the data by up to 15%, near the reaction zone. Agreement was very good further downstream.

Local convection heat transfer with thermochemical heat release

Turbulent flows. Ivernel and Vernotte (1979) determined:

$$q''_{s,\text{conv+TCHR}} = \frac{k_w}{r_b}\left\{0.853\text{Re}_{r_b,w}\overline{\text{Pr}}^{0.4}\right\}\frac{h_e^T - h_w^T}{c_{p_w}} \tag{16}$$

This was valid for $1000 < \text{Re}_{r_b,w} < 45000$. Equation 16 correlated the experimental data within 10%. Although the flow conditions were not given, they are believed to have been turbulent. This is based on a comparison of the reported velocities with those of other studies (see Baukal and Gebhart 1995). Radiation to the target was calculated. It was specifically excluded from the above correlation.

Flames impinging normal to a plane surface

This geometry is shown in Figure 1. The targets have included both disks and rectangular plates. The experimental conditions are given in Table 3.

Local convection heat transfer

Laminar flows. In related studies, Hoogendoorn et al. (1978) and Popiel et al. (1980) determined two empirical correlations, for the heat transfer to the stagnation point. These were based on the axial distance between the target and the burner. The correlation for $2 \le L \le 5$ was:

$$q''_{s,\text{conv}} = \frac{k_{\text{ref}}}{d_n}\{(0.65 + 0.084\text{L})\text{Re}_{n,\text{ref}}^{0.5}\text{Pr}_{\text{ref}}^{0.4}\}\frac{h_e^S - h_w^S}{c_{p_{\text{ref}}}} \tag{17a}$$

The correlation for $L > 12$ was:

$$q''_{s,\text{conv}} = \frac{k_{\text{ref}}}{d_n}\{(1.37 - 1.8\text{L})\cdot 10^{-3}\text{Re}_{n,\text{ref}}^{0.75}\text{Pr}_{\text{ref}}^{0.4}\}\frac{h_e^S - h_w^S}{c_{p_{\text{ref}}}} \tag{17b}$$

Both correlations closely matched the data. For $5 < L \le 12$, the data showed a peak for Nu_n. No correlation was given. A modified form of a semi-analytic solution was also determined. It was a form of the equation recommended by Sibulkin (1952). Sibulkin's equation, along with many variations of it, is discussed by Baukal and Gebhart (1996). The modified correlation was:

$$q''_{s,\text{conv}} = \frac{k_{\text{ref}}}{d_n}\{2.37(\text{L}+1)^{-0.5}\text{Re}_{n,\text{ref}}^{0.5}\text{Pr}_{\text{ref}}^{0.4}\}\frac{h_e^S - h_w^S}{c_{p_{\text{ref}}}} \tag{17c}$$

valid for $8 \le L \le 20$. This underpredicted the data by as much as 60%. This was believed to have been caused by the failure to include the effects of free jet turbulence. Another correlation was recommended, which included those effects:

$$q''_{s,\text{conv}} = \frac{k_{\text{ref}}}{d_n}\left\{\left[\frac{2.37}{(\text{L}+1)^{0.5}} + 2.22\left(\frac{\text{TuRe}_{n,\text{ref}}^{0.5}}{100}\right) - 2.76\left(\frac{\text{TuRe}_{n,\text{ref}}^{0.5}}{100}\right)^2\right]\text{Re}_{n,\text{ref}}^{0.5}\text{Pr}_{\text{ref}}^{0.4}\right\}\frac{h_e^S - h_w^S}{c_{p_{\text{ref}}}} \tag{17d}$$

valid for $L \ge 8$. This compared favorably with the data. For $L \ge 4$, the maximum heat flux occurred at the stagnation point.

Table 3 Experimental conditions for flames impinging normal to plane surfaces

Location	Heat transfer mechanisms: Forced conv.	TCHR	Radiation	Oxidizer	Re_n	ϕ	Fuel(s)	Reference
Local	√			air	1050, 1860	1.0	natural gas	Hoogendoorn et al. 1978, Popiel et al. 1980
	√			air/O_2	857–2000	0.61	CH_4	Kataoka et al. 1984
	√			air	1500–19,000	0.67–0.95	town gas+	Shorin & Pechurkin 1968
	√	?	?	air	17,680–22,700	2.75	CH_4	Buhr et al. 1973, Kremet et al. 1974
	√			air	turbulent[a]	1.0	natural gas	Smith & Lowes 1974
	√			air	turbulent[a]	not given	coke oven gas	Matsuo et al. 1978
	√			air	turbulent[a]	0.95	natural gas	Vizioz & Lowes 1971
	√			air/O_2				
	√			air	laminar[a]	1.33, 1.67	CH_4	Anderson & Stresino 1963
	√	√		O_2		1.0	H_2	
	√	√		O_2		1.0, 1.43	C_3H_8	
	√	√	?	O_2		2.34	C_2H_2	
	√	√		O_2	300–1200	0.83–1.0	$C_3H_8+C_4H_{10}$	Shorin & Pechurkin 1968
	√	√		O_2	turbulent[a]	1.0	C_3H_8	Rauenzahn 1986
Average	√			air	70, 356	∞	natural gas	You 1985
	√			air	1500–19,000	0.67–0.95	town gas+	Shorin & Pechurkin 1968

[a]Calculated from data in reference.

For $L = 2$ and 3, the maximum flux occurred at about $R = 0.5$ and 0.2, respectively. Kataoka et al. (1984) studied an $air/O_2/CH_4$ ($\Omega = 0.39$) flame. The heat transfer to the stagnation point was correlated by:

$$q''_{s,\mathrm{conv}} = \frac{k_e}{2r_{1/2v}}\left\{1.44\mathrm{Re}^{0.5}_{2r_{1/2v},e}\mathrm{Pr}_e^{0.5}(L - X_v)^{0.12}\right\}(t_e - t_w) \quad (18a)$$

$r_{1/2v}$ is the radius from burner axis, near the stagnation point, where the velocity is 1/2 of the velocity along the axis of symmetry. This correlation is valid only for $L > X_v$ with X_v correlated by:

$$X_v = 2.82(\rho_n/\rho_\infty)^{0.29}\mathrm{Re}^{0.07}_{n,X=0} \quad (18b)$$

In a subsequent study, Kataoka (1985) found the maximum heat flux for the same flame occurred at $L = X_v$.

Turbulent flows. Shorin and Pechurkin (1968) empirically determined

$$q''_{s,\mathrm{conv}} = \frac{k_e}{d_n}\{4.04\mathrm{Re}^{0.2}_{n,e}\mathrm{Pr}_e\}(t_e - t_w) \quad (19a)$$

This was valid for $L \le X_v$. The correlation matched the experimental data within 15%. Another correlation was given for the heat flux for $L > X_v$

$$\frac{\mathrm{Nu}_{s,e}(X_v < L < 14)}{\mathrm{Nu}_{s,e}(L < X_v)} = 0.8e^{-0.36\frac{(L-X_v)^2}{L}} \quad (19b)$$

The local heat transfer, as a function of R, was correlated by:

$$q''_{r,\mathrm{conv}} = \frac{k_e}{Rd_n}\left\{3.22\mathrm{Re}^{0.4}_{n,e}\mathrm{Pr}_e e^{-0.36\frac{(L-X_v)^2}{L} - 3.6\frac{R}{L}}\right\}(t_e - t_w) \quad (19c)$$

for $0 < R/L < 0.9$ and $X_r \le L \le 14$. Vizioz and Lowes (1971) investigated industrial-scale, air and oxygen-enriched air ($\Omega = 0.30$) natural gas flames impinging on cooled flat plate targets, located inside a furnace. Three different types of flames, with various levels of swirl, were tested. The data were correlated by:

$$q''_{r,\mathrm{conv}} = (k_e/r)\{\mathrm{Nu}_{r,e}\}(t_e - t_w) \quad (20a)$$

In one set of tests, the surface temperature of a water-cooled steel flat plate was maintained below 373K. The convective heat flux, as a function of radius from the stagnation point, was:

$$0.07\mathrm{Re}^{0.8}_{r,e} < \mathrm{Nu}_{r,e} < 0.22\mathrm{Re}_{r,e} \quad (20b)$$

In another set of tests, the surface temperature of an air-cooled refractory plate was maintained between 1280 and 1420K. The convection correlation was:

$$0.035\mathrm{Re}^{0.8}_{r,e} < \mathrm{Nu}_{r,e} < 0.125\mathrm{Re}_{r,e} \quad (20c)$$

The Prandt number effect was included in the coefficients. In related studies, Buhr et al. (1973) and Kremer et al. (1974) determined the following modification of Sibulkin's equation:

$$q''_{s,\mathrm{conv}} = 0.0371\mathrm{Pr}_e^{-0.6}\rho_e v_e(h^S_e - h^S_w) \quad (21)$$

This was valid for $17680 \le \mathrm{Re}_n \le 22700, 13 \le L \le 65$. The experimental data were correlated within 9%. The peak flux was measured near the reaction zone. The heat flux showed a small linear decline with probe surface temperature, over the range 320–1000K. Smith and Lowes (1974) determined the following empirical correlation:

$$q''_{s,\mathrm{conv}} = (k_e/l_c)\{1.6\mathrm{Re}^{0.54}_{l_c,e}\mathrm{Pr}_e^{0.52}\}(t_e - t_w) \quad (22)$$

where l_c was the width of the cooling channels. The correlation was valid for $0 \le L \le 8.4$. The fluid properties were taken as for air, at the gas temperatures measured near the plate. The axial velocity at 10 cm from the plate was used to calculate the velocity in Re_n. Matsuo et al. (1978) determined the following correlation, for the heat flux to the stagnation point:

$$q''_{s,\mathrm{conv}} = 0.010353(q_f^{0.936}/L^{1.032})(t_e - t_w) \quad (23a)$$

Another correlation was given for the heat flux, as a function of the distance r from the stagnation point:

$$q''_{r,\mathrm{conv}} = \left(q_f^{0.939}\Big/ K_1K_2L^{\left(\frac{6.318}{r^{0.414}}\right)}\right)(t_e - t_w) \quad (23b)$$

where

$$K_1 = -1.903\cdot 10^{-7}r^2 + 3.89\cdot 10^{-3}r + 0.985 \quad (23c)$$

$$K_2 = 8.42\cdot 10^{-7}r^3 - 1.803\cdot 10^3 r^2 + 1.37r + 5.00 \quad (23d)$$

Equations 23a, b were valid for $5 \le L \le 9$ and $148 \le q_f \le 482$. Equation 23b was valid for $0 < r \le 500$, where r was in mm.

Local convection heat transfer with thermochemical heat release

Laminar flows. Anderson and Stresino (1963) found that the heat flux decreased exponentially as R increased:

$$q''_{r,\mathrm{conv+TCHR}} = q''_{s,\mathrm{conv+TCHR}}e^{-\frac{4r}{d_j}\overline{\mathrm{Re}_j}^{-0.34}} \quad (24)$$

where $q''_{s,\mathrm{conv+TCHR}}$ was measured, and d_j was the estimated flame diameter prior to impingement. Shorin and Pechurkin (1968) found for $C_3H_8 + C_4H_{10}$ flames that:

$$q''_{s,\mathrm{conv+TCHR}} = \frac{k_e}{d_n}\{7.8\mathrm{Re}^{0.4}_{n,e}\mathrm{Pr}_e\}\frac{h^T_e - h^T_w}{\overline{c_p}} \quad (25)$$

for $L \le X_v$. This correlated the data within 10%.

Turbulent flows. Rauenzahn (1986) calculated the heat flux at the surface, from the transient temperature field in a copper block. The assumed form of the surface flux distribution, after Anderson and Stresino (1963), was:

$$q''_{r,\mathrm{conv+TCHR}} = q''_{s,\mathrm{conv+TCHR}}e^{-\left(\frac{r}{r_0}\right)^a} \quad (26)$$

r_0 was termed the spreading radius for the heat flux distribution. From the reduction of the data, $a \cong 1$. A transient inverse conduction analysis (Gebhart 1993) was used to calculate $q''_{s,\mathrm{conv+TCHR}}$ and $q''_{r,\mathrm{conv+TCHR}}$. For $q_f = 61$ and 94 kW and $L = 6$ and 8, $q''_{s,\mathrm{conv+TCHR}}$ ranged from 645–1210 kW/m^2, and r_0 ranged from 7.3 to 9.8 cm. The spreading radius generally decreased as L decreased and as q_f increased. Unfortunately, no correlations were given for $q''_{s,\mathrm{conv+THR}}$ and r_0 in terms of L and q_f. Therefore, the constants used in the above equation are expected to be dependent on the specific experimental conditions.

Average convection heat transfer

Laminar flows. You (1985) studied pure fuel jet flames impinging on a plate. The fuel flow from the burner nozzle was laminar. However, the buoyant plume impinging on the plate was turbulent. It was found that the convective heat flux in the stagnation zone was essentially constant:

$$q''_{b,\mathrm{conv}} = 31.2(q_f/l_j^2)\mathrm{Ra}_e^{-1/6}\mathrm{Pr}_e^{-3.5} \tag{27a}$$

for $R < 0.16$. The flux decreased with R in the wall jet region:

$$q''_{b,\mathrm{conv}} = 1.46R^{-1.63}(q_f/l_j^2)\mathrm{Ra}_e^{-1/6}\mathrm{Pr}_e^{-3/5} \tag{27b}$$

for $R > 0.16$. In both cases, $10^9 < \mathrm{Ra} < 10^{14}$ and $\mathrm{Pr} \cong 0.7$. The Rayleigh number was defined as

$$\mathrm{Ra} = g\beta_e q_f l_j^2/\rho_e c_{p_e} v_e^3 \tag{27c}$$

No correlations were given for the measured radiation heat flux. This accounted for up to 26% of the total heat flux. The radiant flux was specifically excluded from the above correlations.

Turbulent flows. Shorin and Pechurkin (1968) determined the average heat transfer as a function of R:

$$q''_{b,\mathrm{conv}} = \frac{k_e}{r_b}\left\{6.44\mathrm{Re}_{n,e}^{0.4}\mathrm{Pr}_e e^{0.36\frac{(L-X_v)^2}{L}} \times r_b^2\left[0.08l_j^2 - 0.08l_j^2 e^{-3.6\frac{R}{L}} - 0.28r_b l_j e^{-3.6\frac{R}{L}}\right]\right\} \times (t_e - t_w) \tag{28}$$

Flames parallel to a plane surface

This configuration is useful for studying the heat transfer to high-speed airfoils, where very high temperatures occur at the leading edge. It is also useful for studying the heat transfer from flames to the walls of a furnace. The experimental conditions for these studies are given in Table 4.

Local convection heat transfer with thermochemical heat release

Laminar flows. Woodruff and Giedt (1966) studied the configuration shown in Figure 4a. This simulated an air foil. The best fit of the experimental data was obtained using

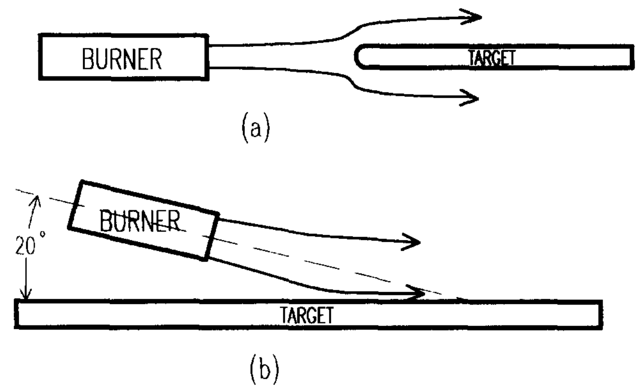

Figure 4 Flames impinging along plane surfaces

$$q''_{r,\mathrm{conv+TCHR}} = \frac{2\mu_e}{\delta}\mathrm{Pr}_e^{-2/3}\left[1 + (\mathrm{Le}_{e,\mathrm{H}} - 1)\frac{h_{e,\mathrm{H}}^C - h_{w,\mathrm{H}}^C}{h_e^T - h_w^T}\right]^{2/3} \times (h_e^T - h_w^T) \tag{29a}$$

The boundary-layer thickness δ was measured with a Pitot tube. An empirical curve fit of that data was given as

$$\delta = 0.0139e^{-0.312r} + 0.152((e^{0.703r} - 1)/e^{0.625r})^{0.5} \tag{29b}$$

r is the distance along the plate, measured in inches, from the leading edge. The best fit of the heat flux data was for $\mathrm{Le} = 3.0$.

Turbulent flows. Giedt et al. (1960) tested an array of fuel-rich C_2H_2 flames in parallel flow over a flat plate, as shown in Figure 4a. The heat flux data were correlated, using a form of the equation recommended by Eckert (1956)

$$q''_{r,\mathrm{conv+TCHR}} = (\bar{k}/r)\{0.0296\overline{\mathrm{Re}}_r^{0.8}\overline{\mathrm{Pr}}^{1/3}\}(t_e - t_w) \tag{30a}$$

r is the distance from the leading edge. This equation underpredicted the experimental data by 25%. A derived correction factor was then added, for the H atom recombination reaction:

$$q''_{r,\mathrm{conv+TCHR}} = \frac{\bar{k}}{r}\left\{0.0296\overline{\mathrm{Re}}_r^{0.8}\overline{\mathrm{Pr}}^{1/3} \times \left[1 + \left\{\frac{Q_{\mathrm{H}}(m_{\mathrm{H},e} - m_{\mathrm{H},w})}{M\bar{\rho}\bar{c}_p(t_e - t_w)}\right\}\right]\right\}(t_e - t_w) \tag{30b}$$

where

$$M = (m_{\mathrm{H},w} - m_{\mathrm{H},e})\Big/\left(\ln\frac{1 - m_{\mathrm{H},w}}{1 - m_{\mathrm{H},e}}\right) \tag{30c}$$

$Q_{\mathrm{H}} = 436$ kJ/kg-mole

Table 4 Experimental conditions for flames impinging parallel to plane surfaces

	Heat transfer mechanisms							
Location	Forced conv.	TCHR	Radiation	Oxidizer	Re_n	ϕ	Fuel	Reference
Local	√	√	?	O_2	laminar[a]	2.5	C_2H_2	Woodruff & Giedt 1966
	√	√	?	O_2	turbulent[a]	2.5	C_2H_2	Giedt et al. 1960
	√		√	air	4800–270,000	1.0	coke oven gas	Beér & Chigier 1968

[a]According to author (only qualitative).

Q_H is the H atom heat of recombination, and m_H is the H atom mole fraction. This correlated the data within 15%.

Local convection and radiation heat transfer

Turbulent flows. Beér and Chigier (1968) studied flames impinging at a 20° angle from the furnace hearth (see Figure 4b). The correlation for the total heat flux was given as

$$q''_{r,\mathrm{conv+rad}} = (k_{\max}/r)\{0.13\mathrm{Re}_{r,\max}^{0.8}\}(t_{\max} - t_w) \tag{31}$$

The velocity in the Reynolds number was the maximum velocity measured at the axial distance from the point of impingement. The temperature $t_{\max}$ was the maximum temperature measured at the axial distance from the point of impingement. A Prandtl number of 0.7 was assumed.

Recommendations

In some of the studies considered here, multiple heat transfer mechanisms have been important. For example, radiation and forced-convection heat transfer were both important in the study by Beér and Chigier (1968). Very little work has been done to try to separate these mechanisms and to determine their relative importance. This should be investigated, because it may help in optimizing heating processes. For instance, if radiation is important for a given set of conditions, it may be beneficial to optimize the surface radiation properties of the target. Some heat transfer mechanisms, such as natural convection and condensation, have received little or no attention. Again, these may be important under certain operating conditions. For example, condensation may be a dominant mechanism when the target surface temperature is below the condensation point of water vapor.

There are many combinations of geometry and operating conditions for which no empirical heat transfer correlations exist. The intent here is not to give a detailed list of all the possible experiments that could be done. Rather, general types of experiments are suggested.

Flames impinging normal to a cylinder. For these flames, only turbulent forced convection at the stagnation point has been studied. No correlations have been determined for the local convection heat transfer with TCHR. Only one correlation, Equation 8, has been given for the average convection heat transfer with TCHR. Unfortunately, the flow type was not specified. The radiation correlations, Equations 9a, b and 10, determined by Hustad and co-workers (1991) and Hustad et al. (1992), are specific to the given operating conditions. These included very-high-speed flows. The equations were not nondimensionalized. They should not be used outside the known experimental conditions.

Flames impinging normal to a hemi-nosed cylinder. Only the heat transfer at the stagnation point has been studied. The heat flux at other locations has not been reported. No averaged heat flux correlations have been given. No radiation or laminar forced-convection correlations of any type have been reported. However, many semianalytic heat transfer equations have been used for this geometry (Baukal and Gebhart 1996).

Flames impinging normal to a plane surface. This has been the most widely studied geometry. No correlations have been reported for local turbulent convection heat transfer with TCHR. The only correlations for average heat transfer have been for forced convection. No correlations of any kind have been specifically determined for radiation.

Flames parallel to a plane surface. This has been the least studied geometry. No pure forced-convection correlations have been reported. No average correlations of any kind are given. Only radiation, in combination with forced convection, has been studied, for a limited set of conditions.

Many other types of geometries might be considered. One example would be flames impinging normal to rectangular bars. This shape simulates a typical industrial application, where metal ingots are reheated prior to entering the rolling mill. There are also many combinations of fuels, oxidizers, and equivalence ratios that remain to be correlated. For example, very little work has been done for oxidizers other than either air or pure oxygen. Only a limited number of studies have investigated the effects of a furnace enclosure. Flame impingement heating represents an important technology that requires further experimental investigation.

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