HEAT TRANSFER IN FLAMES

N. H. Afgan and J. M. Beer, Editors

ADVANCES IN THERMAL ENGINEERING

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FOREWORD

The main theme of this volume is heat and mass transfer from flames and its application to engineering systems.

The practical application of the fundamentals of heat transfer in flames covers a wide range of important engineering systems, which include industrial furnaces, boilers, gas turbine combustors, internal combustion engines and open flames such as flares and fires. In recent years there has been a growing interest in the study of thermal radiation for industrial application. This is due to increasing combustion intensities in industrial applications and the need for prediction methods for detailed distribution of radiant heat flux to bounding surfaces.

A definite need exists for more detailed information on engineering design for heat transfer from flames to heat sinks and also on heat transfer within the flame itself, as the latter affects combustion intensity and the rate at which certain types of flames and fires propagate.

The material presented in this book contains three different aspects of heat transfer from the flame. The first part includes papers which deal with problems of heat transfer in open flames. Special attention is given to the mechanism of fire spread in forests and calculation methods for predicting radiative heat transfer in a diffusion flame. This section also includes a probability analysis of fabric ignition and the burn injury hazard.

The largest part of this volume is devoted to problems of heat transfer in steady confined flames. This group of papers is divided according to subject-matter. It covers radiative properties, measurement techniques, prediction methods (zone and flux methods) and the effect of turbulent fluctuation on radiant heat transfer. The analytical and experimental method for determination of radiative properties of the flame are presented. Special consideration is given to the effect of pressure on the mean absorption coefficient of luminous flames of liquid fuel spray combustion. The theoretical and experimental results of radiative properties in a dispersed system are also offered.

Although there are not many papers dealing with measurement techniques, which are mainly covered under particular topics, the papers presented in this group treat new techniques for radiative heat transfer and their employment in the measurement of scaling deposits in large thermal power stations.

Development of the prediction method for calculation of heat transfer from confined flames is of utmost importance. We are privileged to have Professor H. C. Hottel, the pioneer in radiative heat transfer, to lead this volume with his chapter "First Estimate of Industrial Furnace Performance–One Zone Model Reexamined." Recent advances in methods for predicting radiative heat flux distribution in furnaces and combustors have focused attention on comparison of the zone method and the flux method of analysis. Most of the chapters presented in this part of the volume are directly or indirectly connected with analyzing the advantages of particular methods of predicting radiant heat flux distribution. Those methods have been applied in different industrial systems, and the results of the heat transfer and flow distribution are given for a gas turbine combustion chamber, an industrial water-tube boiler, pulverized fuel furnaces and other types of furnaces.

The third part is devoted to heat transfer in unsteady confined flames. Most of the chapters in this group deal with heat transfer from flames in internal combustion engines. This section reviews the available information of heat transfer from the working fluid to the wall of internal combusion engines. Some chapters deal with the experimental and theoretical method for determination and prediction of radiant heat transfer in internal combustion engines and its correlation with other heat transfer mechanisms.

The scientific organization of the program of the 1973 International Seminar was the responsibility of the Seminar Committee, as follows: W. J. D. Annand, University of Manchester, U.K.; A. Blokh, Central Boiler and Turbine Institute, U.S.S.R.; J. M. Riviera, I.R.S.I.D., France; L. Kreuh, University of Zagreb, Yugoslavia; A. F. Sarofim, Massachusetts Institute of Technology, U.S.A.; P. H. Thomas, Fire Research Station, U.K.; and Chairman J. M. Beer, University of Sheffield, U.K. The Seminar was attended by 112 participants from 20 countries.

Organization of the lectures was financially supported by the following organizations: UNESCO, Paris; Council for Scientific Coordination of the Socialist Republic of Croatia, Zagreb, Yugoslavia; Djuro Djaković enterprise for design and construction of power and industrial plants, Slavonski Brod, Yugoslavia; National Science Foundation, Washington, D.C., U.S.A.; Academy of Sciences of the U.S.S.R., Moscow, U.S.S.R.; Boris Kidric Institute, Belgrade, Yugoslavia.

Acknowledgement should be given to UNESCO for special grants to participants from developing countries. The editors would also like to express their appreciation to the authors for their most helpful cooperation in preparing this volume.

N. H. Afgan

J. M. Beer

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METHOD OF CALCULATION

Chapter 1

FIRST ESTIMATES OF INDUSTRIAL FURNACE PERFORMANCE — THE ONE-GAS-ZONE MODEL REEXAMINED

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Abstract

The one-gas-zone model of industrial furnace performance is set up, with allowance for wall losses, sink temperature variation, and departure from perfect stirring. Judicious choice of the mode of allowance for the last of these keeps the final relation simple enough to visualize. It is shown that although the model equations are in a form for prediction of furnace performance, they may instead be used in a graphical technique for correlating furnace data and for determining two constants in a semi-empirical relation. A solution of the problems of determining the effective sink area A_S, the mean beam length for gas radiation, and the effective sink emissivity is presented, for tubular heaters with wall-mounted tubes as well as for tubes internally mounted in planes with radiating gas on either side.

NOMENCLATURE

A_O	area of furnace openings losing radiation
$A_r(A_p)$	area of refractory (of tube plane)
A_S	effective sink area (old $(\overline{GS})_{R,c}$)
A_1	area of stock or sink in furnace chamber. If tubes on wall, plane of tubes.
$A_{1,e}$	that part of A_1 exclusive of any tube curtain across the gas-exit passage.
a_g	energy fraction of black-body spectrum occupied by gray gas in a gray plus clear mixture
B	ratio of center-to-center tube spacing to diameter
$\underline{C}$	"cold" fraction of furnace enclosure area, $A_1/(A_1+A_r)$
$c_{p,g}$	specific heat of combustion gases, mean value from gas-chamber exit to base temperature
D'	reduced firing density, $\dot{H}_F/\sigma A_S T_F^4(1-T_O')$
d	dimensionless constant in relation $\Delta' = (1-1/d)Q'$
$\mathcal{E}_3$	third exponential integral, $\int_1^\infty (e^{-xt}/t^3)dt$
$\underline{E}$	black emissive power, σT^4
$\overline{F}$	factor for radiation loss through walls, based on inside and outside temperatures and area of opening
$\overline{F}_{iso(gas)}$	fraction of isotropic (gas) radiation intercepted by tube row, directly plus by interception of returning beam from background
$\underline{F_{xy}}$	fraction of radiation from surface x which is intercepted by surface y
$\overline{g_x s_y}$	direct-interchange area between gas x and surface y
$\overline{G_x S_y}$	total-interchange area, ratio of net radiative flux between gas zone x and surface zone y, allowing for reflections at all surfaces, to the difference in black emissive powers (σT^4) of x and y. Sometimes, allowance made for refractory aid without appending sub-R.
$(\overline{GS})_R$	total-interchange area between gas and surface zones, with aid given by equilibrium refractory surfaces included.
$H(T_1)$	enthalpy of stream 1 (sink stream) dependent on temperature
$H_{1,i(o)}$	enthalpy of stream 1 at inlet (outlet)
$\dot{H}_F$	enthalpy rate of any entering streams affecting firing rate, including fuel, air, and recirculated flue gas if any, above dead state of completely burned gaseous products at T_O.

h_i convection heat-transfer coefficient on inside surface of refractory walls

$h_{c+r,o}$ heat transfer coefficient by convection plus radiation, on outside walls of refractory surfaces

K absorption coefficient, l^{-1}

k thermal conductivity of refractory walls

L dimension of parallelepiped

L_m mean beam length for gas radiation

L_O' dimensionless loss coefficient for radiative flux through furnace openings, $\overline{F}A_O/A_S$

L_r' dimensionless loss coefficient for heat loss through refractory walls, $U_rA_r/\sigma A_S T_F^3$

m_g mass flow rate of combustion gases

n number of tubes in a tube row

P pitch of tubes in row, center-to-center distance

p partial pressure of gas-radiating components, atm.

$\dot{Q}_G$ rate of heat transfer from combustion gases

Q' reduced rate of heat transfer from combustion gases, $\dot{Q}_G/\dot{H}_F)(1-T_O')$

r ratio of temperature rise of stock surface to the sum of inlet and outlet temperatures, $(T_{1,o} - T_{1,i})/(T_{1,o} + T_{1,i})$.

S speckledness; 1 for surface with A_1 and A_r intimately mixed

$\overline{s_x s_y}$ direct-interchange area for radiative exchange between surfaces x and y no gas absorption included

$\overline{s_x s_y}'$ same as above, except that gas absorption <u>is</u> included

$\overline{xy}(\overline{xy}')$ shorthand for $\overline{s_x s_y}$ ($\overline{s_x s_y}'$)

T_F adiabatic pseudo-flame temperature, based on Eq. following (2)

T_g mean radiating temperature of combustion gases

T_1 mean temperature of stock or sink surface

T_O base temperature (also ambient)

U_r overall coefficient of heat transfer from combustion gases through refractory wall to ambient

W refractory wall thickness. Also thickness of gas slab, Figs. 7-9

$\overline{1r}$ shorthand for $\overline{s_1 s_r}$, $\equiv A_1F_{1r} \equiv A_rF_{r1}$

α gas absorptivity

Δ gas-radiating temperature minus gas temperature leaving combustion chamber

$\varepsilon_g(\varepsilon_1)$ gas emissivity (effective emissivity (emittance, absorptance) of surface A_1)

ε_1' true emissivity of tube surface

η furnace efficiency, (flux to stock or sink)/$\dot{H}_F$

σ Stefan-Boltzmann constant

τ gas transmissivity ($\equiv$1-absorptivity, $=1-\varepsilon_g$ if gas gray)

ψ angle. See Fig. 7

<u>Subscripts</u>

i,o inlet, outlet

r refractory

1 sink surface, or stock surface

T total, applied to area

<u>Primes</u>

On T_g, T_1, T_O designate the ratio of those temperatures to T_F.

Introduction

The design of a furnace, more specifically the prediction of the performance of a chosen design, can be carried out at several levels of sophistication. Although a determination of the distribution of heat-flux density over the surface of the stock is desirable — sometimes, in high flux-density systems, necessary — the attainment of the simpler objective of determining the total heat transfer rate as a function of firing rate and excess air is a proper orienting first step; and often it suffices. Even if that is the sole objective, knowledge of the detailed interaction of radiation and convection with mass transfer and combustion is in principle necessary. But integral formulations are tolerant of casual treatment of detail, especially in the presence of the leveling effect of radiation, responsive to a high power of temperature; and a surprisingly accurate overall performance is predictable from a relatively simple model. Even though the knowledge of flux distribution over the stock may be the ultimate objective, it is still good engineering practice to start with an almost-quantitative understanding of the overall process. In fact, it may be asserted that prospects of success with the zone method are poor if the simpler and less ambitious approach, which is after all a one-gas-zone example of the zone method, is not thoroughly understood.

It is the object here to set up a simple overall furnace performance model, in form as general as is consistent with the assumption of a single gas-radiating temperature, a single equilibrium refractory temperature and a single term characterizing the exchange area between the combustion gases and the sink, which allows for

1. The effect of the adiabatic flame temperature T_F, dependent on the entering fuel and air enthalpies and the gas heat capacity
2. The effect of stock or sink temperature T_1, measured by the ratio of its mean value to T_F
3. The effect of stock temperature variation, measured by the ratio r of stock temperature rise to its arithmetic mean temperature.
4. The value of the characteristic or effective sink area A_S, that area which, multiplied by the difference of black-body emissive powers of the gas and stock temperatures, gives the flux from gas to stock. The major problem of making the model describe realistically the effects of gas composition, furnace shape, and disposition of heat sinks in the furnace comes in the evaluation of A_S, the total gas-sink exchange area

5. The use of a single gas-radiating temperature T_g, but
6. a difference Δ between the gas-radiating and leaving-gas-enthalpy temperature, which varies with firing rate
7. The loss of heat through the refractory walls
8. The loss of heat by radiation through openings
9. Other factors, contributing to the evaluation of A_S.

The One-Gas-Zone Furnace Model

Although parts of the following development have appeared before(1,2a), parts are new; and for completeness the full derivation will be presented. The well stirred furnace gases are at temperature T_g in consequence of loss of heat (a) by radiative exchange with the stock or sink at T_1, (b) by convection to that part of the sink $A_{1,e}$ which does not include any curtain tubes across the gas exit from the chamber, (c) by convection to and through the refractory walls, and (d) by radiation through furnace openings of area A_O.

a) The net radiative flux to the sink — direct as well as with the aid of refractory surfaces which reflect diffusely or absorb and reradiate — must, if the refractory is radiatively adiabatic and the gas is gray, be proportional to the difference in black-body emissive power of the gas and sink. The proportionality constant, having the dimensions of area, is called the gas-surface total-exchange area $(\overline{GS})_R$ the formulation of which will be discussed later. (The subscript indicates that allowance has been made for the aid given by refractory surfaces). The flux is then $(\overline{GS})_R\ \sigma(T_g^4-T_1^4)$.

b) Convective flux to those surfaces $A_{1,e}$ which affect the stirred-gas enthalpy is $hA_{1,e}(T_g-T_1)$. Because this term is quite small compared to (a), it is convenient to combine the two by forcing the convection into a fourth-power form: $hA_{1,e}(T_g-T_1) \sim hA_{1,e}\sigma(T_g{}^4-T_1{}^4)/4\sigma T_{g1}^3$, where T_{g1} is the arithmetic mean of T_g and T_1. Then

$$(\overline{GS})_R\ \sigma(T_g{}^4-T_1{}^4)+hA_{1,e}(T_g-T_1) = [(\overline{GS})_R + hA_{1,e}/4\sigma T_{g1}]\ \sigma(T_g{}^4-T_1{}^4)$$

The bracket has in other contributions been called $(\overline{GS})_{R,c}$ to indicate that convection has been included. The simpler term A_S, the <u>effective</u> area of the sink, will be used here.

c) Convection to and through refractory walls. If the walls are in radiative equilibrium, convection — gas to wall — equals conduction through the wall. The flux is $U_rA_r(T_g-T_o)$, where T_o is the ambient temperature and U_R is given, conventionally, by $U_r = \dfrac{1}{\dfrac{1}{h_i} + \dfrac{W}{k} + \dfrac{1}{h_{c+r,o}}}$

The inside flux is not in fact equal to the flux through the wall, but the difference is so small compared to the radiative flux as hardly to negate the assumption of radiative equilibrium. Without that assumption one would need to introduce an additional unknown and an additional equation, and the slight improvement in final accuracy does not justify the complication.

d) Radiation through peep holes or other openings, of area A_O. Rigorous allowance for this usually small effect would introduce such complexities as to prevent obtaining a solution capable of easy engineering use. Although the view from the outside through furnace openings is a view of sink and refractory surfaces seen dimly through partly diathermanous gas, the assumption will be made that the effective furnace temperature (the inside plane of the openings) is T_g. With $\overline{F}$ representing the exchange factor to allow for wall thickness ([3]), the loss through the openings becomes $A_O\overline{F}\sigma(T_g{}^4-T_O^4)$. Furnaces with openings large enough to make this casual treatment inadequate are rare.

The equation of transfer from the gas is, from the above,

$$\dot{Q}_G = A_S\sigma(T_g{}^4-T_1{}^4) + U_rA_r(T_g-T_o) + \overline{F}A_o\sigma(T_g{}^4-T_o{}^4) \qquad (1)$$

An energy balance on the gas is needed. Although a single gas radiating temperature has been postulated, it can be a space-mean value rather than the uniform gas temperature of a perfectly stirred chamber; and the gas temperature measuring the gas enthalpy leaving the chamber is usually lower. Let $T_g-\Delta$ represent the leaving-gas temperature, between which and the base temperature T_o the mean heat capacity is $\overline{C}_{p,g}$. Then the energy balance is

$$\dot{Q}_G = \dot{H}_F - (T_g-\Delta-T_o)\dot{m}_g\overline{C}_{p,g} \qquad (2)$$

where $\dot{H}_F$ is the hourly entering enthalpy, chemical plus sensible, in the fuel and air and recirculated flue gas, if any, T_o is the enthalpy-base temperature, and $\dot{m}_g$ is the mass flow rate of gas/hour. Let the same mean heat capacity be used to define an adiabatic pseudo-flame temperature T_F

$$(T_F-T_o) = \dot{H}_F/\dot{m}_g\overline{C}_{p,g}$$

(T_F will in general be much higher than the true adiabatic flame temperature which allows for a temperature-varying C_p and for dissociation). The energy balance may