INCINERATOR OVERFIRE MIXING STUDY

a report to

CONTROL SYSTEMS DIVISION
OFFICE OF AIR PROGRAMS
ENVIRONMENTAL PROTECTION AGENCY

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Arthur D. Little, Inc.

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ABSTRACT

Incineration, as an increasingly important tool in municipal solid waste management, can be a significant source of air pollution in urban areas. Combustible air pollutants (carbon monoxide, soot or char, and hydrocarbons) exist in the gases leaving incinerator furnaces as a consequence of poor combustion. Since these pollutants are removed from the exit gases with difficulty, improving the combustion environment is an attractive approach for improved pollutant emission control.

The processes occurring in a burning refuse bed are analyzed to yield estimates for the rate and quantity of combustibles emitted from the bed. It is shown that gases of over 14% carbon monoxide content can be released. Methods are then developed to analyze the flow through the furnace. The analysis shows that stratification of the flow and incomplete mixing of fuel and sufficient oxygen-containing gases can occur. Design equations are presented describing the behavior of overfire air or steam jets for mixing, tempering and/or bringing oxygen to the combustible gases. Refuse incinerator overfire air system design methods developed from the above analyses are presented and contrasted with similar design methods for solid fuel bed coal-firing furnaces.

ACKNOWLEDGEMENT

This effort was supported by the Control Systems Division, Office of Air Programs of the Environmental Protection Agency under Contract EHSD 71-6. We wish to express our appreciation to Mr. R. C. Lorentz of CSD for his many contributions and suggestions. Our thanks also go to Mr. Willard S. Pratt and others of the City of Newton, Massachusetts for their cooperation and assistance in developing a municipal incinerator overfire mixing test plan.

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A	Duct cross-sectional area (ft^2)
a	Moles CO generated in bed per mole of oxygen (Equation II-1)
$a_{1,2...}$	Constants (dimensionless)
B	Buoyant force on jet in crossflow (lb_f)
B_1	Constant (ft^2/sec^2)
b	Half-width of shear layer (Appendix C) (ft)
C_n	Effective drag coefficient of crossflow on jet (dimensionless)
c	Rate of spread of jet width in crossflow (dimensionless)
$\bar{c}$	Time averaged concentration of jet fluid along radius (lb/lb of mixture)
$\bar{c}_m$	Time averaged jet centerline concentration of jet fluid (lb/lb of mixture)
$\bar{c}_o$	Time averaged concentration of jet fluid at nozzle (lb/lb of mixture)
cp	Heat capacity of ambient fluid (Btu/lb °F)
D	Drag force on jet in crossflow (lb_f)
d	Moles H_2O generated in bed per mole of oxygen (Equation II-1)
d_o	Nozzle diameter (ft)
E	Constant (Equation III-5) (dimensionless)
e	Moles C generated in bed per mole of oxygen (Equation II-1)
F	Force ratio on buoyant jet in crossflow (dimensionless)
F_{xB}	Buoyant force in x direction (lb_f)
F_{xD}	Drag force in x direction (lb_f)
f	Wall friction factor (dimensionless)
G	Solutions to characteristic equation (Equation B-8)
g	Acceleration due to gravity (32.2 ft/sec^2)

LIST OF SYMBOLS (Cont.)

H	Total head (feet water gauge)
H_c	Heat of combustion of furnace gases (Btu/lb oxygen consumed)
h	Jet width in crossflow (feet)
h_{fg}	Latent heat of vaporization (Appendix D) (Btu/lb m)
i	Moles H_2 generated in bed per mole of oxygen (Equation II-1)
j	Moles $C(H_2O)_n$ gasified per mole of oxygen (Equation II-1)
K	Wavenumber of disturbance of interface (Appendix B) (ft^{-1})
k	Coefficient - Equation IV-18 (dimensionless)
k_c	$CO-CO_2$ equlibrium constant ($atm.^{-1}$)
k_w	Water gas shift equilibrium constant (dimensionless)
L	Length of physical system (feet)
L_j	The jet penetration distance - Equation IV-18 (feet)
ℓ	Path length of axis of curved jet (feet)
M	Momentum flux ratio of external to jet flow (dimensionless)
$\dot{M}_x$	Rate of change of jet momentum in x direction (lb_f)
m	Constant
$\dot{m}_o$	Jet mass flow at nozzle (lb_m/sec)
$\dot{m}_x$	Jet mass flow at distance x (lb_m/sec)
N	Number of jets in row - Equation IV-23 (dimensionless)
N_R	Reynolds number (dimensionless)
N_{Fr}	Froude number (dimensionless)
N_{Gr}	Grashof number (dimensionless)
n	Moles H_2O per mole of carbon in refuse
P	Pressure (lb_f/ft^2)

p	Moles CO_2 generated in bed per mole of oxygen (Equation II-1)
Q_L	Energy loss rate by radiation from bed (Btu/lb mole of oxygen)
Q_P	Energy loss rate by convection from bed (Btu/lb mole of oxygen)
Q_R	Energy release rate in bed (Btu/lb mole of oxygen)
Q_T	Total quantity of overfire air (cfm)
Q_u	Energy release rate above bed (Btu/lb mole of oxygen)
q	Term defined in Equation A-6 (lb m/ft^{2}sec^2)
q_H	Heater input power (Appendix D) (Btu/sec)
r	Radial distance from jet centerline (ft)
S	Dimensionless jet spacing (s/d_o)
S_n	Cross-sectional area of jet in crossflow (ft^2)
S_{no}	Cross-sectional area of jet at nozzle (ft^2)
s	Spacing between jets (feet)
T	Gas Temperature (degrees absolute)
T_1	Temperature of furnace gases (°R)
T_C	Temperature of cold gases (°R)
T_F	Minimum ignition temperature of furnace gases (°R)
T_H	Temperature of hot gases (°R)
T_R	Reference temperature for enthalpy (°R)
T_c	Temperature of mixture (after combustion) of nozzle fluid and ambient (°R)
T_j	Average temperature of nozzle and entrained fluid in jet (°R)
T_m	Temperature of mixture (uncombusted) of nozzle fluid and ambient (°R)
T_o	Temperature of jet fluid at nozzle (°R)

LIST OF SYMBOLS (Cont.)

t	Time (sec)
u	Gas velocity (feet/sec)
u_1	Velocity of crossflow fluid (feet/sec)
u'	r.m.s. fluctuating velocity component in axial direction (feet/sec)
u_j	Average velocity of nozzle and entrained fluid in jet (feet/sec)
$\bar{u}$	Time averaged jet velocity along radius (feet/sec)
$\bar{u}_m$	Time averaged jet centerline velocity (feet/sec)
$\bar{u}_o$	Time averaged jet nozzle velocity (feet/sec)
v'	r.m.s. fluctuating velocity component in radial direction (feet/sec)
w	Mass flow per unit area (lb_m/ft^2)
X	Dimensionless distance from jet entry plane
x	Distance from reference point or nozzle (feet)
Y	Dimensionless distance above jet entry plant
y	Vertical distance above reference point or nozzle (feet)
Z	Height of physical system (feet)
z	Height above reference plane (feet)
α	Angle between jet centerline and the horizontal for up-flowing crossflows (degrees)
α_o	Injection angle of jet above horizontal (degrees)
β	Angle between jet and vertical (degrees)
β_o	Injection angle between jet and vertical (degrees)
γ	Secant β (dimensionless)
δ	Temperature coefficient of expansion (reciprocal degrees absolute)

LIST OF SYMBOLS (Cont.)

ε	Exponent in Equation IV-46 (dimensionless)
ζ	Vertical displacement of disturbed interface (Appendix B) (feet)
Λ	Oxygen demand of furnace gases (lb O_2/lb furnace gas)
μ	Viscosity (lb_m/ft sec)
ν	Characteristic length (feet)
ξ	Plane jet half-angle of spread in a channel (radians)
ξ_o	Plane jet half-angle of spread in unconfined space (radians)
ρ	Gas density (lb_m/ft^3)
ρ_a	Ambient fluid density at nozzle (lb_m/ft^3)
ρ_c	Cold gas density (lb_m/ft^3)
ρ_h	Hot gas density (lb_m/ft^3)
ρ_j	Average density of nozzle and entrained fluid in jet (lb_m/ft^3)
ρ_o	Jet fluid density at nozzle (lb_m/ft^3)
σ	Growth exponent (Appendix B) (sec^{-1})
$\phi(x,y)$	Potential function (Appendix B)
ψ	Characteristic thickness of shear layer (Appendix B) (feet)
Ω	Concentration of a component of nozzle fluid in the jet relative to the nozzle concentration (dimensionless)

CHAPTER I

INTRODUCTION AND SUMMARY

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INTRODUCTION AND SUMMARY

A. BACKGROUND

Solid waste disposal by incineration is now and will continue to be an important part of our national solid waste management program. Because of this reality and in the light of increasingly stringent environmental constraints and standards, reliable and effective incinerator design methodologies are needed. As described in a previous Arthur D. Little, Inc. (ADL) report titled "Systems Study of Air Pollution from Municipal Incineration", the technology supporting municipal incinerator design has evolved slowly and is yet immature. Indeed, there are few design parameters which have been well-characterized and proven useful in practice. Most often, the only measures of design adequacy have been such pragmatic "variables" as plant availability or the lack of maintenance and operating headaches. In the areas of combustion efficiency and air pollution performance, there is almost no guidance for the designer; only the documented facts of unacceptably high emissions.

The estimates presented in Table I[1] indicate the magnitude of the incinerator air pollution problem both now and in the future. These emission projections show clearly the important contribution to total emissions of combustible gaseous and particulate pollutants. Previous work has shown that the very existence of these pollutants in incinerator flue gas is a consequence of inadeqaute mixing within the furnace and flues of the incinerator. Specifically, theoretical and experimental studies indicate that at the temperature and air-fuel conditions which are obtainable in well-mixed incinerators, the majority of combustible pollutants are destroyed within a small fraction of the typical gas and particle residence times. Thus, if means can be found to assure that complete mixing occurs witiin incinerator furnaces, the emission levels of most combustible pollutants can almost be eliminated.

The design of most existing municipal incinerators is seriously deficient in provision for overfire mixing. Practical engineering knowledge of design and operating parameters required to optimize mixing and consequent burnout in incinerator combustion chambers is also lacking. A study of combustion chamber mixing factors leading to development of design principles would offer one of the most readily applicable and least expensive means for upgrading operation and reducing combustible emissions from municipal incinerators.

TABLE I

ESTIMATED INCINERATOR EMISSIONS

(thousands of tons per year)

	1968 Emissions Estimate			2000 Emissions Estimate		
	Furnace	Stack	%	Furnace	Stack	%
Combustible Particulate	38	32	7.1	131	49	3.3
Carbon Monoxide	280	280	62.2	829	829	56.0
Hydrocarbons	22	22	4.9	64	64	4.3
Polynuclear Hydrocarbons	0.01	0.005	0	0.03	0.0009	0
Subtotal (Combustibles)	340	334	74.2	1,024	942	63.6
Mineral Particulate	90	56	12.4	708	118	8.0
Sulfur Dioxide	32	32	7.1	161	160	10.8
Nitrogen Oxides	26	22	4.9	147	114	7.7
Hydrogen Chloride	8	6	1.3	219	147	9.9
Volatile Metals	0.3	0.3	1.0	0.055	0.025	0
TOTAL	496	450	100.0	2,259	1,481	100.0

Reference 1

B. OBJECTIVE & SCOPE

Because of the lack of reliable design guidelines for combustible pollutant control, the Office of Air Programs (OAP) of the Environmental Protection Agency initiated studies aimed at providing this technology to the incinerator design industry. This report documents the results of the first steps in these studies:

- A review of the literature and other available sources of information relating fuel bed combustion and combustion chamber mixing to overall furnace combustion performance and when available emission characteristics.
- Development of theoretical and analytical models which describe basic elements of the combustion or mixing processes within the refuse incinerator for use as design guidelines.
- Conduct of laboratory experiments to confirm or amplify the developed theoretical mixing models.
- Development of a plan for testing in an existing municipal incinerator the effectiveness of the overair mixing guidelines established during the program.

C. APPROACH

In order to avoid duplication of past work and to incorporate the experience gained in incinerators and other stoker-fired combustors, a review of the literature was carried out (Chapter VII). Also, discussions were held with firms now engaged in the design of incinerator and fossil-fuel-fired boilers.

After analysis of this information, it was evident that the objectives of the program could best be met by limiting the study to the effects of overfire air and steam jet mixing and of the bed processes giving rise to the combustibles. The effects of baffle systems on mixing were not extensively considered.

The approach in developing design guidelines was to carry out analyses to:

1. Suggest, based on coal-burning experience, step-by-step methods to specify incinerator overfire air mixing system parameters; and

2. Analyze pollutant generation and destruction processes and jet dynamics to contribute to an understanding of the behavior of the pertinent components of the furnace system: the burning bed, the enclosure and its influence on flow and mixing; and jet behavior in furnace environments.

Recognizing the limitations of analysis, an experimental evaluation of operation and emission variables in a state-of-the-art incinerator was seen as the next logical step in providing the fundamental engineering data needed to validate the design guidelines. For this purpose, an incinerator of modern design was selected and a test program was prepared which included, in addition to emission rate characterization, an evaluation of the engineering configuration (flow rates, heat release rates and so forth) of the unit during the tests.

D. SUMMARY

A review of the prior art (documented in the Bibliography--Chapter VII) in combustion design technology yielded nothing oriented specifically toward overfire mixing in incineration systems. Overfire air jet correlations were found which had been developed for application to coal-burning systems. Although these approaches to jet design (Section D of Chapter IV) could be interpreted as directed at realization of complete overfire combustion, their experimental basis shows them to be directed at smoothing the temperature distributions in the gases rising from the bed[2] or the even more pragmatic goal of assuring penetration of the jet flow across the full width of the furnace.[3] As a consequence, their effectiveness in meeting the objective of combustible pollutant control in incinerators is unknown.

Design technology for passive (baffle) systems was found to be largely an empirical art. Consideration of furnace baffles as mixing devices, however, showed them to be relatively ineffective in generating intense turbulence. Baffles are well suited to meeting specific gas flow control needs for a given furnace design but they are too inflexible to effect any reliable level of control of the constantly varying combustion process. Since they have little effect on the needs and design parameters of jet mixing systems, little effort was expended on this aspect of the design art.

The results of our study of the bed burning process (Chapter II), combustion chamber flow (Chapter III), and jet behavior (Chapter IV) yielded useful contributions to better understanding of the incineration process and suggested techniques for design evaluation and equipment specification.

1. THE BED BURNING PROCESS

In order to provide a tractable analysis problem, the refuse pyrolysis and gasification processes were evaluated on an overall basis. Based on data[4] which suggested the off-gases were approximately in equilibrium according to the reaction:

$$CO_2 + H_2 \rightleftarrows CO + H_2O$$

energy and material balances were written which allow estimates of the combustible pollutant loading (CO and H_2) as functions of the refuse composition, moisture content, and underfire air rate. Using this model of bed combustion, the following result:

a. Combustible pollutants will always be emitted from the bed into the overfire volume and, except when large fractions of the undergrate air bypass the bed (channeling), the off-gases always present an oxygen demand.

b. The oxygen demand can be calculated allowing estimation of the minimum overfire air demand as a function of pertinent operating and refuse variables.

c. The refuse pyrolysis mass-addition rate to the undergrate air flow can be estimated to provide input data to analysis of the flow in the furnace enclosure.

d. Calculations of bed behavior for different assumptions regarding refuse and underfire air parameters provide estimates of the flexibility required of jet systems, fans and the like.

2. FURNACE FLOW BEHAVIOR

Complex heat and mass transport phenomena, compounded by combustion effects, make rigorous analysis of furnace pattenns impossible without further simplifying assumptions. In this study, the bed processes were considered as a generator of an input stream to the overfire volume. Combustion effects in the gas phase were ignored and the gas flow was assumed to move, without significant mixing, at velocities corresponding to potential flow in the presence of buoyant forces. As a consequence of the temperature and mass flow distribution along the grate, this resulted in the acceleration of the hot gases entering the enclosure in the pyrolysis region. Methods were developed to relate gas velocity to the input temperature and mass-flow distributions calculated from the bed-burning model and to the physical dimensions and configuration of the furnace enclosure. Methods were also developed to allow simulation of turbulent mixing to develop estimates of the velocity, temperature, and composition (degree of mixing) nonuniformity that could be expected at the discharge plane of the physical system in question.

Using this model of furnace dynamics, the following results:

a. The velocity field in the incinerator furnace can be estimated by multi-zone treatment of the system.

b. Calculated furnace gas velocities can be used in subsequent analyses to determine the magnitude of crossflow effects on overfire air or steam jet trajectory.

c. Calculated velocities can be used to anticipate erosion problems. This can be of considerable importance if boiler tube passes are located in the outlet flue of the furnace.

d. Estimates of the adequacy of alternatives in furnace design to effect needed turbulent mixing without jets can be made.

3. JET BEHAVIOR

At present, most jet behavioral analyses used in incinerator design are based on relationships which describe the behavior of jets discharging into a fluid at rest under relatively isothermal, non-reacting conditions. In an incinerator, however, the jet is cold and the ambient is hot, suggesting the importance of buoyancy effects on jet trajectory. Of equal or greater importance, the jet traverses a crossflowing stream of combustible gases. Thus, in contrast to the bed and flow analyses, the effort on jets sought complexity as it was needed to provide reliable predictions of jet behavior in a complex flow environment.

The effects of crossflow and buoyancy, acting alone, appeared adequately described by existing correlations and experiment. Their combination, however, in the "competitive" situation where buoyancy tends to drop the jet trajectory and crossflow to raise it, was not well characterized. While some workers have indicated that buoyancy effects could be neglected, our analysis showed that under many conditions of incinerator design and operation, buoyancy could dominate the flow. We confirmed the qualitative validity of our analysis by modeling experiments. A method was developed to determine the conditions where buoyancy dominates under given furnace environment conditions (calculated using the bed and flow analyses) to allow avoidance of designs where the jet would sink and disturb the bed (entraining fly ash or overheating the grates). Also, relationships were developed permitting a rough estimate of the effect of combustion on jet temperatures as they may effect overheating of surfaces on which they impinge.

The following summarize the results of the jet behavior analysis:

a. Correlations were developed to compute jet trajectory under combined crossflow and buoyancy conditions, using inputs from the bed and flow analyses which support more confident estimation of jet behavior than in the past.

b. The effect of buoyancy was shown to be important under some design and operating conditions and means were provided to allow evaluation of the trajectory under these conditions.

c. A calculational method was developed for rough evaluation of combustion effects on jet temperature distributions.

4. DESIGN GUIDELINES

The design guidelines reported or developed here fall into two categories: a rule-of-thumb method derived from coal-burning technology and an analytical method to predict the behavior of alternatives in system design. Two generalized design approaches based on coal-burning experience are presented. The first approach is that of Ivanov[2] who based his method on the results of experiments in a laboratory flow apparatus and confirming tests in full-size boilers. His method (Section C-5a and D-4 of Chapter IV) is directed toward smoothing composition and temperature profiles. To the extent that adequate destruction of combustible pollutants will occur under such conditions, the method appears sound.

A second generalized design approach arises from work done at Bituminous Coal Research, Inc. for the National Coal Association.[3] This method (Section C-5b of Chapter IV) appears based on isothermal jet behavioral relationships and has the objective of assuring jet penetration. It does not explicitly address the problem of obtaining mixing or allow compensation for the effects of cross-flow. With these apparent weaknesses, however, it should be noted that these correlations have proven successful in visible smoke abatement in grate bed coal-fired boilers.

The design guidelines based on analysis methods developed under this program are described above. Their application in a design situation is described in Chapter V. There, the analysis is shown to provide estimates of air requirements, burn-out levels, flue gas velocities and temperatures and so forth to give the designer perspective and quantitative estimates of system behavior under varying conditions. It should be recognized that a number of reasonable but largely untested assumptions are required in the course of the analysis. The engineering data to confirm the assumptions are lacking but, to an extent, will be sought in the test program to follow.

5. TEST PROGRAM

The characteristics of the proposed test program are described in Chapter VI. The tests should be divided into three groups. The first test series would provide a reference baseline as to the emissions in the absence of overfire air mixing. The second test series would include experiments to study emission characteristics and chamber flow patterns as they relate to system variables by measurement of gas compositions, velocities and temperatures throughout the furnace. The third test series would show the effects on emission rate of various operating configurations of overfire air jet systems. It is reasonable to expect that with a relatively small number of tests, the effectiveness (if not the reason for the effectiveness) of the overfire air jet system could be demonstrated.

E. REFERENCES

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2. Y. V. Ivanov, "Effective Combustion of Overfire Fuel Gases in Furnaces", Astonian State Pub. House, Tallen (1959).

3. "Layout and Application of Overfire Jets for Smoke Control in Coal-Fired Furnaces", National Coal Association, Washington, D.C., Section F-3, Fuel Engineering Data, December 1962.

4. E. R. Kaiser, personal communication to Walter R. Niessen.

CHAPTER II

BED BURNING MODELS

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BED BURNING MODELS

A. INTRODUCTION

The processes occurring in a refuse bed are extraordinarily complex, involving the interaction of heat transfer, mass transfer, chemical kinetics and fluid flow in a heterogeneous material with a chemical composition and physical configuration which varies in both space and time. Exact analysis of such processes is clearly out of the question. But the processes occurring in the fuel bed generate the volatile matter that must be burned in the overfire regime, and qualitative and rough quantitative models of the fuel bed are needed in order to define the design of the overfire air supply.

To provide a means to estimate the approximate composition and flow rate of combustibles into the overfire volume, a method will be presented to enable calculation of these quantities. The analysis draws from coal and refuse combustion theory and data.

B. PREVIOUS WORK

Guidance for the development of bed burning models is provided by the experimental studies on the Oceanside Incinerator by E. Kaiser[1], on simulated refuse beds by the Bureau of Mines[2], and on coal, coke and lignite beds by a number of investigators.[3,4,5,6]

The pertinent results of Kaiser include measurements of the gas concentration at a number of positions above a refuse bed, and calculations, based on the gas analyses, of the distribution of heat release rate both within and above the fuel bed.

The schematic elevation and plan views of the Oceanside Incinerator, Figure I-1, show the positions at which gas samples were obtained. The operating conditions of the unit corresponded to a refuse feed rate of 300 TPD and a total air flow of 150-200 percent excess air, 60 to 80 percent of which was supplied under the grate through three of the four available windboxes at windbox pressures as shown in Figure II-1. Additional observations on the operation of the Oceanside Incinerator by Kaiser are that:

1. The ignition plane intersected the grate at approximately the midpoint of the second windbox from the feed chute for regular refuse and toward the end of the second windbox for refuse with 43 percent moisture content.

2. Approximately half the undergrate air was supplied by the second windbox, three-eighths in the third, and one-eighth in the fourth. No forced undergrate air was supplied to the first windbox.

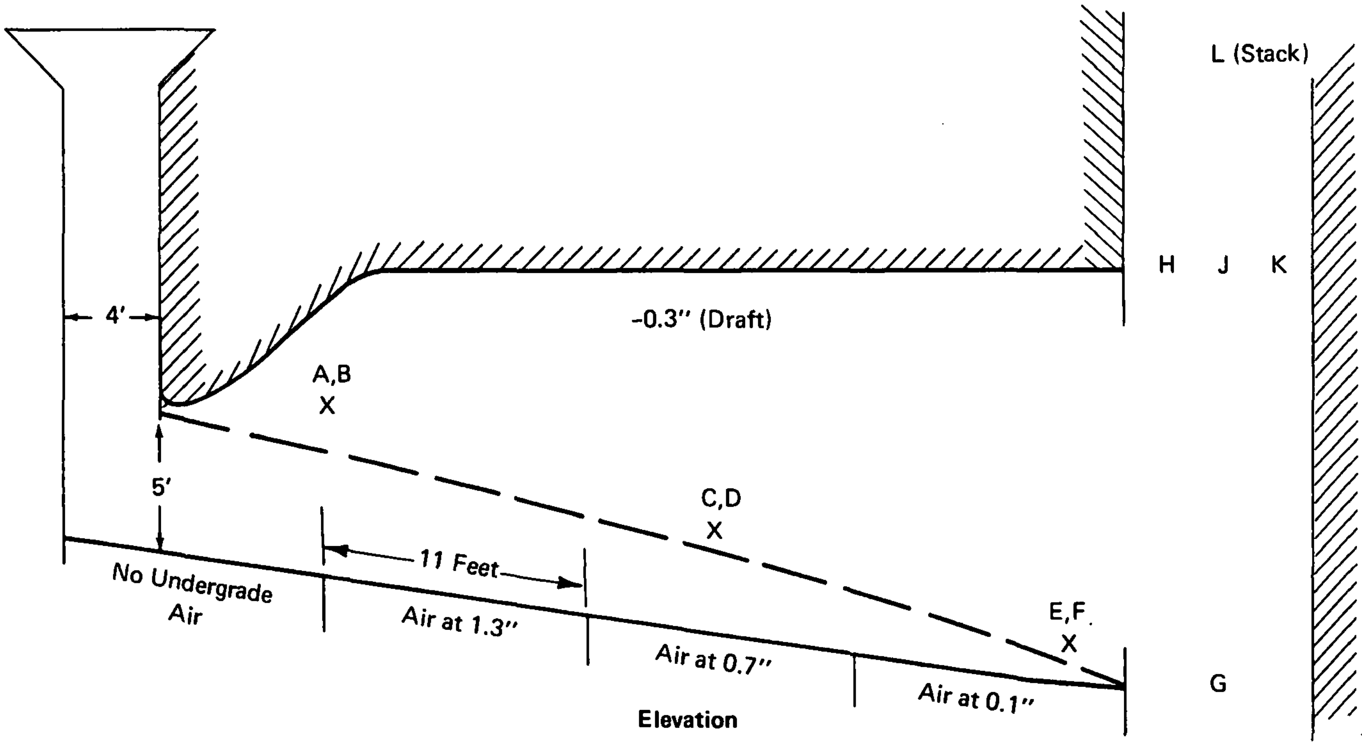
3. Very little flame was observed over the fourth windbox.

4. Approximately 9 percent of the total feed, containing 3 to 4 percent of the combustible, was lost as siftings through the openings (20 percent of the total area) in the grate.

5. Two percent of the feed was carried off as fly ash containing 50 percent carbon and 16 percent, containing 4 percent combustible, was discharged off the end of the grate.

A summary of the gas compositions and temperatures reported by Kaiser are given in Table II-1. The total carbon in the gases was obtained from the sum of the carbon contents of the CO_2, CO, CH_4 and of the particulate and liquid collected in the sample train, with the exception of two runs in which only the gas contributions were included.

From consideration of the above data and observations, Kaiser computed the distribution summarized in Figure II-2 of the rate of energy release within and above the fuel bed. These results by Kaiser are the only substantive figures for incinerators, and merit special consideration.



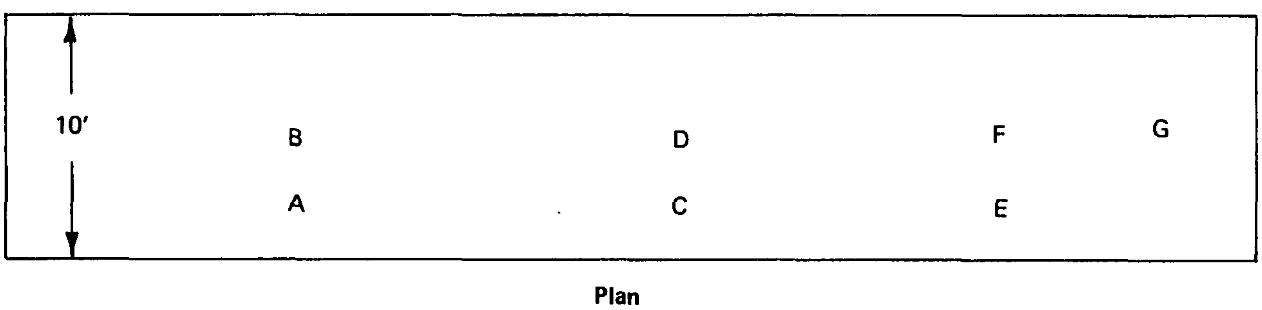


FIGURE II–1 SCHEMATIC OF OCEANSIDE INCINERATOR SHOWING LOCATION OF GAS SAMPLE POSITIONS

TABLE II-1

SUMMARY OF OCEANSIDE DATA [1]

Location	CO_2	O_2	CO	H_2	CH_4	N_2	T (°F)*	% C Burned to CO_2	% C Burned to CO	Unburned	Steam M lbs/hr	Remarks
A	12.53	6.38	0.75	0.45	0.20	79.69	1418	92.90	5.58	1.52	45.0	
B	14.30	1.83	4.47	3.38	0.13	75.89	1693	75.61	23.70	0.69	69.2	Gas Only
B	15.37	0.83	6.43	3.07	0.77	73.53	1789	68.07	28.51	3.42	72.0	
C	8.40	10.40	1.73	1.36	0.30	71.81	1316	76.41	15.54	8.05	40.3	
D	13.43	2.47	7.91	4.71	0.41	71.07	1634	58.32	34.73	6.95	55.3	
E	0.17	20.67	0.00	0.00	0.20	78.96	1679	45.79	0.0	54.21	58.1	
F	0.27	20.46	0.00	0.07	0.32	78.88	1695	40.68	0.0	59.32	62.1	Gas Only
G	0.0	20.60	0.00	0.07	0.00	79.33	1510	0.0	0.0	0.0	59.3	
H	9.25	10.38	0.0	0.04	0.18	80.15	1287	97.20	0.0	2.80	29.9	
H	12.78	5.85	0.0	0.01	0.20	81.16	1605	97.98	0.0	2.02	64.8	
J	3.53	16.53	0.0	0.14	0.15	79.65	1460	95.28	0.0	4.72	28.7	
K	0.95	19.38	0.0	0.02	0.23	79.42	1236	99.75	0.0	20.25	48.0	
L	5.20	14.78	0.0	0.02	0.20	79.80	1322	95.15	0.0	4.85	39.1	
L	5.80	13.83	0.0	0.02	0.15	80.20	1563	97.41	0.0	2.59	62.5	

* Thermocouple near sidewall at J

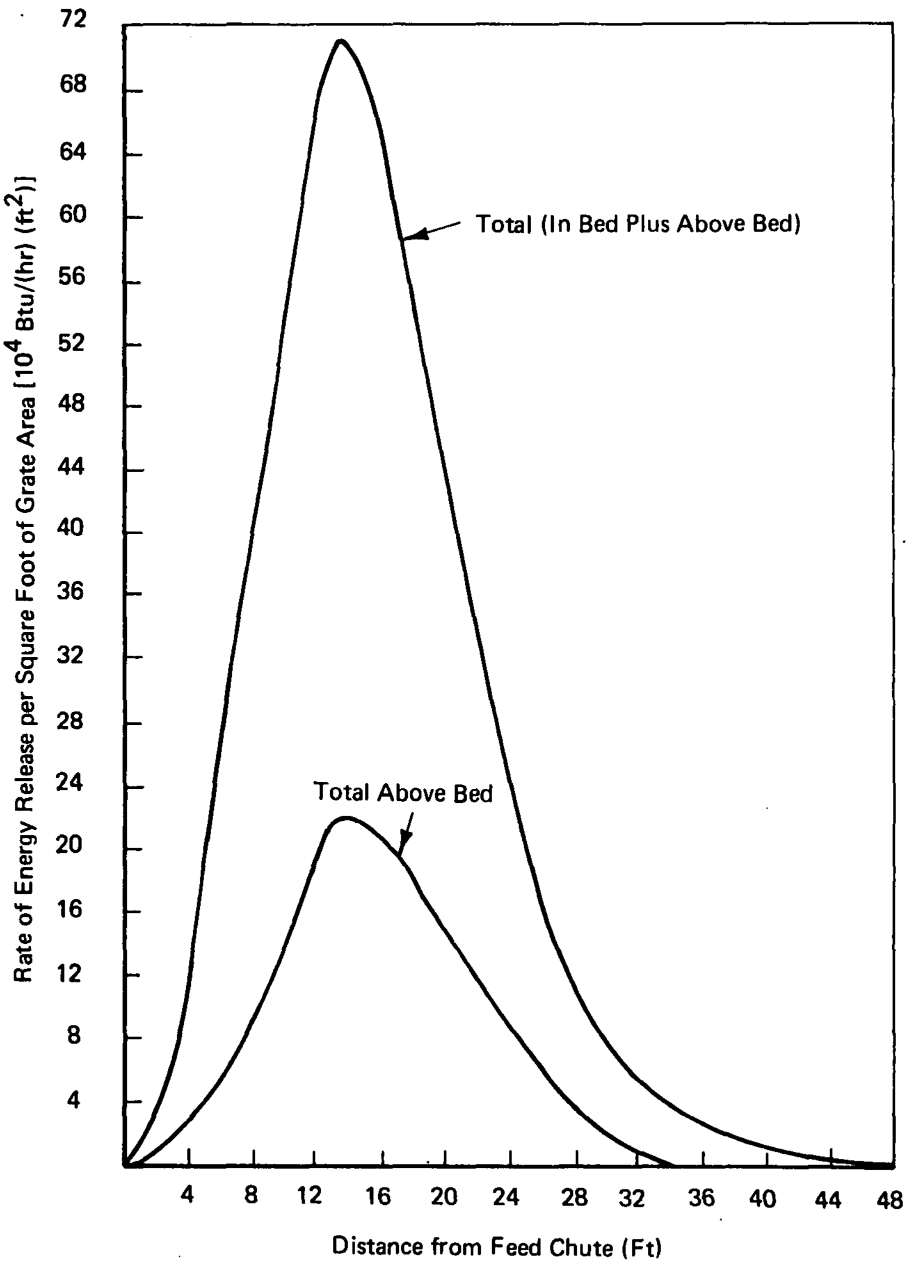


FIGURE II–2 ESTIMATED ENERGY RELEASE RATES FOR OCEANSIDE INCINERATOR[1]

One can make the following further observations:

1. In the gasification and burning regimes of interest, most of the carbon in the refuse is oxidized to either CO_2 or CO. An apparent exception is at locations E and F where the composition is close to that of air. This resulted from the fact that at these grate positions burnout of the refuse was essentially complete and only trace amounts of combustion and gasification products were present.

2. Some of the air does not react within the fuel bed as a consequence of by-passing the combustible through blow-holes or through an area on the grate covered by non-combustibles. From the data in Table II-1, the amount of unreacted air by-passing can be estimated. The values shown in Table II-2 were obtained by assuming that the combustible content of refuse could be approximated by $C(H_2O)_n$. Clearly, the amount of air by-passing the refuse will vary with refuse type and loading conditions. The limited available data suggest that on the average about 25 percent of the air by-passes the fuel.

TABLE II-2

PERCENTAGE OF UNREACTED AIR BY-PASSING

Position	A	B	B	C	D
Percentage	32.8	10	4.2	53	12.4

3. The products of combustion appear to reflect equilibrium of the water-gas shift reaction: ($CO_2 + H_2 \rightleftarrows CO + H_2O$). To test this postulate, the water vapor content of the gases leaving the bed must be estimated by assuming the H/C ratio is in the range 1.5 to 2.0, bracketing data for municipal waste.[7] Table II-3 presents the values of the water-gas shift constant and the corresponding equilibrium temperature for the two postulated H/C ratios. The calculated temperatures are within the range 1500-2000°F of temperatures commonly found in burning fuel beds. These results suggest that the water-gas equilibrium is approached within the fuel bed. This behavior is very similar to that observed in the gasification of coal, lignite and wood[8,9] for which equilibrium is also approached when the temperatures are in the range 1500-2000°F.

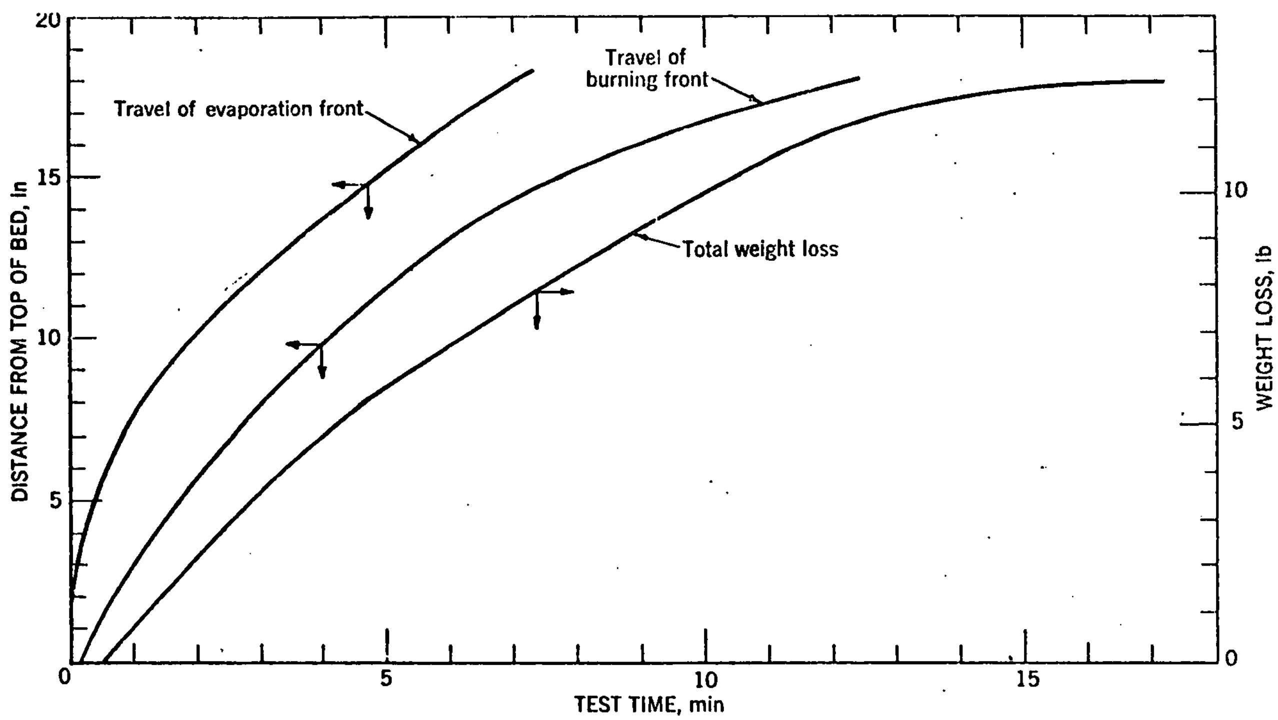
TABLE II-3

WATER-GAS-SHIFT EQUILIBRIUM TEMPERATURE

	Position	A	B	B	C	D
H/C = 2.0	$\frac{P_{H_2} P_{CO_2}}{P_{H_2O} P_{CO}}$	0.595	0.7	0.405	0.73	0.464
	T(°F)	1810	1690	2190	1635	2020
H/C = 1.5	$\frac{P_{H_2} P_{CO_2}}{P_{H_2O} P_{CO}}$	0.813	1.03	0.595	1.053	0.685
	T(°F)	1603	1500	1810	1490	1710

The additional observation made by Kaiser that an ignition wave propagates through the fuel bed is supported by results on a U.S. Bureau of Mines batch fed 19-inch I.D. test incinerator. Thermocouples placed within a simulated refuse bed at 6-inch intervals showed the propagation of a drying plane and an ignition plane through the bed. Figure II-3, taken from the U.S. Bureau of Mines Report[2], shows the rates of propagation of the evaporation and burning front. Additional unpublished results provided to ADL by the U.S.B.M. support the conclusions drawn from Figure II-3. Unfortunately, the burning rates obtained in the U.S.B.M. tests were obtained mostly with little or no underfire and therefore are not directly applicable to municipal incinerators.

The results by Kaiser and U.S.B.M., though limited, illustrate that burning refuse beds have characteristics similar to those observed in the burning of other solid fuels. The extensive literature on the burning and gasification of coal in beds is, therefore, very pertinent to the present study.



Source: Reference 2

FIGURE II–3 TRAVEL OF EVAPORATION AND BURNING FRONTS AND TOTAL WEIGHT LOSS AS FUNCTIONS OF TEST TIME IN A TYPICAL BED OF REFUSE HAVING A MOISTURE CONTENT OF 50 PERCENT AND A BED DEPTH OF 18 INCHES IN BU MINES (19-Inch Diameter Test Incinerator)

The classical experimental and theoretical studies of the processes occurring in a burning fuel bed are those of Kreisinger et. al.,[3,5] Nicholls,[4] and Mayer.[10] In the discussion to follow, distinctions will be made between overfeed, underfeed, and channel burning, which roughly approximate the processes occurring at various positions on a traveling-grate stoker. As shown schematically in Figure II-4, air and fuel flow cocurrently in an underfeed stoker, and countercurrently in an overfeed stoker. The former corresponds roughly to the processes occurring at the front end of a traveling-grate stoker where the ignition plane is traveling towards the grate and the latter to the char burnout region where the carbon burns preferentially at the ash char interface. Channel burning refers to air flowing through a fuel channel and is presented here as an approximate model of the conditions in a blowhole in a refuse bed. It is important to note the distinction between the propagation of the ignition plane, corresponding to the "plane" at which the fuel ignites, and of the burning plane, corresponding to the plane at which the fuel has been completely burnt.

Concentration profiles obtained by Nicholls for underfeed operation with an Illinois Coal (35% volatile) are shown in Figure II-5. The oxygen is rapidly consumed above the ignition plane, with CO_2 as the main product. The CO_2 subsequently reacts with the coked coal particles to form some CO. Small amounts of soot, tar and methane are formed primarily in a pyrolysis zone near the ignition plane. The concentration of the products correspond to a water-gas-equilibrium temperature of about 1800°F, but is subject to uncertainty resulting from probable errors in the estimates of the low H_2O concentrations. The results are in agreement with the conclusions drawn from Kaiser's results on the Oceanside Incinerator that most of the carbon is converted to CO_2 or CO and that the water-gas reaction is close to equilibrium. For the runs on the Illinois coal approximately one-third of the heat of combustion is released within the bed and two-thirds above the bed, in contrast to Kaiser's estimates shown in Figure II-2 which suggest that approximately two thirds of the heat is released in the bed of the Oceanside Incinerator.

Additional information of value derived from Nicholls' results is the effect of underfire air rate on the rates of ignition and burning. The values for a high-temperature coke are shown in Figure II-6. At low air velocities, the rate of ignition is greater than that of burning, and, therefore, the depth of the burning fuel zone increases with time until the ignition plane intersects the grate. The rate of ignition imposes an upper limit on the rate of burning. As the air velocity is increased, the burning rate increases faster than the ignition wave up to the point that the two velocities become equal. The change in structure of a coal-burning bed as a consequence of the above trends has been shown by Marksell and Miller.[6] Figure II-7, taken from their paper, shows the volatile and ash content of six slices within their bed with the bottom slice numbered 1. As the ignition wave propagates through the bed, the volatile content fall from the original value of 36 percent to zero with the topmost layers being devolatized first (Figures II-7a and 7c). At low air rates, the burning rate is relatively slow and the residual char combustion is not completed

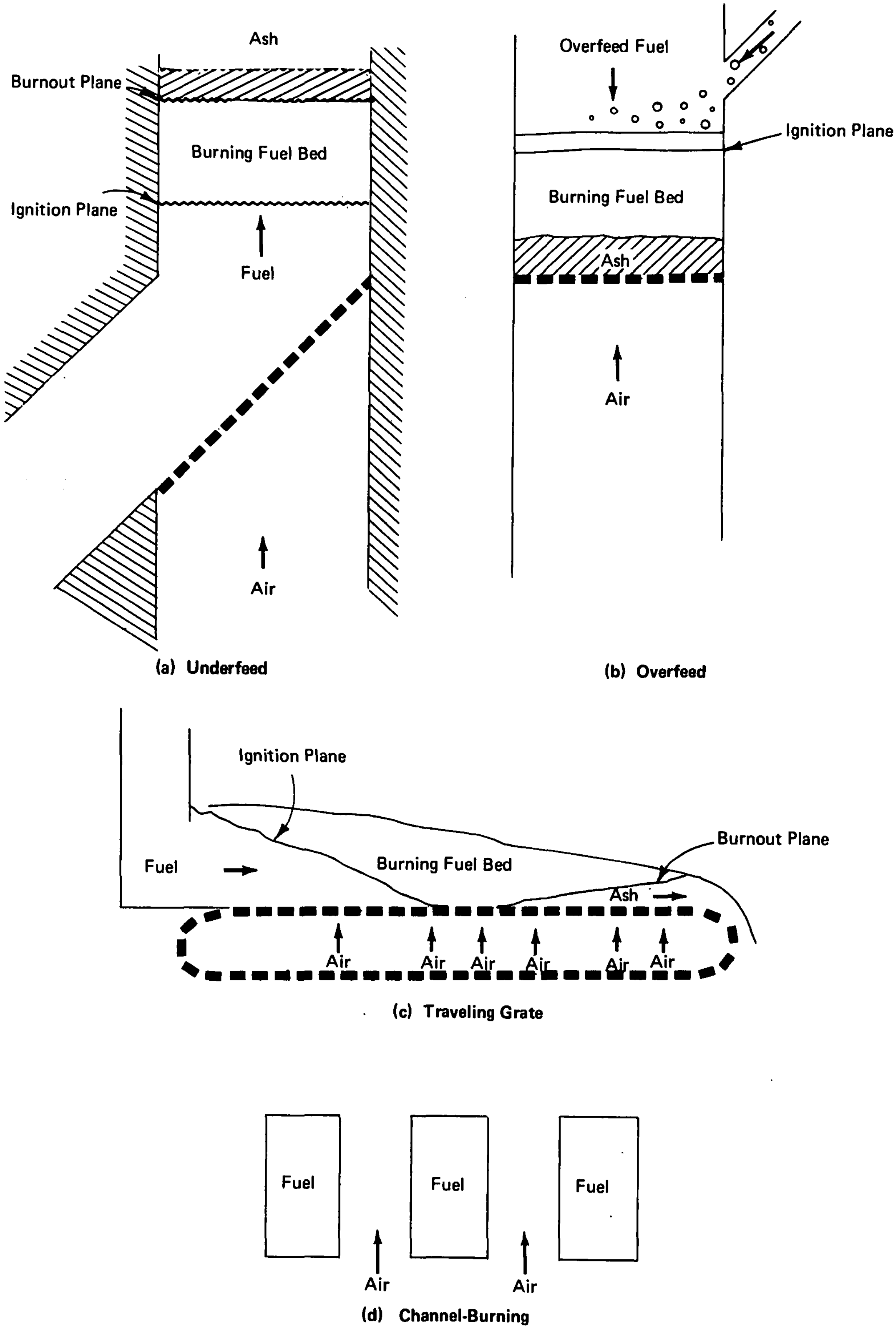


FIGURE II–4 SCHEMATIC DIAGRAMS OF BED-BURNING MODELS

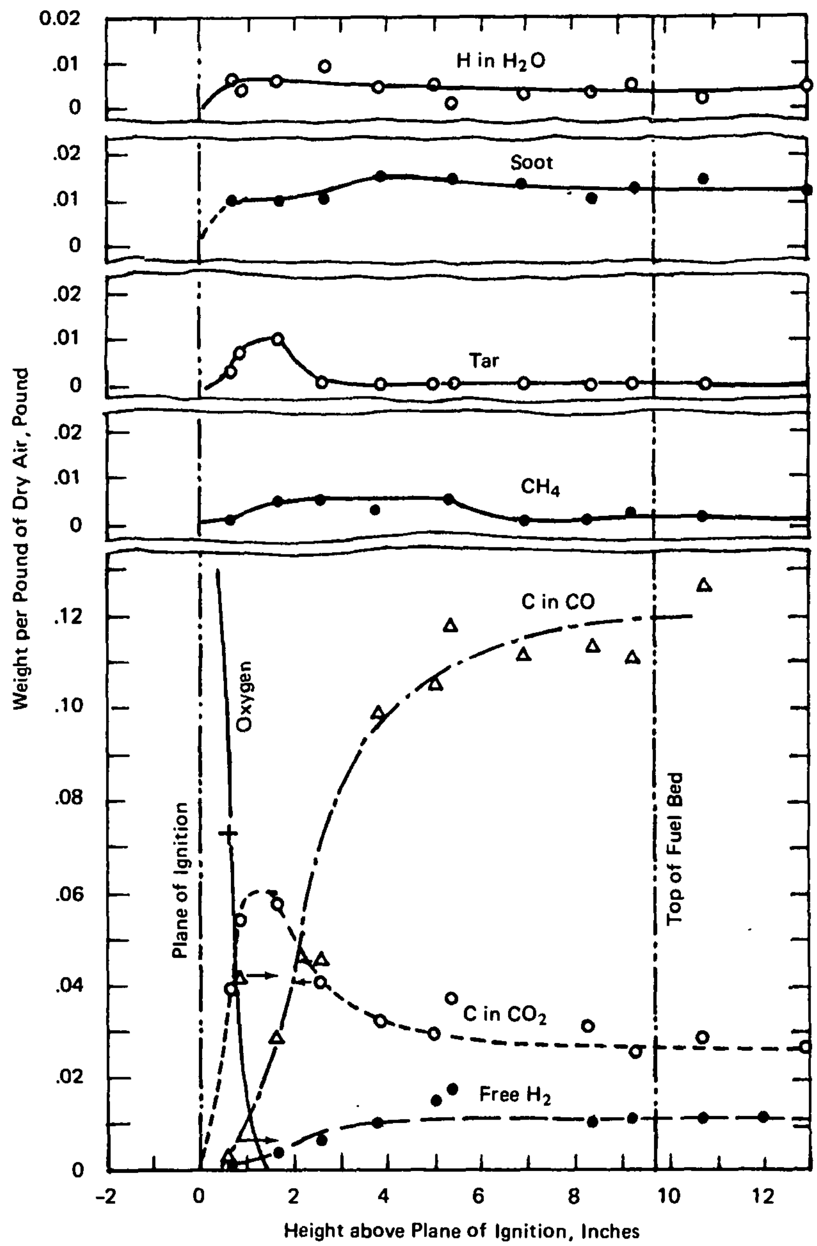


FIGURE II–5 UNDERFEED BURNING, ACTION THROUGH COAL BED EXPRESSED AS WEIGHT OF FUEL PRODUCTS CARRIED PER POUND OF DRY AIR SUPPLIED, 3/4 - 1 INCH – ILLINOIS COAL

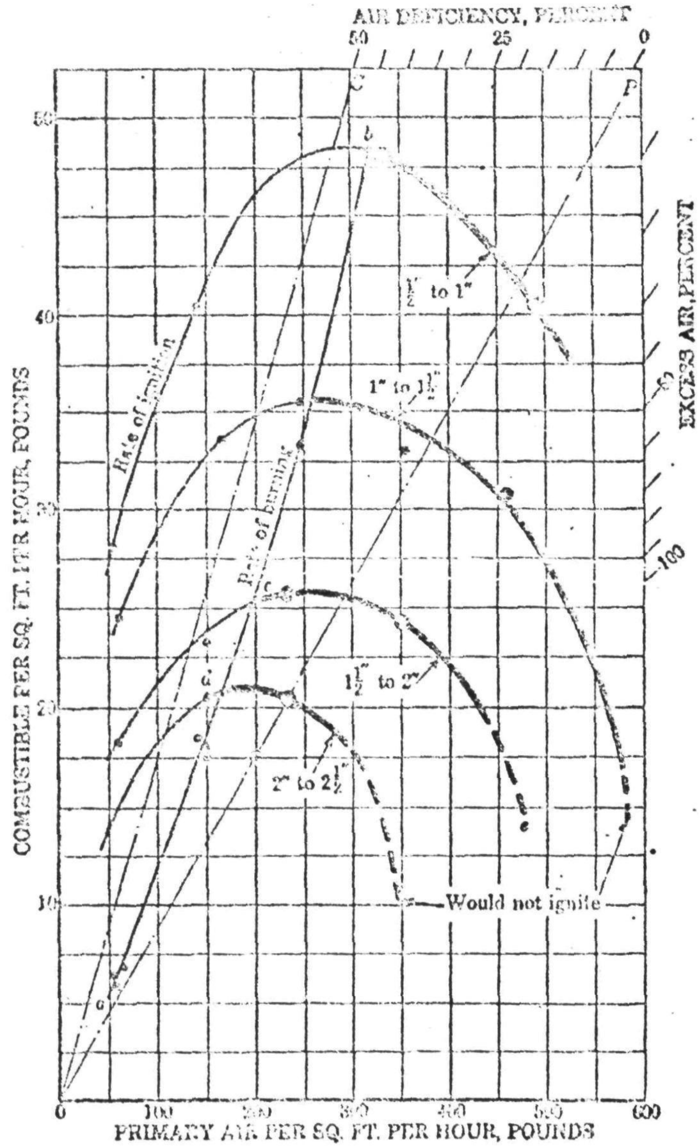
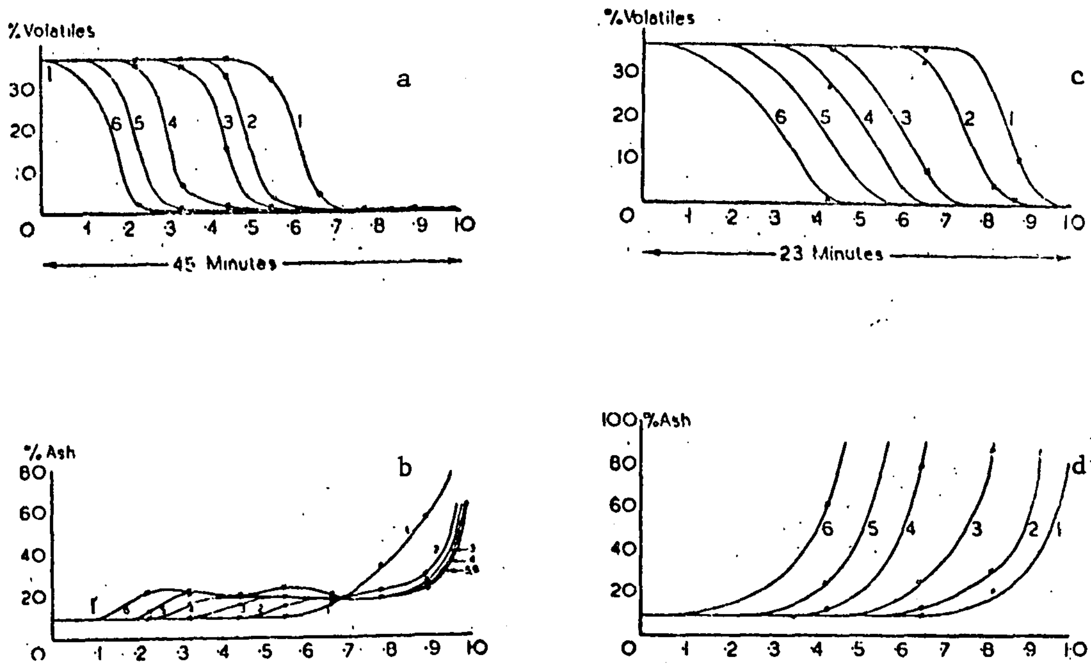


FIGURE II–6 UNDERFEED BURNING, HIGH-TEMPERATURE COKE; RATE OF IGNITION AND RATE OF BURNING WITH RATE OR PRIMARY AIR AND SIZE OF COKE AS VARIABLES



Fraction of Combustion Time

Fraction of Combustion Time

FIGURE II–7 VOLATILE MATTER AND ASH CONTENTS OF COAL BED LAYERS, LOW AIR RATE

until the ignition wave reaches the grate, at which stage the bed burns in an overfeed mode with the ash content building up to 100 percent in the bottom layer first and the top layer last (Figure II-7b). At high air rates, the rates of ignition and burning are equal and devolatization and ashing take place simultaneously (Figures II-7c and 7d). The iso-volatile and iso-ash counters for the conditions of Figure II-7 are presented in Figure II-8. Plots (a) and (b) show devolatilization preceding burnout; (c) and (d) show simultaneous devolatilization and burnout.

The high air rate case has been presented here for completeness. For refuse, excessive particle entrainment from the fuel bed is expected at velocities lower than that at which the propagation of the ignition wave becomes limiting. In such cases, the rate of burning within the bed is expected to be proportional to the underfire air supply.

Studies[5] on lignite with high moisture (40%) and low fixed carbon content (25%) showed results very similar to those for coal. It is expected, therefore, that the same principles apply in the different fuel beds and that consequently the above conclusions can be used to develop qualitative and quantitative models for refuse beds.

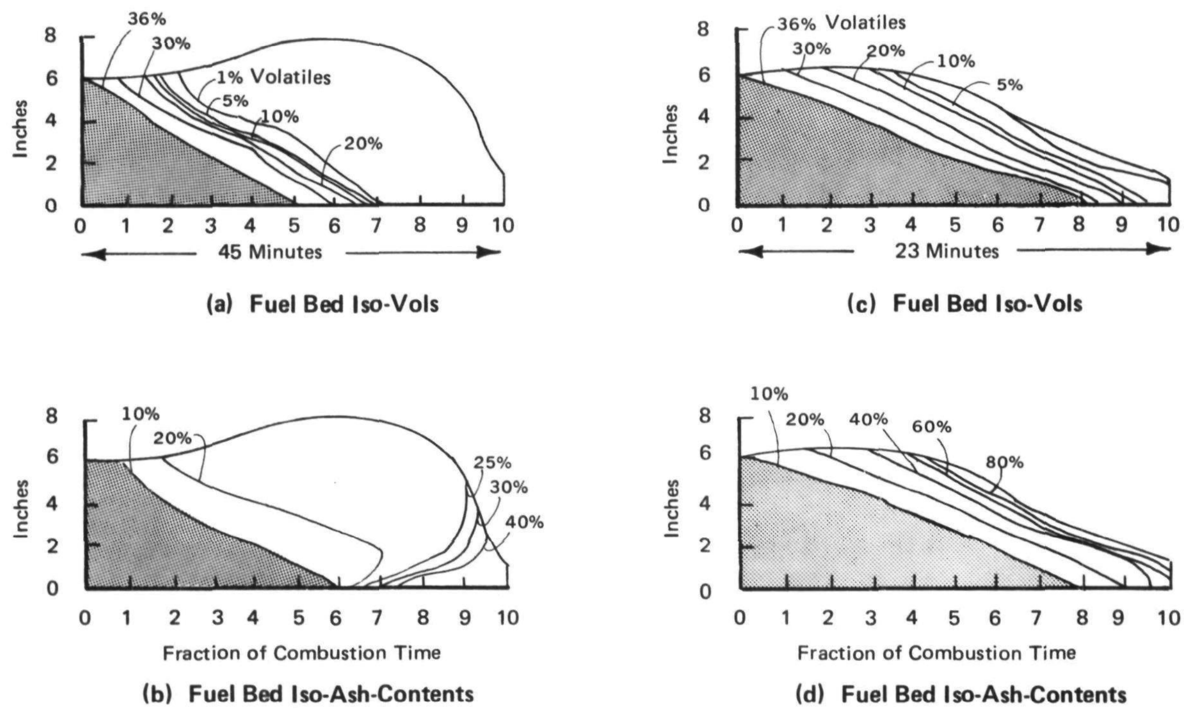


FIGURE II–8 ISO-VOL. AND ISO-ASH CONTENTS THROUGHOUT COAL BED (a) AND (b), LOW AIR RATE, (c) AND (d) AT HIGH AIR RATE

C. BED COMBUSTION MODEL

A qualitative model of a refuse bed is shown in Figure II-9. Drying and ignition waves propagate through the refuse, followed by a devolatization zone. The gases from the pyrolysis section pass through the char bed from the top of which a mixture of mainly low-molecular weight gases emerge with small amounts of soot and tars. The char that is formed follows an overfeed burning mechanism. Order of magnitude estimates of the dimensions of the different zones may be obtained from the discussions in the preceding sections. The rate of propagation of the drying and ignition waves is determined by the rate at which energy is transferred ahead of the propagating front. The rate has been shown to be a function of undergrate air supply (Figure II-6), particle size (Figure II-6), air preheat,[4] moisture content,[2] and fuel type.[4] For refuse, the rate estimated by Kaiser for tests on the Oceanside Incinerator vary from 0.3 feet per minute for wet refuse to 0.5 feet per minute for average refuse. The U.S.B.M. tests indicate that the distance of separation of the ignition and drying wave is of the order to 0.5 feet, but their ignition rates measured with little underfire air were lower than those observed by Kaiser.[1]

The total burning rate will be determined by the rate of supply of undergrate air. These can be calculated by drawing on the results from pyrolysis studies[11], showing that the amount of char produced is expected to be in the range of 0.10 to 0.2 pounds per pound of refuse. The oxygen requirement for the 0.9 to 0.8 pounds of refuse gasified will be determined by the water-gas shift and enthalpy requirements as will be discussed below. Once the refuse has been completely devolatized, the rate of burnout of the char will be determined by the rate of oxygen supply, with the combustion first yielding CO_2, which then reacts with more carbon to yield CO. The thickness of the regimes of thermal pyrolysis and of burning char observed in Nicholls' experiments on coal are of the order of a few inches. For refuse, these will be significantly larger, since the larger refuse elements will present major diffusional resistances to heat transfer for the pyrolysis reactions and to oxygen and CO_2 transfer in the char burning regions. Determinations of the depth of these layers would require much more detailed modeling of the kinetic processes occurring within a bed. The development of such models has been pioneered by Mayers[10] who was able to predict with remarkable accuracy the gas concentration profiles measured by Nicholls in coke beds. A modification of the Mayers' model which is simpler to apply has been proposed by Stewart[13], but both Mayers and Stewart do not make allowance in their models for the diffusional resistance within pyrolyzing and burning particles.

The above sections indicate that the gross behavior of burning fuel beds are very strongly tied to the air distribution below the bed. Increase in air rates results in proportional increases in overall gasification or char-burning rates. These increased air rates often will result in increases in the widths of the gasification (or volatization) and char-burning zones. In the present study, the thickness of the different zones is of secondary importance and, therefore, emphasis will be focused on the gasification rates.

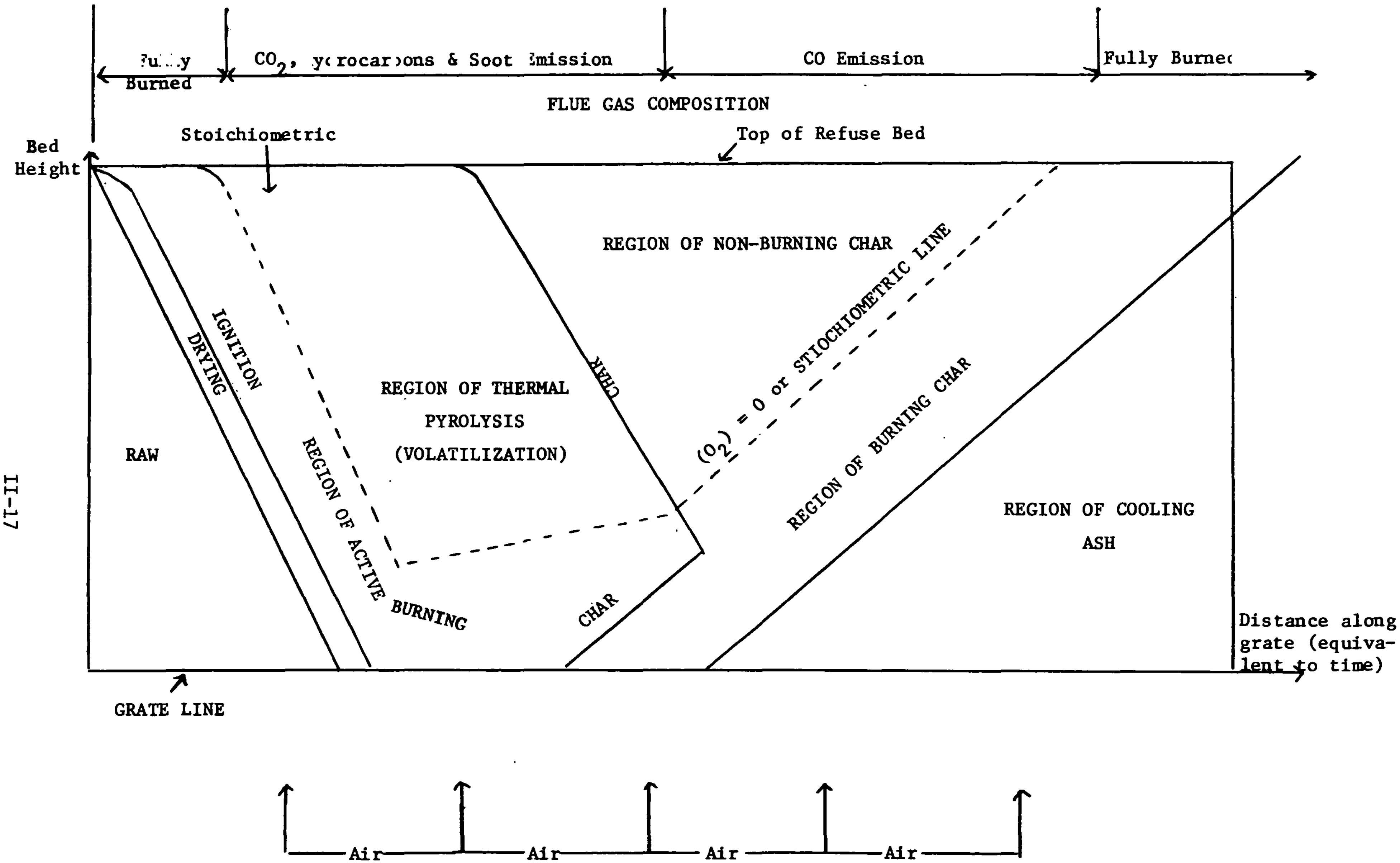


FIGURE II–9 SCHEMATIC OF CROSS-FEED BED BURNING PROCESS
(Assuming Combustion Process Raw→Dry→Volatilize→Char→Ash)

A quantitative model will be developed first for complete gasification, corresponding, in terms of Nicholls' experiments, to the regime where the rates of ignition and burning are equal. For purposes of simplifying the calculations, the organic content of refuse will be modeled by cellulose $[C_6(H_2O)_5]_n$. A synthetic refuse with 22 percent inert, 25 percent moisture, and 53 percent cellulose would have properties very similar to an average municipal refuse with a higher heat combustion of 4300 Btu/lb.

The gasification reactions occurring within the bed produce CO, H_2, H_2O, CO_2, and small amounts of CH_4, tar, and soot from the refuse. As a good approximation, as may be seen from both Kaiser's and Nicholls' results, the CH_4, tar and soot may be neglected. The gasification products can then be calculated from consideration of the stoichiometric relationship between products and reactants, the water-gas shift equilibrium, and an overall energy balance. For purposes of computation ease, the composition of the inert-free content of moist refuse can be simulated by $C(H_2O)_n$ with n having a value of (5/6) for a dry cellulose, 1.55 for a refuse with 23% inerts and 25% moisture, and 2.0 for a refuse with 23% inerts and 34% moisture. Designating the number of units of $C(H_2O)_n$ gasified by one mole of oxygen by x, one can formulate the gasification reaction as follows:

$$O_2 + j\ C(H_2O)_n + 3.76\ N_2 \rightarrow a\ CO + p\ CO_2 + i\ H_2 + d\ H_2O + e\ C + 3.76\ N_2 \qquad \text{(II-1)}$$

Equation (II-1) contains six unknowns; "e", the amount of char residue produced per mole of oxygen during gasification, has been shown in the discussions above to be a function of the rates of ignition and burning. At high air rates, when the burning and ignition rates are equal, e is equal to zero. At lower air rates, the value of e will depend on the rate of gasification, but it has an upper limit imposed by the energy requirements of the overall gasification reaction. In the numerical illustrations to follow, the values of 1 and 0.5 will be assumed as the fraction of carbon gasified (equal to the quantity [1-e/x]) to cover cases of high and intermediate air rates.

With the relationship between e and x assumed, the residual unknowns in equation (II-1) number five, and solutions can be obtained from consideration of carbon, hydrogen, and oxygen balances, the water-gas shift equilibrium, and an overall energy balance. The element balances yield:

Carbon	$a + p + e = j$	(II-2)
Hydrogen	$i + d = nx$	(II-3)
Oxygen	$p + (a + d)/2 = 1 + nj/2$	(II-4)

For the water-gas equilibrium,

$$\frac{[CO_2][H_2]}{[H_2O][CO]} = \frac{(p)(i)}{(d)(a)} = k_w \qquad \text{(II-5)}$$

The basis for the energy balance is the requirement that the heat generated by the gasification reactions provide the sensible energy of the gasification products and the heat loss by the section of the fuel bed in question to other parts of the fuel bed or to the furnace enclosure and grate. The amount of energy release Q_R, in Btu per pound mole of oxygen, is given by the difference of the heats of formation of the products and reactants:

$$Q_R = 47{,}560a + 169{,}290p + 104{,}240d - 320{,}940(\frac{j}{6}) - 104{,}240\ (n-5/6)j \qquad \text{(II-6)}$$

The first three terms represent the CO, CO_2, and H_2O contributions; the fourth term, the heat of formation of cellulose; and the last term is a subtractive term for the moisture content of the refuse exclusive of the (5/6) mole per carbon atom that is chemically bound in the cellulose. The heat requirements are those for the vaporization of the (n-5/6)x moles of water vapor and the heating of all the gasification products to the temperature at which the gases leave the fuel bed. Although the temperature of the leaving gases depends on the conditions within the bed, the variation of temperature with conditions is not large, and will be neglected here. A temperature of 2000°F is selected since this is representative of Nicholls' results on coke and coal beds and is also consistent with the water-gas-shift equilibrium temperatures calculated from Kaiser's concentration measurements above the fuel bed in the Oceanside Incinerator. The average heat capacity in Btu/(mole)(°F) between 60°F and 2000°F for the gasification products are 13.8 for CO_2, 11.0 for H_2O, 8.3 for CO, 7.6 for H_2, and 8.2 for N_2. The energy convected out of the bed by the gasification products is then given by:

$$Q_P = (8.3a + 13.8p + 7.6i + 11d + 30.8)(1940) + (18{,}700)(n-5/6)j \qquad \text{(II-7)}$$

where the units of Q_P are Btu per pound mole of oxygen used in the gasification reaction. If the rate of heat loss from the bed, again in Btu per mole of oxygen, is Q_L,

$$Q_R = Q_P + Q_L \qquad \text{(II-8)}$$

D. CALCULATIONS USING BED COMBUSTION MODEL

Equations (II-2) through (II-8) provide the relations necessary for the calculation of the composition of the gasification products. These are presented in Table II-4 for a limited number of combinations of moisture content, heat loss or gain by the bed, and fraction of carbon gasified. Case 1 is that of the complete gasification of a dry refuse with no net heat loss or gain. For every mole of oxygen introduced with the undergrate air, 2.22 moles CO, 0.62 moles CO_2, 1.47 moles H_2, and 0.91 moles of H_2O are generated within the bed. The moles of $C(H_2O)_{5/6}$ gasified per mole of oxygen equal 2.85. For the dry refuse, the combustion products are rich in CO and hydrogen as shown in the first column of Table II-4. The energy release Q_u above the bed can be calculated from the energies released on the completion of combustion of the carbon monoxide and hydrogen. In Btu per mole of oxygen, it is given by:

$$Q_u = 121{,}730a + 104{,}240i$$

For the conditions of Case 1, it is predicted that 74 percent of the energy will be released above the bed.

Comparison of Cases 1 and 2 in Table II-4 show the effect of increasing the moisture content of the simulated refuse from 0 to 33 percent. The amount gasified is reduced to 58 percent of its previous value; the combustible content of the gases generated is greatly reduced; and the predicted percentage of heat released above the bed is reduced to 44 percent from 74 percent. The explanation for these trends is evident when gasification is thought to consist of the following sequential steps:

$$O_2 + 3.76\ N_2 + C \rightarrow CO_2 + 3.76\ N_2,\ Q = 169{,}290\ \text{Btu/mole}$$

$$O_2 + 3.76\ N_2 + C(H_2O)_n \rightarrow CO_2 + nH_2O + 3.76\ N_2,\ Q = 54{,}950 - 22{,}200\ (n-5/6)$$

$$C(H_2O)_n \rightarrow C + nH_2O,\ Q = 8{,}600 - 22{,}200\ (n-5/6)$$

$$H_2O + C \rightarrow CO + H_2,\ Q = -36{,}620$$

$$CO_2 + C \rightarrow 2CO,\ Q = -44{,}200$$

The first step is the exothermic oxidation of carbon or cellulose to CO_2, as shown in the first two reactions. The heat evolved in these steps provides the energy for the endothermic CO_2-C and H_2O-C reations. The energy release given after each reaction includes that required to heat the gases up to 2000°F. The decomposition of refuse to char shown in the third reaction may be exothermic or endothermic depending on the moisture content. For moist refuse the excess energy generated by the first three reactions is reduced and therefore the last two gasification reactions can-

TABLE II-4

GASIFICATION PRODUCTS AS A FUNCTION OF MOISTURE CONTENT, FRACTION OF CARBON GASIFIED, AND HEAT LOSS

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
A. ASSUMED CONDITIONS						
Moisture, moles per C atom $\equiv (n - 5/6)$	0	1.0	1/6	1/6	1/6	1/6
Fraction of carbon gasified $\equiv 1-e/j$	1.0	1.0	0.5	0.5	0.5	0.5
Heat Gain (Loss) Btu/mole $O_2 \equiv Q_L$	0	0	182,000	(21,000)	(56,000)	(94,000)
B. CALCULATED VALUES						
Mole CO/mole $O_2 \equiv a$	2.22	0.67	5.0	1.49	0.82	0.13
Mole CO_2/mole $O_2 \equiv p$	0.62	0.98	2.1	1.26	1.12	1.00
Mole H_2/mole $O_2 \equiv i$	1.47	0.63	7.3	2.01	1.05	0.13
Mole H_2O/mole $O_2 \equiv d$	0.91	2.39	7.0	3.49	2.82	2.13
Mole $C(H_2O)_n$ gasified/mole $O_2 \equiv x$	2.85	1.65	14.3	5.5	3.88	2.27
Gas Composition, Dry Basis,						
%CO	27.5	11.1	27.8	17.5	12.2	2.6
%CO_2	7.7	16.2	11.6	14.7	16.6	19.8
%H_2	18.2	10.4	39.9	23.6	15.5	2.6
%N_2	46.6	62.3	20.7	44.1	55.7	75.0
% of energy released above bed	74.0	44.0	81.0	60.0	46.0	11.0

not proceed to any significant extent. The importance of supplying energy for the gasification reactions is underlined by the results for cases 3 to 6, calculated for a postulated fixed low moisture content (9%) and a fixed percentage of carbon gasified (50%), but for a varying rate of energy addition or loss. When energy is added to the bed (Case 3), for example, by intense radiation from an overhead flame in the early portion of the grate (before an insulating layer builds up above the gasification zone), the refuse gasified per mole of oxygen and the combustible content of the product gases increase. When energy is withdrawn, (Cases 4 to 6) for example, by radiation to cooled walls (as in a boiler) or to the grate, these quantities decrease.

The gas compositions and the amount of energy released above the bed cover a wide range which encompasses the results obtained by Kaiser[1] and summarized in Table 1. Good matches with the measured values could be obtained by suitable adjustment of the moisture content and energy loss. For Cases 3 to 6 in which part of the carbon in the refuse is left as a char during gasification, the gasification reactions will be followed by a carbon burnout zone. The rates of reaction in this zone are again limited by the rate of oxygen supply. The product gases are mostly CO, sometimes with a little CO_2. The composition of the gases leaving the bed is often determined by the $CO_2 + C \rightarrow 2CO$ equilibrium at a temperature close to 2000°F and is predominantly CO. The total length of the char burnout section may then be calculated from the length required to supply oxygen to convert the carbon residue, mostly to CO.

The above analyses are based on the assumption that the gasification and burnout processes are limited by the rate at which oxygen diffuses to the reactants. In refuse, however, objects that are oversized or that have a very high moisture content, e.g., telephone books or melons, will react at a much slower rate than predicted from the rate of oxygen supply. For the watermelon case, the water will vaporize at a rate determined by the heat transfer rate to the watermelon until the moisture content is significantly reduced. For the telephone books, long times are needed for the pyrolysis and ignition waves to propagate to the center. Either case will lead to combustible evolution in zones where most other elements are completely reacted. The prediction of the occurence of these isolated sources of combustibles requires a statistical analysis of the distribution of such items in refuse. The design of the overfire air system should make allowance for the presence of these items.

E. PRACTICAL IMPLICATIONS

The model developed in the preceding section, although admittedly approximate, has the capabilities of generating much information of practical significance.

The first major conclusion is that the refuse bed behaves as a gasifier and that therefore air must be provided above the grate, in amounts that will depend on fuel conposition, undergrate air flow and the rate of heat loss. Most of the combustibles above the bed are expected to be CO and H_2 but some tars, soot and CH_4 will also be present.

The analysis further shows how the underfire air requirements increase with increasing moisture content, a characteristic well known to incinerator operators. (It should be here emphasized that the increased air rate is to support an increased energy release rate to vaporize water and not to act as a moisture carrier.) The amount of air required above the bed correspondingly decreases with increases in moisture content. The pronounced effect of cooling on the rate of gasification has significance in the start-up period when refractory walls are cold or for waterwall units.

Although some of the above conclusions were known to incinerator operators, they had not previously been known quantitatively. It is felt that it will now be possible to anticipate the effect of changes in both refuse composition and incinerator design on the underfire and overfire air requirements.

This information and the aerodynamic models described in Chapter III provide the basis for the design and regulation of incinerator air-supply systems.

F. SUMMARY

One of the most important characteristics of continuous-feed incineration systems is the combustion process occurring in the refuse bed. This region is the source of combustible pollutants. A description of its behavior as a function of operating conditions and refuse composition thus provides the designer with a powerful tool in the selection and placement of overfire air jets. To provide this tool, an overall pyrolysis-combustion model has been developed which avoids consideration of the diversity in refuse shape and composition by using data which suggests approximate equilibrium is realized in the off-gases for the reaction:

$$CO + H_2O \rightleftarrows CO_2 + H_2.$$

Using this assumption and energy and material balances, a set of simultaneous equations are developed which allow calculation of the off-gas composition as a function of refuse composition and undergrate air flow. Perturbation of the independent variables gives the designer perspective as to the range of gas compositions, velocities and temperatures which can be expected. Also, information on expected refuse burnout and heat release distributions can be developed.

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CHAPTER III

FURNACE FLUID FLOW

CHAPTER III

FURNACE FLUID FLOW

A. INTRODUCTION

The flow of fluid in an incinerator furnace may be thought to originate in the vicinity of the grates. Typically, underfire air is blown up through the grates by fans, passes through the refuse bed and is heated as it engages in the pyrolysis and combustion processes occurring within the bed. The gas mixture passes upward and into the volume above the grates where, in many cases, it is further mixed with air injected over the bed, i.e., overfire air. The hot gases pass into the main volume of the furnace where further mixing and combustion occur. Finally, the gases pass out of the furnace and into downstream units where, in some installations, heat is extracted by an array of boiler tubes, or the gas may simply be quenched and cleaned by a scrubber or other gas cleanup equipment before passing into the stack.

The still-warm gas in the stack, being lighter than ambient air, creates a draft that supplies some of the motive force for moving gases through the furnace. The pressure in the furnace is normally maintained at somewhat below atmospheric so that flow through cracks and openings in the walls will be air leakage into the furance rather than hot gas leakage out of the furnace. In many incinerators, such air inleakage represents a considerable portion of the total air flowing out through the stack. For the furnace to operate below atmospheric pressure, either stack draft or induced draft fans are needed. to provide the motive force for exhausting the gases. Thus, the amount of underfire or overfire air that can be blown into the furnace by fans, while still maintaining the furnace pressure below ambient pressure, can be limited by the exhaust system.

In the following sections, we will focus our attention on the flow from the burning refuse bed through the high-temperature volume of the furnace where, ideally, combustion should be completed. The fluid flow in this region produces mixing and combustion that may have a strong influence on the level of combustible pollutants produced by the incinerator. The nature of the flow in the furnace will be determined by the interaction of pressure, gravity, viscous and inertia forces on the fluid. Combustion above the bed will tend to raise the gas temperature, lower its density, and increase the volume flow. It will also create significant density gradients which, through buoyant effects, can exert considerable influence on the flow fluid.

The flow field can be roughly described by the spacial distributions of velocity (u), temperature (T), density (ρ), and viscosity (μ), and by one or more characteristic dimensions of the furnace [e.g., diameter (D), height(Z), or length (L)]. Local gas temperatures range from outside air temperatures to 3500°F; velocities from 1 to 40 feet per second; and characteristic dimensions from 2 to 50 feet. The relative importance of the various forces in affecting the flow field is indicated by the usual non-dimensional parameters. The Reynold's number, given by

$$N_R = \frac{\rho u L}{\mu}$$

is a measure of the ratio of inertia to viscous forces in the fluid. The Froude number, given by

$$N_{Fr} = \frac{u^2}{gZ}$$

(where "g" is the acceleration due to gravity) is a measure of inertia to gravity forces in the fluid. The Grashof number, given by

$$N_{Gr} = \frac{L^3 \rho^2 g \delta \Delta T}{\mu^2}$$

where δ is the temperature coefficient of expansion) is a combined measure of the ratio of buoyant to viscous forces times the ratio of inertia to viscous forces. Considering the ranges of velocities, temperature and characteristic lengths noted above and assuming that the properties of the gases are similar to those of air, we can expect the Reynold's number will be in the range 10^4 to 10^5, the Froude number in the range 0.1 to 3 and the Grashof number in the range 10^9 to 10^{11}. Clearly, the gross flow field will be turbulent, and turbulent mixing will dominate the viscous effects. The effects of gravity, i.e., buoyant forces, will also be important in determining certain aspects of the flow field.

The fluid flow in the furnace, accompanied by combustion, is an extremely complex process. Analytical determination of the detailed nature of the flow is beyond the present state of the art. Moreover, only very limited data on furnace flow are available. Therefore, we have directed our efforts to obtaining a rough description of the flow field, including such information as levels of velocities, approximate temperature distribution, approximate streamline patterns and estimates of degree of mixing between fuel-rich and air-rich portions of the flow before the gases exit from the furnace. The concepts developed provide some certain insights about operation and performance.

B. PREVIOUS WORK

As noted above, the furnace flow is turbulent, and buoyant effects are important. A considerable amount of background has been developed in flows where buoyant forces alone cause fluid motion, i.e., in free-convection flows.[1,2,3] Of necessity, most analytical work on free-convection deals with laminar flow. Various configurations have been analyzed, and in all but the very simplest of cases, numerical solution of the differential equations via high-speed comuuter is required. Turbulent flows are less amenable to rigorous analysis because the relationship between shear stresses in the fluid and other flow parameters can only be approximated. Prandtl's mixing-length theory for turbulent flows is perhaps the most widely used of the approximations.[4] It has been applied with considerable success to flows in which a characteristic dimension that represents the scale of turbulence can be readily identified, such as the flow of a turbulent jet, where the width of the jet is a good measure of the scale of turbulent eddies. It is doubtful that the flow in a furnace can be characterized throughout by a single characteristic dimension for turbulence; nor can the distribution of eddy size throughout the furnace flow be readily determined. The presence of density gradients, buoyant effects, convective heat transfer and combustion complicate the picture further.

It is perhaps not surprising that our literature survey (Chapter VII) has not revealed any attempts to make a detailed analysis of the turbulent flow in furnaces. Most of the work on turbulent free-convection has been experimental in nature and directed toward obtaining useful correlations for heat and mass transfer for relatively simple geometries, i.e., flow over a flat plate, around cylinders, between flat plates, etc.[1,2] Most analytical treatments of furnace dynamics[5,6] have been limited to open-hearth furnaces or comparable systems where the characteristics of the flow are dominated by the action of jets, i.e., pressure, inertia and viscous forces are controlling. Most physical modeling studies are also carried out with flows that are primarily governed by these forces. Almost none of these studies have dealt with the effects of buoyancy or combustion heat release on the flow.

The long-time interest in gas flow in furnaces had led to a number of highly simplified treatments. In a very old work, Groume-Grjimailo[7] likened the buoyant flow of hot gases in furnaces to an inverted flow of water in open channels. He presented design equations for configurating reverberatory furnaces that were derived from analysis of the flow of water over weirs. Harris[8] investigated the validity of Groume-Grjimailo's equation experimentally in a small apparatus, and found that the form of the equation was correct but that the constants were somewhat in error. These old works clearly demonstrate the importance of buoyant effects in furnace flows and draw attention to the commonly observed phenomenon of the hotter gases rising to the upper section of the furnace and flowing along the upper surfaces.

Experimental measurements of velocity distribution at the furnace outlets have been made for two incinerators as part of an evaluation of their emission characteristics.[9] The measurements in one of the incinerators (at Newton, Massachusetts) show evidence that even near the furnace outlet, hotter gases tend to flow at higher velocities in the upper sections of the furnace.

Gas velocity measurements at several sections in a grate-rotary kiln incinerator [10] show that gas velocities are not everywhere uniform. Near the furnace outlet, velocities in the upper part of the duct are nearly three times those near the bottom.

C. ANALYSIS OF FURNACE FLOW

1. OVERFIRE REGION

We begin by considering the flow in a continuous feed incinerator furnace. It is clear that the hottest gases are produced near the input end of the burning grate, while the gases at the discharge end are cooler because of the higher percent excess air. If, as is frequently the case, the flow of overfire gases is in the same direction as the fuel bed, the gases will stratify with the hot, fuel-rich components, forming a zone near the roof of the furnace. The hot gases will be accelerated as they flow upward in the pressure field produced by the denser, cool gases. The nature of the flow field out through the furnace will depend on the furnace configuration. Figure III-1 shows one common configuration. There, the flow will be turned as it approaches the roof, and will enter the breeching area of the furnace with a predominantly horizontal direction. In some incinerators, the flow from the furnace is directed upward through an array of boiler tubes.

Because of the relative uniformity of refuse and air distribution across the grate and the fact that often the width of the furnace is relatively small compared to vertical and horizontal dimensions, it seems reasonable to consider the flow field as being two-dimensional. Referring again to Figure III-1, we will imagine that, in the region above the bed where vertical flow and turning into the horizontal direction occurs, buoyant effects are most important and that viscous and heat transfer effects are secondary. Considering this flow to be that of an inviscid, non-conducting fluid without eddies, in which fluid mixing and heat transfer effects are not present, should yield useful information on its behavior. For a flow of this nature, a Bernoulli-type equation is applicable along each streamline, i.e., the total head, H, is constant along a streamline. H is defined by:

$$H = \frac{Pg_c}{\rho_c} + \frac{u^2}{2g} + z \qquad \text{(III-1)}$$

where:

P = pressure

ρ = gas density

g = acceleration of gravity

u = gas velocity

z = elevation above some reference plane

H may vary from streamline to streamline. Now, referring to Figure III-1, we will make the further simplications that the flow originates at a surface above the refuse bed, Section a, and is divided into two distinct zones with fluids of different densities, Zone 1 and

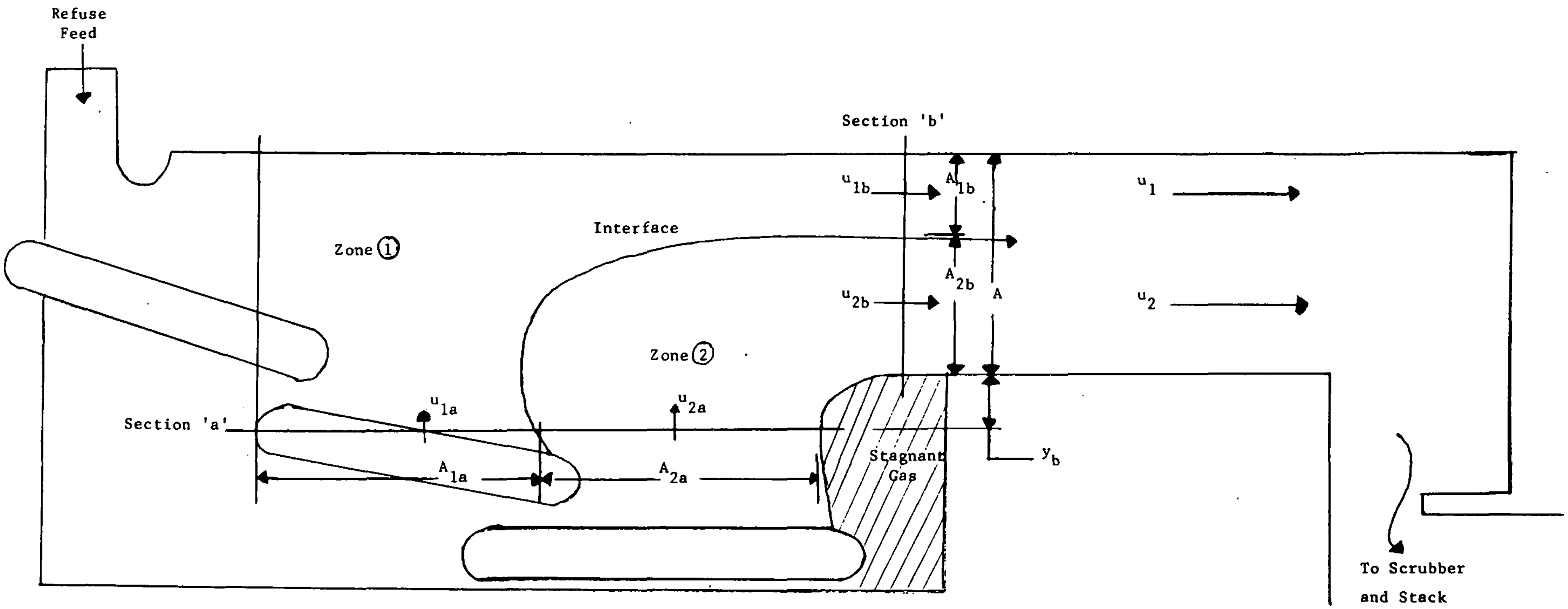

FIGURE III-1 FURNACE FLOW MODEL FOR ANALYSIS

Zone 2. If the pressure is constant across Section a, and the mass flow per unit area is uniformly distributed over Section a in each zone, the total head, H, for all streamlines in each zone is the same, and known, and one can readily compute velocities at certain points in the flow field. For example, at Section b, where the streamlines are essentially parallel, the pressure gradient perpendicular to the streamlines will be due only to the hydrostatic pressure gradient in the fluid, and the velocity in each zone will be constant, but different in the two zones. Therefore, calculation of conditions at Section b given those of Section a is straightforward. The equations relating the conditions at Section b to those at Section a, based on the above assumptions, are developed in Appendix A. Given the width and velocity at Section a for each zone, and the density of the fluid in each zone, the widths and velocities for each zone at Section b can be calculated. The approximate velocities in the region between Section a and Section b can then be inferred from a general knowledge of this type of curved potential flow field. Case studies based on this analysis are described subsequently.

The analytical model described above can also be applied to incinerators where the flow out of the furnace is directed upward or at angle to the horizontal. If a horizontal plane through the vertical flow channel, where the streamlines were more or less parallel, were defined as Section b, the equations of Appendix A could be used to calculate conditions there, given those just over the refuse bed, at Section a. The magnitude of velocities in the flow field between the sections could then be inferred.

2. INSTABILITY OF HORIZONTAL FLOW

In an incinerator like that shown in Figure III-1, the flow from Section b to the furnace outlet is essentially horizontal with warm, lower-density gases tending to remain at the top and to flow with higher velocity as a result of acceleration in the vertical flow from the refuse bed. It is desirable that, in this part of the furnace, turbulent fluid mixing will enable completion of combustion. Hence, it is of interest to determine whether the stable density distribution will significantly inhibit turbulent mixing of the hot, fuel-rich gas in the upper regions with the cooler, air-rich gases below. An analysis of the stability of this type of flow is presented in Appendix B. It is concluded that, for common incinerator configurations, turbulent mixing will not be suppressed by the thermal stratification of the gases. Hence, the horizontal flow downstream from Section b will be dominated by turbulent mixing and velocity, temperature or composition gradients at Section b will tend to be dissipated.

3. TURBULENT MIXING IN CHANNEL FLOW

In an incinerator configuration like that shown in Figure III-1, the flow out of the furnace, i.e., downstream of Section b, can be likened to two-dimensional, turbulent flow in a duct of constant width. As shown in the previous section, thermal stratification does not suppress turbulent mixing. A turbulent shear region characterized by a velocity gradient between the higher velocity, hot gas near the top of the duct and the low velocity, cooler gas below will develop.

When the shear layer is narrow compared to the total height of the flow channel, the exchange of mass and momentum effected by the turbulence will act to increase the width of the shear layer, but leave the bulk of the flow unaffected. Once the shear layer has spread to include most of the flow, the effect of turbulent exchange is to reduce the difference in velocities on either side. In a configuration such as Figure III-1, conditions at Section b are probably best approximted by considering that the shear layer includes the entire flow.

While there may be gradients of other variables in the flow, i.e., temperature and chemical composition, it is the gradient of velocity which drives the turbulence and, thereby, controls the rate of change of all other variables. Therefore, the appropriate calculation procedure is to deal with the velocity field first, and then calculate the effect on other variables in proportion to the velocity exchange.

The decay of a velocity differential in a channel filled with a shear layer can be related to the half-angle of growth of an unconfined shear layer by the assumption that the momentum exchange across the shear layer is the same in both cases--only its manifestation is different. It is shown in Appendix C that this assumption leads to the result:

$$\frac{1}{(u_1 - u_2)} \frac{d(u_1 - u_2)}{dx} = \frac{-3\xi}{2A} \qquad \text{(III-2)}$$

where ξ is the half-angle of spread in the unconfined case. The half-angle of spread can be related to the half-angle of spread of the edge of a plane jet into quiescent fluid, ξ_o, by a Galilean transformation. Thus,

$$\xi = \xi_o \frac{u_1 - u_2}{u_1 + u_2} \qquad \text{(III-3)}$$

where ξ_o is a constant equal to about .049.

Substituting this result into (III-2) gives

$$\frac{d(u_1 - u_2)}{dx} = - \frac{3\xi_o}{2A} \cdot \frac{(u_1 - u_2)^2}{(u_1 + u_2)} \qquad \text{(III-4)}$$

Note that the rate of decrease of the shear velocity $(u_1 - u_2)$ with distance, is proportional to the square of the shear velocity, which is to be expected since the turbulent shear stress is proportional to the square of the shear velocity. This equation is easily integrated, assuming $u_1 + u_2$ to be approximately constant, yielding the result

$$\frac{u_1 - u_2}{u_1 + u_2} = \left[\frac{3\xi_o x}{2A} + E \right]^{-1} \qquad \text{(III-5)}$$

Estimates of the effects of mixing for typical cases, based on Equation (III-5), are presented in Section D.

Two other effects which have been neglected in the discussion above arise from the turbulence that would exist in the absence of internal shear--natural channel turbulence. One effect of this turbulence is to retard the high-velocity portion of the stream more than the low-velocity portion by surface shear against the channel walls. The other is the internal momentum transfer due to this turbulence. A comparison of the shear stress associated with momentum exchange depicted in Figure C-1(b) to the shear stress at the wall of pipe shows that the two are equal when

$$\frac{u_1 - u_2}{u_1 + u_2} = \sqrt{\frac{f}{2\xi_o}} \simeq 0.22 \qquad \text{(III-6)}$$

where f is the wall friction factor.

There is another consideration of importance in evaluating the effectiveness of turbulent mixing. The scale of the dominant turbulence is related to the width of the shear region, which in the present case is the entire channel dimension. The reduction of shear velocity, and the associated reductions of temperature and composition differentials, is due primarily to mixing of fuel-rich and fuel-lean constituents on the gross scale of the dominant turbulence. That is, at any station in the flow field, a time sequence of samples will show gas compositions alternating between rich and lean and passing by in large, relatively distinct chunks. The completion of chemical reactions requires mixing on a scale comparable to the diffusive processes which ultimately control combustion, and times and flow lengths much longer than are realized in existing incinerators. Thus, even though a significant amount of gross mixing may be predicted by the analysis outlined here, the effect on gas composition (burnedness) may be much less within the flow length available. The most effective means of promoting combustion is to generate high turbulence levels at a very small scale.

D. CASE STUDIES

1. OVERFIRE REGION

Using the analytical model described in Section C-1, we have analyzed a number of cases for various conditions of incinerator operation. The bases for the cases analyzed are summarized in Table III-1. In addition, the following assumptions are made in all cases:

a. The furnace configuration is that shown in Figure III-1, with y_b = 3.7 feet and A = 12 feet.

b. Both the hot and cool gases have a molecular weight of 29.

c. The air input under the grates is uniformly distributed along the length of the grates.

d. The average refuse burning rate per unit of grate area equals 60 $lb/hr\text{-}ft^2$.

e. The absolute pressure level is approximately 1 atmosphere everywhere.

f. The majority of refuse gasification (mass addition to the undergrate air flow) takes place in Zone 1.

Other assumptions implicit in the 2-zone flow model have been described in Section C-1.

Cases A-1 and A-2 differ in that the refuse is moist in A-1 and dry in A-2. Case A-3 is similar to A-2, except that air bypass through voids in the refuse bed (channeling) is assumed in A-3, in an amount equal to that taking part in the gasification reactions.

Cases A-4, A-5, and A-6 are identical to A-1, A-2, and A-3, respectively, except that the hot gas temperatures above the gasification section of the bed differ. In Cases A-1 through A-3, this temperature is assumed to be 2000°F, a value thought to be representative of average conditions across the gasifier section. In Cases A-4 through A-6, hot gas temperatures corresponding to complete combustion are assumed. The latter represent an upper limit that might be achieved well above the bed where mixing with additional air might enable completion of combustion. Thus, an actual case where further combustion occurs well above the bed, but not to completion, might lie somewhere between Cases A-1 and A-4, for example.

Cases B-1, B-2, and B-3 have only 125 percent of stoichiometric air flow, but are otherwise similar to Cases A-1, A-2, and A-3, respectively, which have 150 percent of stoichiometric air flow.

TABLE III-1

BASES FOR CASES ANALYZED

Case	Refuse* Characteristic	Total Airflow**	Hot Zone Airflow		Hot Zone Total Gasflow, lb/hr	Cool Zone Total Gasflow, lb/hr	Hot Gas Temperature for Complete Combustion, °F	Assumed Hot Gas Temp. °F	Assumed Cool Gas Temp. °F
			Air input to Gasification Reaction in bed, lb/hr	Air By-Pass through Voids in Refuse (Channeling), lb/hr					
A-1	Moist	150	37,515	0	66,420	54,735	2,860	2,000	1,000
A-2	Dry	150	21,525	0	39,975	70,725	3,400	2,000	1,000
A-3	Dry	150	21,525	21,525	61,500	49,200	3,400	2,000	1,000
A-4	Moist	150	37,515	0	66,420	54,735	2,860	2,860	1,000
A-5	Dry	150	21,525	0	39,975	70,725	3,400	3,400	1,000
A-6	Dry	150	21,525	21,525	61,500	49,200	3,400	3,400	1,000
B-1	Moist	125	37,515	0	66,420	39,360	2,860	2,000	1,000
B-2	Dry	125	21,525	0	39,975	55,350	3,400	2,000	1,000
B-3	Dry	125	21,525	21,525	61,500	33,825	3,400	2,000	1,000
B-4	Moist	125	37,515	0	66,420	39,360	2,860	2,000	1,500
B-5	Dry	125	21,525	0	39,975	55,350	3,400	2,000	1,500
B-6	Dry	125	21,525	21,525	61,500	33,825	3,400	2,000	1,500

*"Moist" is equivalent to a moisture content of 35 percent by weight and "dry" to 0 percent moisture.

**Airflow quantities are expressed as percentages of equivalent stochiometric air flow.

Cases B-4, B-5, and B-6 are identical to B-1, B-2, and B-3, respectively, except that the cool gas temperatures differ. The higher cool gas temperature of 1500°F for Cases B-4 through B-6 will reduce the density difference between hot and cool gases. Hence, these cases will show reduced buoyant effects.

The results of calculations for the various cases are summarized in Table III-2.

The gas in both Zones 1 and 2 is accelerated between Sections a and b. The average velocity must increase because the flow area decreases. However, the hot gas in Zone 1 experiences a larger velocity increase due to its buoyancy relative to the cooler, denser gas in Zone 2. In effect, a vertical pressure gradient due to the gravity field, and equal to $(\rho_1 - \rho_2)g$, is imposed on the gas in Zone 1, in addition to any gradient associated with flow area change. It produces the greater acceleration in that zone.

As would be expected, the velocity increase of the hot gas is most pronounced in Cases A-4 through A-6 where the temperature difference between hot and cool gases and, hence, the density difference, is largest. The velocity increase of the hot gas is least in Cases B-4 through B-6 where the temperature difference is smallest. Cases A-2, A-5, B-2, and B-5 represent conditions with dry refuse and no air bypass in the gasification section. The air requirements for gasification are least with dry refuse, and with no bypass the flow rate of hot gas is minimal. In these cases, the hot gas at Section B occupies only about one-quarter of the height of the duct. Case A-5, in which the temperature of the hot gas is that corresponding to complete combustion, produces the highest hot gas velocity at Section B of any of the cases. This case represents an extreme in terms of the hot gas being confined to a narrow, high-velocity zone near the roof of the furnace. Cases B-4 and B-6 indicate the opposite situation, i.e., the velocity difference between hot and cool gases is minimal and the fraction of the flow area occupied by hot and cold gases is nearly the same. A vertical velocity profile in the breeching area corresponding to these cases would indicate minimum velocity gradients. The latter conditions seem to agree best with the limited observations that have been made in the incinerator at Newton, Massachusetts. Finally, the differences in conditions at Section b brought about by the reduction of air flow from 150 percent of stoichiometric to 125 percent, i.e., comparing Cases A-1 through A-3 to B-1 through B-3, do not appear significant.

Table III-2 indicates conditions only at Sections a and b. The velocities near the lower edge of the hot zone will range between the values at Sections a and b, generally increasing as the flow proceeds to Section b. A pressure gradient perpendicular to the stream lines will be associated with their curvature, i.e., the turning of the flow. The pressure in the upper left-hand region of Zone 1 will be somewhat higher than near the interface between Zones 1 and 2. Hence, the velocities in the upper left-hand region will be lower.

TABLE III-2

RESULTS OF CALCULATIONS*

Case	A_{1a} ft	A_{2a} ft	u_{1a} ft/sec	u_{2a} ft/sec	A_{1b} ft	A_{2b} ft	u_{1b} ft/sec	u_{2b} ft/sec
A-1	16.3	23.7	8.8	3.0	5.38	6.61	26.5	10.5
A-2	9.3	30.7	9.2	2.9	3.04	8.95	28.2	10.0
A-3	18.7	21.3	7.1	3.0	5.22	6.77	25.3	9.3
A-4	16.3	23.7	11.8	3.0	5.52	6.47	34.9	10.8
A-5	9.3	30.7	14.5	2.9	3.19	8.80	42.2	10.2
A-6	18.7	21.3	11.1	3.0	5.49	6.50	37.7	9.7
B-1	19.5	20.5	7.3	2.5	5.88	6.11	24.2	8.2
B-2	11.2	28.8	7.7	2.5	3.23	8.76	26.6	8.1
B-3	22.4	17.6	5.9	2.5	5.67	6.33	23.3	6.8
B-4	19.5	20.5	7.3	3.3	7.04	4.96	20.3	13.6
B-5	11.2	28.8	7.7	3.3	4.20	7.80	20.4	12.2
B-6	22.4	17.6	5.9	3.3	7.17	4.83	18.4	12.0

*The physical significance of the noted quantities (A_{1a}, u_{1b}, etc.) may be seen by reference to Figure III-1.

Since we do not have an accurate picture of the actual flow in an incinerator furnace, we can only speculate on the differences between an actual flow and the highly simplified model described above. To begin with, air flow may not be introduced uniformly along the bed. Operation with lower pressures in the wind boxes under the grates near the discharge end is not uncommon. Overfire air may also be introduced at various places over the bed. The air flow just above the refuse bed will not be sharply divided into two zones. Instead, continuous velocity and temperature distributions will exist as a result of the combustion and local mixing processes. As the warmer gas flows upward and is accelerated, mixing between it and the cooler gases would induce momentum exchange, tending to slow down the hot gas and accerate the cool gas in the mixing region, and further smooth out any sharp velocity gradients. The cool gas tends to flow at much lower velocities, particularly when air flow is reduced toward the end of the bed. The momentum exchange in the mixing zone adjacent to the hotter gases will tend to produce higher, cool gas velocities in that region and lower velocities in the region away from the hot gas. In effect, a kind of circulation can be induced in the cool gas. Under certain conditions, it might even be possible to have cool gas flowing downward in the region farthest away from the hot gas. By the time the flow reaches Section b, we would expect a smoothed out velocity distribution of the Gaussian-type, with higher velocities at the top and lower nearer the bottom. The maximum and minimum velocities are expected to be of the magnitudes indicated in Table III-2, depending on the operating conditions.

The discussion thusfar has dealt with furnace configurations like that shown in Figure III-1. As previously noted, the analysis of Section C-1 could also be applied to different configurations, such as to those where flow out of the overfire region were directed upward or at at angle to the horizontal. If, in such configurations, the vertical distance from just above the refuse bed to a section across the outflow duct is larger than the height $(y_b + A_{2b})$ in Figure III-1, the gravity pressure gradient acting on the hot gas $(\rho_2 - \rho_1)g$, could produce larger velocity increases than for the cases of Tables III-1 and III-2. Thus, the tendency for hot gases to flow in a narrow, high-velocity zone in the duct would be more pronounced. Application of the equations of Appendix A to such configurations could provide estimates of gas velocities and zone widths like those in Table III-2.

2. CHANNEL FLOW REGION

The flow from Section b to the furnace outlet will be modified by turbulent mixing. Velocity, temperature and composition gradients will tend to be reduced. If we assume that the shear layer at Section b occupies the entire duct, Equation (III-5) can be used to estimate the reduction in velocity differential. Taking $x = 0$ at Section b, and using the representative velocities (per Table III-2) of

$$u_{1b} = 30 \text{ FPS}$$

$$u_{2b} = 10 \text{ FPS}$$

we find that

$$\frac{u_{1b} - u_{2b}}{u_{1b} + u_{2b}} = 0.5$$

and from Equation (III-5), C = 2. Assuming that the flow length to the furnace outlet is approximately three times the height of the breeching ($x \simeq 3A$), we find that at the outlet:

$$\frac{3\alpha_o x}{2A} = \frac{9}{2}\alpha_o = .221$$

and

$$\frac{u_1 - u_2}{u_1 + u_2} = [2.221]^{-1} = 0.45$$

Thus, the differential velocity ($u_1 - u_2$) would be reduced by $(\frac{0.5 - .45}{0.5})$ x 100 = 10%. To achieve a 50 percent reduction would require $\frac{3\alpha_o x}{2A} = 2$, or $x \simeq 30A$. The latter estimate is conservative because as the shear velocity decays, natural channel turbulence becomes relatively more important and will augment the decay.

A 10 percent decrease in differential velocity coressponds to only a 5 percent change in both u_1 and u_2, i.e., u_1 would decrease by 5 percent and u_2 increase by 5 percent. Exchanges of energy (or temperature) and composition between the high and low velocity regions would be of the same magnitude. Thus, it appears that for conditions represented by the ranges of parameters in Tables III-1 and III-2, a rather limited degree of mixing can be realized downstream of Section b.

The percentages of air addition to the fuel-rich gases required to enable completion of combustion for the cases of Tables III-1 and III-2 are shown in Table III-3 and are based on the results of the bed analysis presented in Chapter II. Comparison of these values to the magnitude of composition mixing indicated by the example above suggests that mixing of the fuel-rich and air-rich portions of the gas flow downstream of Section b will not be effective in completing combustion.

TABLE III-3

PERCENT AIR ADDITION TO FUEL-RICH HOT GAS FOR COMPLETE COMBUSTION

Cases	Volume Percent Air Addition
A-1, A-4, B-1, B-4	36
A-2, A-5, B-2, B-5	100
A-3, A-6, B-3, B-6	30

As noted in Section C-3, mixing on a scale comparable to the diffusion processes that limit the combustion rate is required to assure burnout of combustible pollutants. Hence, the gross mixing indicated by the discussions above must be considered as an upper limit on the changes in chemical composition on a scale appropriate to chemical reactions.

The dependence of mixing on the shear velocity ratio $\left(\frac{u_1 - u_2}{u_1 + u_2}\right)$ and the observations above suggest that ideal mixing, in terms of combustion reactions, might be accomplished by small-diameter, high-velocity jets flowing in the opposite direction to the main stream, i.e., with negative u_2 either in the furnace or in the breeching. The effect would be to maximize shear velocity, minimize the average velocity of the shear layer which carries the fluid downstream, and to produce turbulence at the appropriate scale. Though perhaps difficult to achieve in practice, the concept may serve as a useful goal to approach.

E. APPLICATION TO DESIGN AND PERFORMANCE EVALUATION

The preceding sections have outlined techniques and reasoning processes for establishing the general features of the flow field in the furnace. Using the approaches discussed, the approximate levels of velocities for the hotter and cooler gases can be estimated at a cross-section where the flow leaves the overfire region and the flow streamlines are essentially parallel. The width of the hot gas and cool gas zones can also be estimated and used to establish a rough velocity profile across this section. Velocities between the section just above the refuse bed and the outlet from the overfire region can then be roughly inferred. To make these estimates, knowledge of the gas flow from the refuse bed is required. Such information can be developed from the estimates of the undergrate air flow distributions and the concepts described in Chapter II.

An approximate description of the general flow field is useful in defining the flow environment for the injection of air or stream jets to promote mixing in the furnace. Knowledge of the flow field can suggest locations of the jets, the direction in which they should point and their velocity level. It can also provide an indication of where the hottest gases will flow, and the maximum velocities they can achieve. Such information can be helpful in arriving at furnace configurations that avoid the deleterious effects that might be caused by hot gases impinging on certain surfaces (such as boiler tubes or bridgewalls) with relatively high velocities.

The need for the addition of air to the hot, fuel-rich gases is evident from the discussion of the bed combustion process in Chapter II. Even if the total underfire air flow were greater than 100 percent of stoichiometric for complete combustion, the tendency for thermal stratification of gases in the overfire region and for limited mixing between hot, fuel-rich gases and the cooler, air-rich gases below, as the flow proceeds out of the furnace, indicates the need for auxiliary means of mixing the air-rich and fuel-rich gases. The addition of overfire air to foster complete combustion can also be effective, provided it is introduced in a way that promotes turbulent mixing with the fuel-rich gases on the relatively small-scale that promotes chemical reactions. The potential effectiveness of jets flowing counter to the hot gas has been noted. In short, our findings about the refuse bed combustion process and the general nature of flow in the furnace provides a rationale and certain criteria for the use of overfire air and/or steam jets to promote turbulent mixing in the furnace.

The fact that turbulence with small characteristic dimensions is required to effectively complete combustion also has a bearing on the design of baffles, grids or other passive, stationary mixing devices for furnaces. Large baffles with dimensions of the same magnitude as the height of the flow passage can redirect the gross flow and produce turbulent mixing as a result of flow separations and eddies downstream. However, this turbulence will be on the scale of the baffle dimension and, in effect, will at best only promote gross mixing. Though such mixing is useful, it is but one step in the process leading to complete combustion.

For the remainder of the process, the use of baffles with characteristic dimensions measured in inches, or even fractions of an inch, would appear to be superior to those with dimensions measured in feet. One can speculate that, for a furnace which is ideal in terms of achieving complete combustion, gross mixing should be accomplished early in the flow process, i.e., in the overfire region, by the use of air or steam jets, or stationary mixing devices. Further mixing on a small scale should then be promoted by stationary mixing devices with small dimensions, at, say, the inlet to the breeching area, so that completion of reactions can be accomplished before the flow exits from the furnace. In practice, attention must be given to the problems of draft loss and plugging with such closely spaced baffles. Experience with convective boiler surfaces, however, which provide such a mixing action, suggests that these problems are not insurmountable.

Finally, calculations performed for various operating conditions such as those of Table III-1, or for other appropriate conditions, can indicate how operating conditions will affect flow conditions and turbulent mixing in the furnace.

F. SUMMARY

The combustible pollutants arising from the burning refuse bed flow through the furnace and flues in a non-uniform manner. Temperature, gas composition and velocity in the flow are different as a consequence of the geometry of the system, the spatial distribution of their entry points and the inadequacy of self-mixing. Understanding the flow field and the effects of design and operating variables on the flow can assist in the furnace enclosure design; indicate roughly the degree of self-mixing which occurs; support the design of baffles and other flow-control surfaces; importantly affect decisions on the placement and size of convection boiler tubes; and indicate the flow environment which interacts with overfire air or steam jets.

As for the bed processes, analysis of the flow field required simplification. The approach taken was to assume a two-dimensional flow pattern and to break the flow into several regions. In some portions of the furnace, an approximation to potential flow occurs and, after including buoyancy effects in the equations, estimates were made as to gas velocity as a function of position. In other regions of the furnace, shear layers are developed and consideration must be given to turbulent mixing. Here, concepts drawn from plane jet theory were applied to yield estimates of mixing levels and degrees.

From these results, flow patterns and velocity distributions were developed which provide the designer with new insights into system behavior. These results, both quantitatively and qualitatively, are particularly helpful in establishing the positions and flow environments of air and steam jets for combustible pollutant control.

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CHAPTER IV

OVERFIRE JETS

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OVERFIRE JETS

A. INTRODUCTION

The preceding chapters on the behavior of refuse beds and on the flow dynamics of furnace gases indicate that situations exist with incinerator furnaces where jet systems could be of assistance in realizing better burnout of combustible pollutants and in controlling furnace temperature distributions. A review of the design and operating characteristics of existing incineration systems[1] and discussions with incinerator designers suggest the need for better correlations supporting the design of these jet systems. As discussed below (Section B), the fluctuating conditions of gas movement and composition within incinerator furnaces present a considerable challenge to the detailed analysis of any device which interacts with the flow and combustion processes. As a consequence, our analysis was necessarily somewhat simplified. We feel, however, that our results will support the design of practical systems which will perform effectively and in accord with the expectations of the designers.

This chapter consolidates existing overfire jet design correlations. Building on existing theory and incorporating the results of original experiment and analysis carried out as part of this effort, improved design methods are then suggested for application to refuse incineration systems.

1. USE OF JETS

Jets have been utilized for many years as an integral part of furnaces, boilers and other combustion systems. In boilers fired with pulverized coal, for example, air jets are used to convey the fuel into the combustion chamber, to control the heat release patterns and to supply secondary air for complete combustion. In processes employing a burning fuel bed, properly placed air jets supply secondary air where needed above the fuel bed to complete combustion. Also, jets of air and/or steam are used to induce turbulence and to control temperature by dilution of furnace gases.

The important characteristics of jets which underlie all of these uses are:

- The controlled addition of mass to contribute to the oxidation process (air jets) or to serve as a thermal sink to maintain gas temperatures below levels where slagging, corrosion, or materials degradation may occur (air or steam jets); and

- The controlled addition of momentum to promote mixing of the jet-conveyed gas with gases in the combustion chamber or to promote mixing of gases from different parts of the combustion chamber. In the latter case, high-pressure steam jets are often used to provide high momentum fluxes with a minimum introduction of mass.

The basic challenge to the combustion system designer is to employ these characteristics to maximum advantage in meeting his overall design goals. How jet capabilities relate to the design goals of the incinerator designer is discussed in Section B of this chapter.

2. GENERAL CHARACTERISTICS OF JETS

Because of the long-standing practical interest in the use of jets, a large body of literature has been developed which quantitatively characterizes the nature of jet flow. Jets are conveniently categorized, according to flow regime (laminar or tubulent, supersonic or subsonic) and geometry (round or plane). Laminar jets occur only at very low jet velocities and are of no interest here. Supersonic jets, of interest for describing high-pressure steam flows, are of potential interest but are not considered here. Plane jets, which issue from a slot finite in one dimension and effectively infinite in the other, are primarily of academic interest. We will, therefore, focus in this discussion on round, low-subsonic, tubulent jets and return later to the fact that a row of closely-spaced round jets behaves, to a degree, like a plane jet.

Other important parameters characterizing the jet flow behavior include the relative densities of the jet and ambient fluids, the velocity of the ambient fluid relative to the jet velocity, and the degree to which the space into which the jet issues is confined by walls. Also, in situations where the combustion can occur (jets of fuel into air as in burners and jets of air into fuel vapors--the so-called "inverted flame"), the initial temperature and combustible content of the jet and ambient fluid are of interest. All of these factors are important in the application of jets to incinerators, and their effects, singly and in combination, on jet characteristics are discussed in Sections C and D. To set the stage for this discussion, we consider here the basic characteristics of jets issuing into an infinite atmosphere of quiescent fluid of the same density as the jet fluid.

The round, isothermal turbulent jet shows three characteristic regions (Figure IV-1). Immediately adjacent to the nozzle mouth is the mixing region. Fluid leaves the nozzle with an essentially flat velocity profile. The large velocity gradients between this "potential core" and the ambient fluid induce turbulence which causes ambient fluid to mix into the jet. The mixing results in momentum transfer between the jet and ambient fluids and progressively destroys the flat velocity profile. In a distance of about 4.5 jet nozzle diameters downstream, and turbulent diffusion has worked its way to the centerline of the jet and eliminated the potential core.[19]

It is important to note that the "nozzle diameter" characterizing jet flow is not necessarily the physical dimension of the orifice from which the jet issues. If, for example, the jet issues from a sheet-metal plenum, a flow contraction to about 60% of the open discharge area (the area of the vena contracta) characteristic of the flow past a sharp-edged orifice, will define the effective nozzle diameter and the location of the effective jet discharge plane will be displaced about two-thirds of a diameter downstream of the orifice (the location of the vena contracta). If a relatively long (2-3 diameters) constant area section lies upstream of the discharge plane, the nozzle diameter may be taken as the orifice diameter. Attention should be given, therefore, to the geometry of the entire nozzle fluid delivery system in determination of the flow rate of jet fluid.

In the region from 4 1/2 to about 8 diameters downstream, the transition of the flat entrance velocity profile to a fully developed profile is completed. Beyond this transition zone, the velocity profile remains more or less of constant shape relative to the velocity on the axis of the jet and is referred to as "self-preserving."

Important jet characteristics include:

a. The centerline velocity and concentration changes with axial distance from the nozzle mouth;

b. The shape of the velocity and concentration radial profiles in the fully developed region.

c. The intensity of turbulence in the jet; and

d. The rate of entrainment of ambient fluid into the jet.

These characteristics are all interrelated; turbulence generated by high velocity gradients induces entrainment which causes momentum and mass transfer between the jet and the ambient fluid.

These characteristics are important in practice because they determine the quantititive effect of firing a jet into a combustion chamber. The axial decay of velocity establishes how far the jet effectively penetrates into the chamber. The radial velocity distributions determine how large a volume is affected by the jet. The entrainment rates determine how effectively furnace gases are mixed along the jet path.

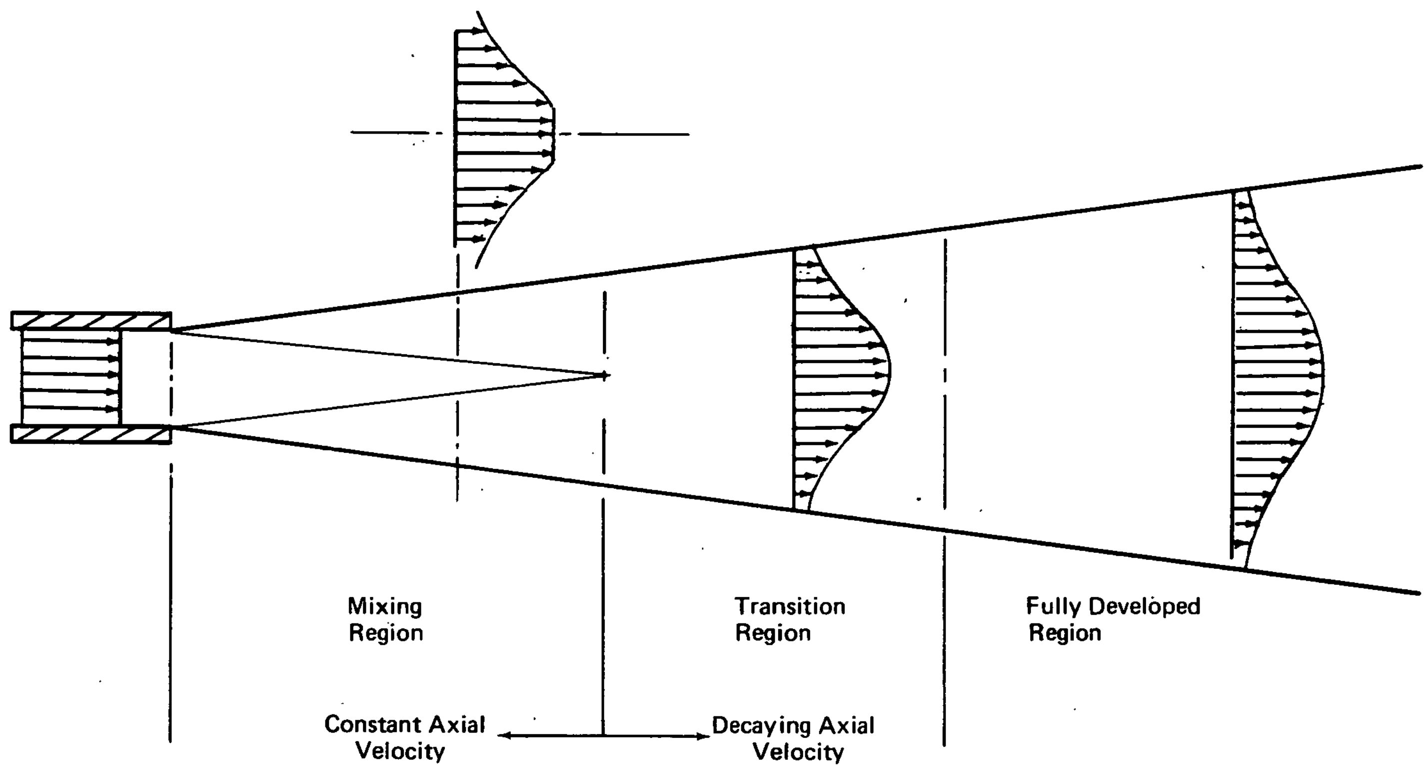


FIGURE IV-1 REGIONS IN JET FLOW

B. THE USE OF JETS FOR COMBUSTION CONTROL

As described in Chapter I, municipal incineration is an important source of combustible pollutants. The analytical developments in Chapters II and III showed that these pollutants would necessarily arise in the pyrolysis zone of the grate and could possibly arise in the discharge zone. Further, it was shown that the turbulent mixing processes naturally occurring in the flow through the furnace could be inadequate to supply the stoichiometric oxygen to the fuel-rich gases. Even when this oxygen is admixed in the latter stages of furnace flow, however, it is entirely possible that the residence time remaining may be too short for complex organic species or for thick-sectioned char particles to completely burn.

These observations lead to the conclusion that systems are needed to provide air near the pyrolysis zone and/or to induce high-intensity turbulence at strategic locations within the incinerator furnace. Although passive mixing systems, such as baffles or checkerwork, may have some value in the inducement of turbulence, they clearly are unuseful in supplying air and are not alterable to cope with changes in the distribution of combustible pollutant release along the bed and throughout the chamber as refuse composition and burning characteristics change. As a consequence, incinerators, as one of a family of grate-burning systems, have turned to the use of overfire jets of either steam or air as a low-cost and effective control technique.

1. JET DESIGN FOR INCINERATORS--A STATEMENT OF THE PROBLEM

Combustible pollutants appear to be generated along the full length of the incinerator grate, although their discharge rate into the overfire volume is relatively low in the drying and ignition zones prior to the introduction of underfire air. From the standpoint of total pounds per hour per square foot release rate, the pyrolysis zone probably qualifies as the single most important source of carbon monoxide, soot and hydrocarbons (Figure II-6 of Chapter II). Carbon monoxide and coked ash material will be evolved in the region between the pyrolysis zone and the burnout region. Also, data developed in the course of this program (see Chapter II) indicate that combustibles may be evolved in the discharge grate section. Our analyses, data and speculations indicate that overfire air is definitely required in the region of pyrolysis and char burnout; reduced undergrate air flows and turbulence inducement is required in the area over the discharge grate; and some means may be required to increase the general level of turbulence throughout the upper regions of the incinerator furnace.

The specification of jets for incinerator applications meeting the requirements listed above places great demands upon the designer. It is clear that the jet behavior should be known in a flow field where combustion ,cross-flow and buoyancy effects are all potentially important; and, for some systems, the jets must operate over long distances. This latter characteristic arises from the shape of most continuous-feed incinerators which tend to be long and narrow, thus devices acting over the discharge grate region which are expected to carry bed off-gases back towards the pyrolysis region must act over distances of 10-30 feet (20 to 100 or more jet diameters). The location, number, and flow parameters appropriate to these jets should be consistent with the overall furnace geometry, be easily maintained and operated, and should be controllable to the extent demanded by the fluctuations in refuse composition and burning characteristics. Particularly in the case of air addition, the jet design should add sufficient air to meet the oxygen requirement of the rising fuel vapors yet not provide so much air as to overly cool the gases, thus quenching combustion. Also, the draft capabilities of the furnace must be considered in determining the amounts of air introduced.

2. EXPERIENCE IN JET APPLICATION FOR COAL-BURNING SYSTEMS

Overfire air systems have been used for over ninety years in coal-burning practice. In some respects, the combustion characteristics of coal burning on a grate are similar to those of refuse. Typically, however, coal ignites more readily (partly due to its lower moisture content), burns with more regularity and predictability and, for overfeed or cross-feed situations, is typically burned in furnaces with grates which are short relative to those used in many continuous-feed refuse-burning incinerators. Therefore, although the probems are not identical, it is of value to review experience in coal-burning practice as an indication of the potential of jet systems for combustion control.

The use of controlled overfire air in industrial solid fuel combustion systems was stimulated by the desire to improve boiler efficiency through complete combustion of soot and carbon monoxide and to reduce smoke emissions. Although the historical pattern of technological development of overfire air systems is unclear, Stern[2] mentions that patents and active marketing of steam-air jets, primarily for smoke control, began in 1880. Quantitative appreciation of the benefits of smokeless combustion on overall fuel economy was widely argued until documented by Switzer[3] in 1910. Switzer's work, carried out at the University of Tennessee, involved measurements of jet system steam consumption, smoke intensity and boiler efficiency on a hand-fired return--tubular boiler fired with bituminous coal. The results of his tests showed an increase in thermal efficiency from 52.6% to 62.1%, an increase in the effective range of the boiler from 80% to 105% of its rated capacity before smoking occurred, and a steam consumption for the overfire jet system of only 4.6% of the total steam raised.

Recognition of the importance of overfire air and mixing stimulated considerable research in the first decade of this century. Some of the more completely documented and detailed laboratory and field data was produced by the Bureau of Mines who were conducting "investigations to determine how fuels belonging to or for the use of the United States Government can be utilized with greater efficiency." Kreisinger et. al.[4] studied the combustion behavior of several coals in a special research furnace under a variety of combustion air and firing rate conditions. Their work showed a strong relationship between the burnedness of the flue gas, the properties of the coal, and the size of the combustion space (Figure IV-2). Their results were interpreted in agreement with prior suggestions of Breckenridge[5], to result

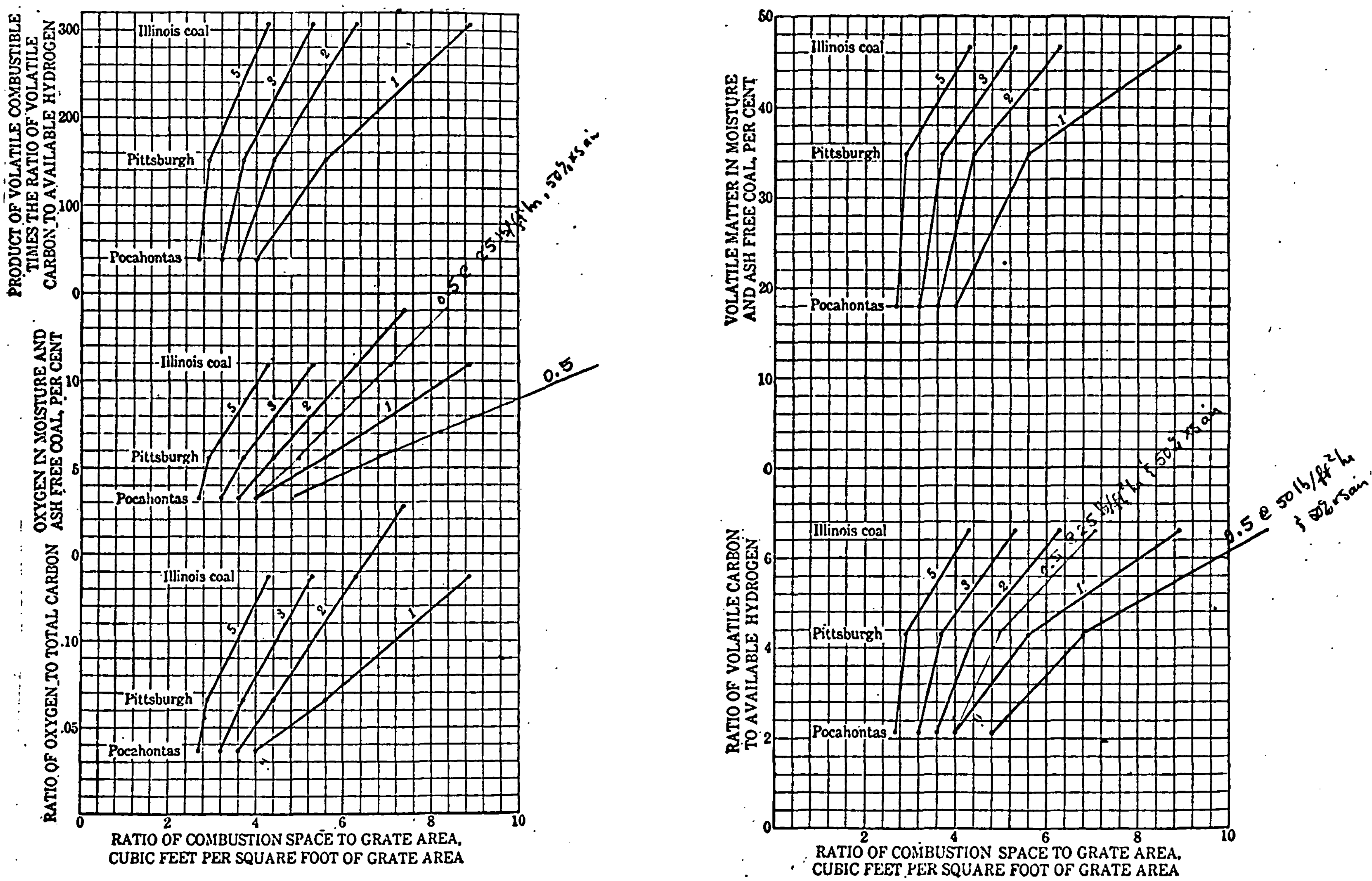


FIGURE IV–2 RELATION BETWEEN COAL CHARACTERISTICS AND THE SIZE OF COMBUSTION SPACE REQUIRED IN USBM TEST FURNACE AT COMBUSTION RATES OF 50 LBS/HR FT² AND 50% EXCESS AIR

from differences in the emission of volatiles between coal varieties and the rate-limiting effects of inadequate mixing. Their correlating parameter, which they called the "undeveloped heat of combustible gases", represented the heat of combustion of the flue gases relative to the heat of combustion of the coal burned. In view of the trends shown in Figure IV-2 and the composition data in Table IV-1, it would appear that the volume requirements for complete burnout in refuse incineration may be considerably in excess of those acceptable in coal-fired combustors. Quantitative extrapolation from their data to incinerators, however, would be highly speculative. Unfortunately, within the scope of the U.S.B.M. experimental program, generalized design guides for the flow rate and locations appropriate for overfire air systems were not developed.

Some of the earliest test work directly aimed at finding the benefits of overfire air in utility combustion systems was conducted in 1926 by Grunert[6] on forced draft chain grate stokers at the Commonwealth Edison Company in Chicago. Grunert's data showed that overfire jets discharging over the ignition zone could reduce the carbon monoxide levels at the entrance to the first pass of boiler tubes from an average value of 1% to essentially zero. Also, the gas temperature and composition profile could be made considerably more uniform. Of importance to fuel economy, it was found that although additional air was introduced through the overfire air jets, the total combustion air was susceptible to reduction. Similar work in Milwaukee, reported by Drewry[7] also showed performance improvement (an increase of 7.2% in boiler efficiency), smoke elimination, and complete burnout of combustibles within the firebox. Once again, however, design correlations generally applicable to the coal-burning industry were not presented.

Major contributions to the overfire jet design art were published in the mid-1930's. Of particular importance were reports on a number of meticulous test programs carried out in Germany; perhaps typified by the work of A. R. Mayer.[8] Although still not providing generalized design criteria, Mayer made gas composition traverses (45 points) within a traveling grate stoker furnace firing low-volatile bituminous coal. His results are shown in Figure IV-3.

FIGURE IV-3 LINES OF EQUAL HEATING VALUE (KG-CAL PER STD CU M) OF FLUE GAS FIRING LOW-VOLATILE BITUMINOUS COAL AT A FUEL RATE OF 28 LB PER SQ FT PER HOUR[8]

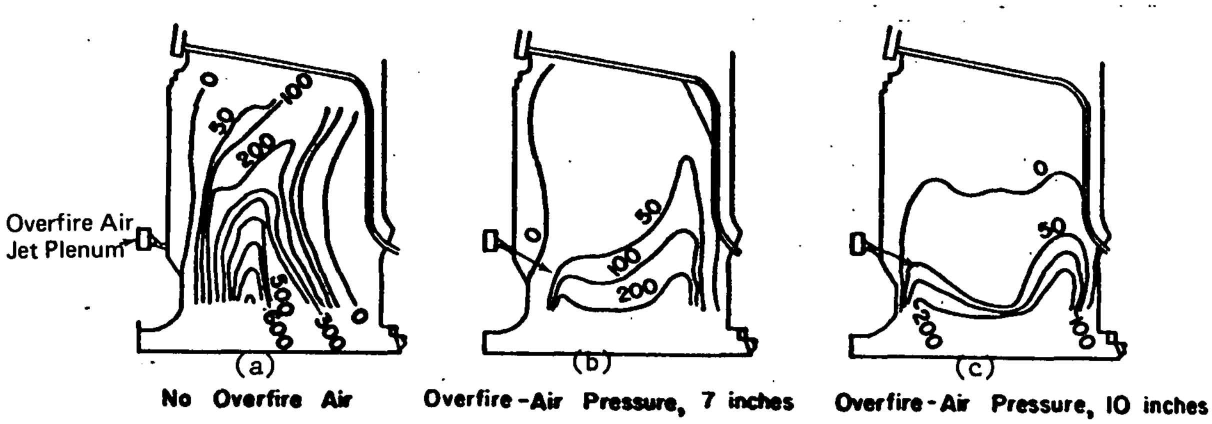


TABLE IV-1

CHEMICAL CHARACTERISTICS OF COAL[4] AND REFUSE[1]

Item	Characteristics	Pocahontas Coal	Pittsburgh Coal	Illinois Coal	Refuse
1	Volatile Matter[a]	18.05	34.77	46.52	88.02
2	Fixed Carbon[a]	81.95	65.23	53.48	11.99
3	Total Carbon[a]	90.50	85.7	79.7	50.22
4	Volatile Carbon[a] (Item 3 - Item 4)	8.55	20.47	26.22	38.22
5	Available Hydrogen[a]	3.96	4.70	3.96	1.57
6	Ratio Vol. C to Avail. H_2[a]	2.16	4.35	6.60	24.34
7	Oxygen[a]	3.32	5.59	10.93	41.60
8	Nitrogen[a]	1.19	1.73	1.70	1.27
9	Percentage of Moisture Accompanying 100 Percent of M & AF coal or refuse	2.53	2.88	22.07	55.19
10	Product of Items 1 and 6	39	151	307	2142
11	Ratio of Oxygen to Total Carbon[a]	0.0367	0.0652	0.137	0.828
12	Total Moisture in Furnace Per Pound of Coal or Refuse Reduced to M & AF Basis (Pounds)	0.409	0.501	0.700	1.161

a. Percent on moisture and ash-free basis (M & AF)

As a measure of the completeness of combustion in the overfire space, Mayer determined the heating value of the gases (Btu/cu ft) as calculated from the complete gas analysis. Without overfire air, as seen in Figure IV-3(a), strata of combustible gases rise into the combustion chamber and persist as the gases enter the first boiler pass. This is indicated by the zero heating value curve which is not closed. Figure IV-3(b) shows the effect of medium-pressure overfire air jets. It can be seen that combustion is improved, yet some fraction of the combustible still enters the boiler passes. Figure IV-3(c) shows the effect of further increases in the overfire air plenum pressure. Under these latter conditions, combustion is complete within the furnace volume. For these tests, Mayer employed jets directed towards the bed just beyond the ignition arch.

Also in the 1930's, developments in fluid mechanics by Prandtl and others provided mathematical and experimental correlation on the behavior of jets. Application of this understanding to furnace situations was presented in some detail by Davis.[9] His correlations, although based on greatly simplified assumptions, were of considerable interest to the furnace designers in that time. As an example of the applicability of his work, Davis explored the trajectories anticipated for jets discharging over coal fires and compared his calculated trajectories with data by Robey and Harlow[10] on flame shape in a furnace at various levels of overfire air. The results of this comparison are shown in Figure IV-4. Although general agreement is shown between the jet trajectory and the flame patterns, correlation of the meaning of these parameters with completeness of combustion is unclear and not supported by Robey and Harlow's data. The results do give confidence, however, that jet trajectories can be calculated under a variety of furnace conditions to produce reasonable estimates of behavior and thus permit avoidance of impingement of the jet in the bed.

Although work on the applications and advantages of overfire air continued through the war years (particularly with reference to avoidance of smoke in naval vessels to preclude some easy identification and submarine attack[2]), the wide introduction of pulverized coal-firing in electric utility boilers and the rapid encroachment of oil and gas into the domestic, commercial, and industrial fuel markets rapidly decreased the incentive for continued research into overfire jet systems for application in stoker-fired combustors. Indeed, the number of literature references on this topic fall off rapidly after 1945.

In summary, during the seventy or more years during which overfire air jet application to stoker-fired systems was of significant importance, no generalized design criteria of broad applicability had been developed. An art had arisen regarding the use of overfire jets, typically over the ignition arch and in the sidewalls of traveling grate stokers burning bituminous coal. Sufficient jet design technology had been developed to allow specification of jets which would penetrate adequately the upflow of gases arising from the bed and which served to smooth the temperature and gas composition profiles at the entrance to the boiler passes above. Even in 1951, however, the comment was made by Gumz[11], a well-recognized contributor to combustion technology, that "the number of nozzles, their location and direction are the most disputed factors in the use of overfire air jets."

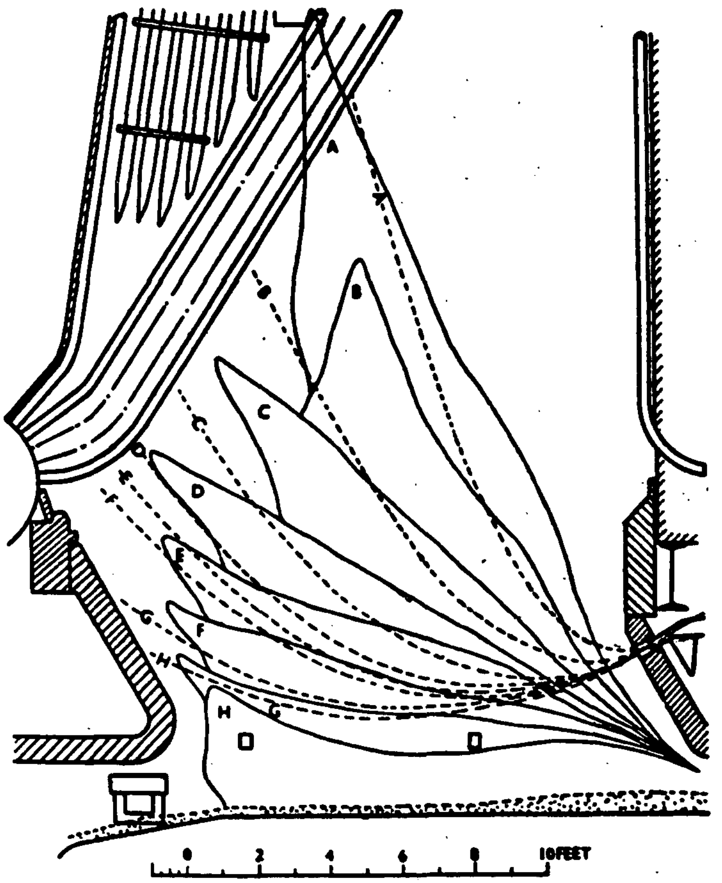


FIGURE IV–4 COMPARISON OF OBSERVED FLAME CONTOURS AND CALCULATED TRAJECTORIES OF OVERFIRE AIR JETS [9]

Percentage of overfire air at the following points; A, 5.65; B, 10.5; C, 16.6; D, 20.0; E, 21.4; F, 22.8; G, 26.8; H, 28.8.

C. REVIEW OF THE PRIOR ART

In 1880, the first patents were issued for steam-air jet devices to supply overfire air and induce turbulence in hand-fired furnaces burning bituminous coal. The development and marketing of these proprietary jet systems reflected a need for improvements in combustion efficiency and for means to reduce smoke emissions. Since that time, a number of refinements in the physical arrangements and design characteristics of overfire jets have been offered to the technical community. In the literature surveyed in the course of this effort, however, few instances were found where comprehensive design correlations were presented. In the great majority of cases (e.g., references 12, 13, and 14), the technical content of the papers was limited to documentation of improvements in performance, particularly with reference to smoke abatement, resulting from the use of specific arrays of overfire jets in a specific combustor. Most design information dealt with such topics as the pumping efficiency or the estimation of steam consumption in steam ejectors. We found few instances where attempts were made to couple an analysis of jet behavior to an analysis of furnace behavior.

To some extent, the tendency of early workers to report only empirical results reflected the limitations of theoretical understanding or mathematical treatment techniques of their time. Also, the complexity of furnace dynamics presents a considerable challenge to the analyst and thus generalization is difficult. It is noteworthy, for example, that the rigorous mathematical treatment of the behavior of two-dimensional plane jets has only recently been solved in detail.[15] Solution of this problem required the use of high-speed computers and complex numerical techniques. Efforts at a similar analysis of the axi-symmetrical round jet are now in process but, at the present time, trial solutions exceed reasonable core storage and computation time on the fastest, modern computers.

Ideally, the design basis used in overfire air jet designs for incinerator applications should recognize the effects of buoyancy, cross-flow and combustion as they are experienced in real incineration systems. With two exceptions[16,17], our review of the existing art showed no correlations able to cope with this full spectrum of potential interactions. These workers, however (Davis[16] and Ivanov[17]), were concerned with coal-fired systems, and application of their correlations to incinerators must be approached with caution. It is of value, however, to review the prior art as it reflects the basis of our analysis. Much of the prior work, although supported by analytical studies, rests heavily on experimental results, particularly to provide the constants used in the equations. Indeed, in view of the complexity of even modest deviations from simple isothermal free jet behavior, our work showed the need for empirical results as a check on analytical predictions.

1. ROUND ISOTHERMAL JETS

The behavior of circular jets discharging into a quiescent, non-reacting environment at a temperature similar to that of the jet fluid provides the starting point in any review of jet dynamics. Indeed, the behavior of jets under such conditions has often been the primary guideline in the design of overfire jets for incinerator applications.[18] Because of the relatively simple nature of jet structure under such conditions, this configuration is perhaps the most studied, both analytically and experimentally, and good correlations are available describing jet trajectory, velocity, entrainment, turbulence levels, and the like.

The correlation of data taken by many experimenters leads to the following expressions for the axial decay of centerline velocity and concentration.[19]

$$\frac{\bar{u}_m}{\bar{u}_o} = 6.3 \left(\frac{\rho_o}{\rho_a}\right)^{1/2} \frac{d_o}{(x + 0.6\, d_o)} \qquad \text{(IV-1)}$$

$$\frac{\bar{c}_m}{\bar{c}_o} = 5.0 \left(\frac{\rho_o}{\rho_a}\right)^{1/2} \frac{d_o}{(x + 0.8\, d_o)} \qquad \text{(IV-2)}$$

where $\bar{u}_m$ and $\bar{c}_m$ are the time-averaged centerline velocity and concentration of jet fluid at distance x downstream from the nozzle mouth; $\bar{u}_o$ and $\bar{c}_o$ are the comparable characteristics at the nozzle mouth; and d_o is the nozzle diameter. ρ_o and ρ_a are the densities of the jet and ambient fluids respectively. These equations apply only in the fully developed region (i.e., $x/d_o > 8$).

Abramovich[20] gives the axial velocity and concentration decay to be:

$$\frac{\bar{u}_m}{\bar{u}_o} = 0.48 \left(\frac{d_o}{a_1 x}\right) \qquad \text{(IV-2a)}$$

$$\frac{\bar{c}_m}{\bar{c}_o} = 0.35 \left(\frac{d_o}{a_1 x}\right) \qquad \text{(IV-2b)}$$

where the value of a depends on the velocity profile at the nozzle mouth. For a flat profile, $a_1 = 0.066$. For an equilibruum turbulent velocity profile, $a_1 = 0.076$. Substituting the latter value into Equations (IV-2a&b) yield

$$\frac{\bar{u}_m}{\bar{u}_o} = 6.3 \left(\frac{d_o}{x}\right) \qquad \text{(IV-2c)}$$

$$\frac{\bar{c}_m}{\bar{c}_o} = 4.6 \left(\frac{d_o}{x}\right) \qquad \text{(IV-2d)}$$

which agree well with Equations (IV-1) and (IV-2).

The experimentally measured radial velocity and concentration profiles in the fully developed region can be represented by either Gaussian or cosine functions. The Gaussian representations are:[19]

$$\frac{\bar{u}}{\bar{u}_m} = \exp\left[-96 \left(\frac{r}{x}\right)^2\right] \qquad \text{(IV-3)}$$

$$\frac{\bar{c}}{\bar{c}_m} = \exp\left[-57.5 \left(\frac{r}{x}\right)^2\right] \qquad \text{(IV-4)}$$

where $\bar{u}$ and $\bar{c}$ are the time-averaged velocity and concentration at distance x downstream and distance r from the jet centerline.

Abramovich[20](p. 89-97) applies Taylors physical model of turbulence and shows that this theory predicts that

$$\frac{\bar{c}}{\bar{c}_m} = \left(\frac{\bar{u}}{\bar{u}_m}\right)^{0.5} \qquad \text{(IV-4a)}$$

The relationship between Equations (IV-3) and (IV-4)

$$\frac{\bar{c}}{\bar{c}_m} = \left(\frac{\bar{u}}{\bar{u}_m}\right)^{\varepsilon} \qquad \text{(IV-4b)}$$

or

$$\exp\left[-57.5\left(\frac{r}{x}\right)^2\right] = \left[\exp\left[-96\left(\frac{r}{x}\right)^2\right]\right]^{\Sigma} \qquad \text{(IV-4c)}$$

so that

$$\Sigma = \frac{-57.5}{-96} = 0.6$$

which agrees quite well with the value of 0.5 derived by Abramovich.

The spread of the jet is defined in terms of the half-angle to the half-velocity point (i.e., the angle subtended by the jet centerline and the line from the centerline at the nozzle mouth to the point where the velocity is one-half of the centerline velocity . This angle is independent of distance from the nozzle mouth in the fully developed region, a consequence of the self-preserving nature of the velocity profile. The half-angle of the half-velocity point is 4.85°; based on concentration in the same way, the half-angle is 6.2°.

The turbulent intensity of the jet is defined in terms of u' and v', the r.m.s. fluctuating velocity components in axial and radial directions, respectively. Data of Corrsin[21] show that the intensity ratio u'/u_m and v'/u_m depend on the ratio r/x. At $x/d_o = 20$, each velocity ratio varies from about 27 percent at the centerline to about 5-7 percent at $r/x = 0.16$.

Ricou and Spaulding[22] measured entrainment rates and determined that the mass flow rate ($\dot{m}_x$) in the jet is linearly related to x according to

$$\frac{\dot{m}_x}{\dot{m}_o} = 0.32 \left(\frac{\rho_a}{\rho_o}\right)^{1/2} \left(\frac{x}{d_o}\right) \qquad \text{(IV-5)}$$

A similar relationship can be computed from Equations (IV-1) and (IV-3) for the uniform density case. Defining

$$\dot{m}_x = \int_0^\infty \rho \bar{u} 2\pi r dr \qquad \text{(IV-6)}$$

and

$$\dot{m}_o = \int_o^{d_o/2} \rho \bar{u}_o 2\pi r dr = \frac{\rho \bar{u}_o \pi d_o^2}{4} \qquad \text{(IV-7)}$$

Taking the ratio of Equations (IV-6) and (IV-7), substituting equations (IV-1) and (IV-3) to give us a function of x and r, and

$$\frac{\dot{m}_x}{\dot{m}_o} = 0.26 \left(\frac{x}{d_o}\right) \qquad \text{(IV-8)}$$

which agrees well with Equation (IV-5).

These relationships describing the behavior of isothermal jets entering a quiescent fluid are well-documented and form the basis for our design criteria relative to the use of jets in incinerators. In this application, however, the effects of crossflow of the ambient fluid and density differences between the jet and ambient fluids are important. These effects are less well-documented, and their inclusion in the design criteria poses some difficult problems. These matters are dealt with below.

2. BUOYANCY EFFECTS

When the jet and ambient fluids are of different density, the buoyant forces acting on the jet can cause deflections of the jet trajectory. This effect is potentially important in incinerator applications since the air introduced by the jets will be much colder than the furnace gases and hence of higher density. From an incinerator design and operating standpoint, this could be critical: jets could "sink" from an anticipated flow trajectory passing above the bed to one causing entrainment of particulate from the bed or causing overheating of the grates with a "blowpipe" effect.

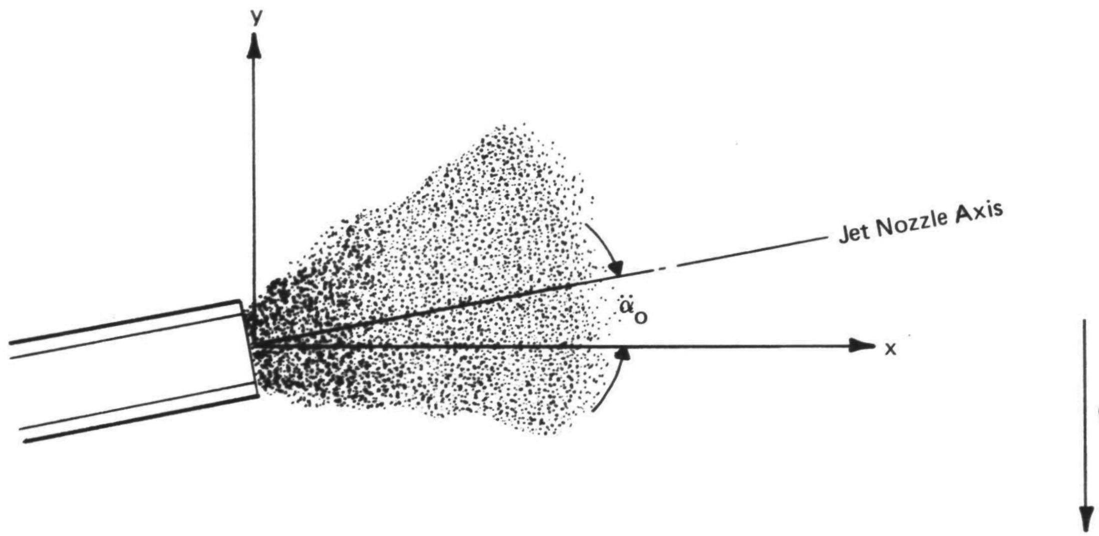
Relatively little experimental or theoretical work has been done to characterize jet performance under these conditions. Figure IV-5 shows the geometry of the system considered and defines symbols used in the discussion. A jet of density ρ_o issues at a velocity of u_o from a circular nozzle of diameter d_o. The ambient fluid is at rest and of density ρ_1.

Abramovich[20] analyzed the trajectory of a heated jet issuing into a cold ambient fluid and compared his theoretical result with the data of Syrkin and Lyakhovskiy.[23] The resulting expression is:

$$\left(\frac{y}{d}\right) = 0.052 \left(\frac{gd}{u_o^2}\right) \left(\frac{\rho_a - \rho_o}{\rho_o}\right) \left(\frac{x}{d}\right)^3 \qquad \text{(IV-9)}$$

Figure IV-6 shows a comparison of this expression with experimental data in which the ratio $(\rho_a - \rho_o)/\rho_o$ was varied in the approximate range of 0.2 to 0.8. Equation (IV-9) generally underestimates the buoyancy induced deflection of the jet.

Field et. al.[19] also considered the behavior of a buoyant jet and obtained the expression:



Jet Properties:
Nozzle Velocity u_o
Density ρ_o

Ambient Fluid Properties
At Rest
Density ρ_a

FIGURE IV-5 SCHEMATIC OF JET FLOW

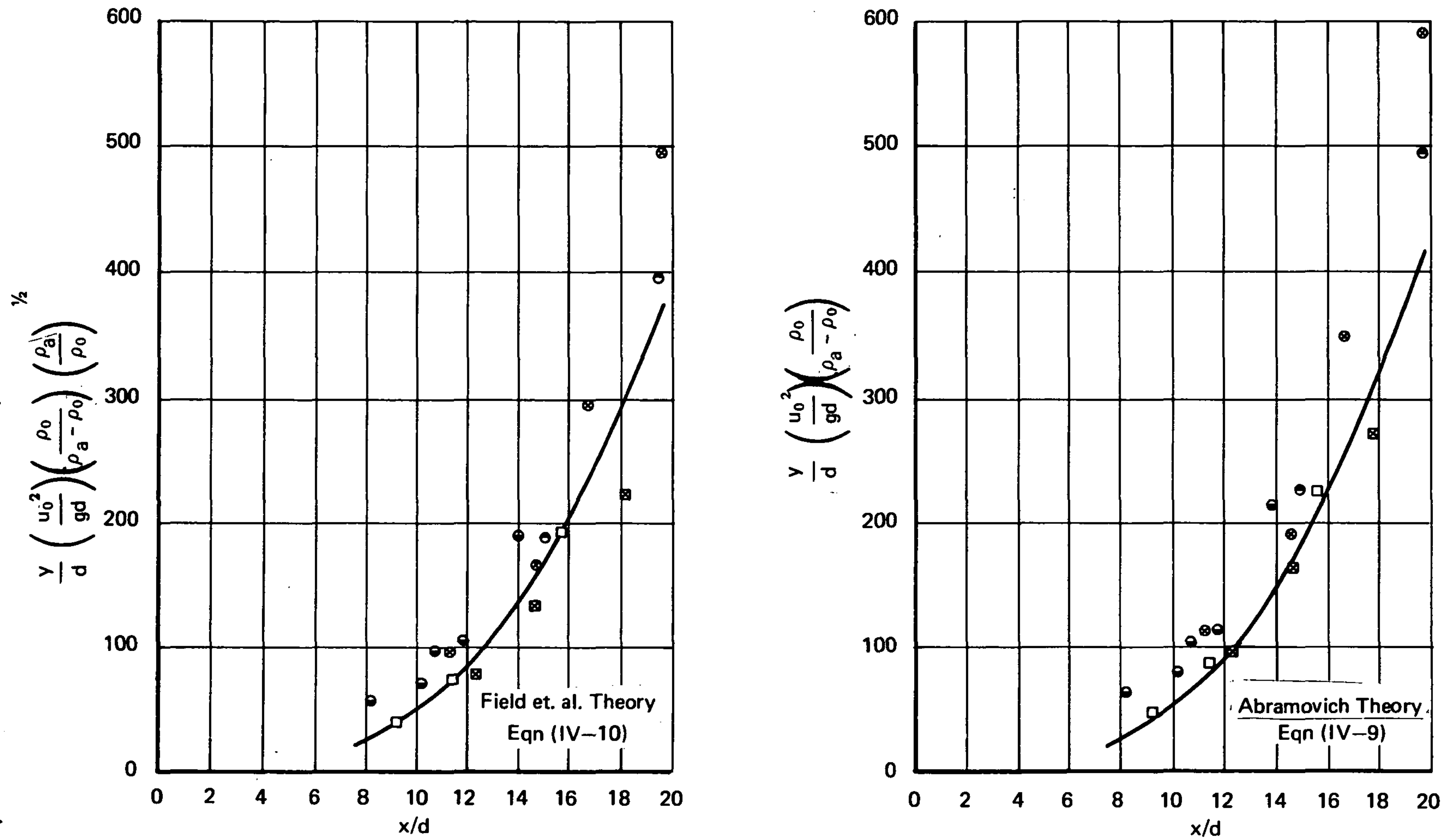


FIGURE IV–6 COMPARISON OF THE PREDICTIONS OF ABRAMOVICH[20] AND FIELD ET. AL.[19] WITH THE DATA OF SYRKIN AND LYAKHOUSKY[23] ON BUOYANT JET BEHAVIOR

$$(\frac{y}{d}) = (\frac{x}{d}) \tan \alpha_o + \frac{(0.047)}{\cos \alpha_o} (\frac{gd}{u_o^2}) (\frac{\rho_a - \rho_o}{\rho_o}) (\frac{\rho_a}{\rho_o})^{1/2} (\frac{x}{d})^3 \qquad \text{(IV-10)}$$

For a jet injected normal to the gravity field ($\alpha_o = 0$), Equation (IV-10) reduces to

$$(\frac{y}{d}) = 0.047 (\frac{gd}{u_o^2}) (\frac{\rho_a - \rho_o}{\rho_o}) (\frac{\rho_a}{\rho_o})^{1/2} (\frac{x}{d})^3 \qquad \text{(IV-10a)}$$

which differs from Equation (IV-9) in the value of the leading constant (0.047 as opposed to 0.052) and the presence of the term $(\rho_a/\rho_o)^{1/2}$. In incinerator applications, where the jet and ambient temperatures are approximately 100°F (560°R) and 2500°F (3060°F), respectively, this term has the value of

$$(\rho_a/\rho_0)^{1/2} = (T_o/T_1)^{1/2}$$
$$= (560/3060)^{1/2}$$
$$= 0.43$$

The deflections predicted by the two equations will differ by a factor of two.

Figure IV-6 shows a comparison of Equation (IV-10a) with the data of Syrkin and Lyakhovskiy.[23] In these data, the term $(\rho_a/\rho_o)^{1/2}$ varies from about 1.1 to 1.4. The inclusion of the term $(\rho_a/\rho_o)^{1/2}$ results in better agreement with the data, particularly at the larger values of x/d_o.

3. CROSSFLOW EFFECTS

The need to understand the behavior of a jet issuing into a crossflow normal to the jet axis arises in the analysis of furnaces, plume dispersion from chimneys and elsewhere. The deflection of the jet by the crossflow has been studied extensively, both experimentally and analytically, although most workers have limited their work to descriptions of centerline trajectory and gross entrainment rates. To our knowledge, no analysis has been carried out, either experimentally or theoretically, which characterizes in detail the radial distribution of velocity or concentration.

With crossflow, the interaction of the flows deflects the jet and alters the cross-sectional shape of the jet. Figure IV-7 is a diagram of the jet cross-section several nozzle diameters along the flow path. The originally

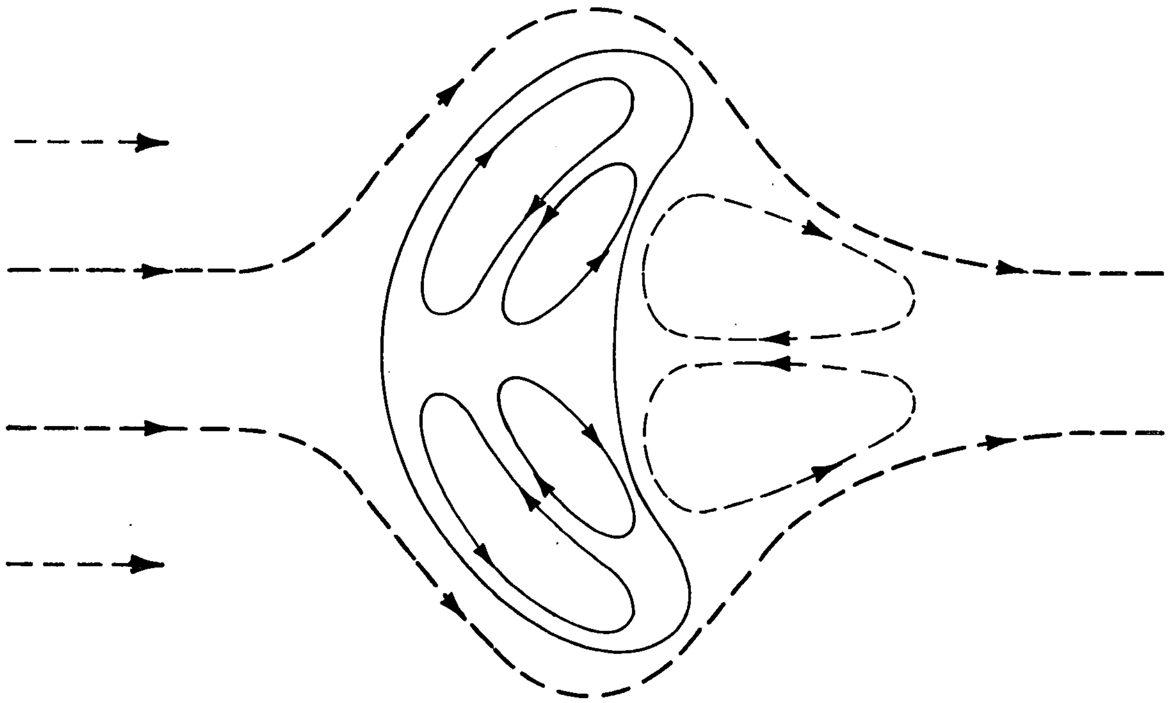


FIGURE IV–7 JET CROSS SECTION AND CIRCULATION PATTERNS FOR ROUND JETS IN CROSS FLOW circular cross-section has been distorted into a horseshoe shape by the shearing action of the external flow around the jet, and internal patterns of circulation have been set up. Measurements in the external flow around the jet show a decreased pressure downstream of the jet, recirculation of the external fluid, and a process leading to the periodic shedding of vortices into the wake of the jet. These phenomena are similar to those observed in the wake of a solid cylinder exposed to crossflow.

Dimensional analysis considerations suggest that the coordinates of the jet axis (x/d_o, y/d_o) should depend on the ratio of momentum fluxes in the external and jet flows $M = \left(\frac{\rho_a u_1^2}{\rho_o u_o^2}\right)$ and the Reynolds number $Re = \left(\frac{\rho_o \bar{u}_o d_o}{\mu_o}\right)$.

For turbulent jets in the Reynold's number range above 10^4, correlation of experimental data suggest that the Reynold's number effect is negligible and the momentum ratio is the predominant variable characterizing the flow.

In terms of the geometry illustrated in Figure IV-8, the following expressions for computing the axial trajectory of a single jet have been reported. The jet axis is taken to be the locus of maximum velocity.

$$\left(\frac{y}{d_o}\right) = 1.0\,(M)^{1.12}\left(\frac{x}{d_o}\right)^{2.64} \qquad 0 < M \leq .023,\ \alpha_o = 0 \qquad \text{(IV-11)}$$

Reference: Patrick[24] (1967)

$$\left(\frac{y}{d_o}\right) = M\,\frac{x}{d}^{2.55} + (1 + M)\,[\tan \alpha_o]\left[\frac{x}{d_o}\right] \qquad .046 \leq M \leq 0.5 \qquad \text{(IV-12)}$$

Reference: Shandorov[25] as cited in Abramovich[20]

$$\left(\frac{y}{d_o}\right) = (M)^{1.3}\left(\frac{x}{d}\right)^{3} + [\tan \alpha_o]\left[\frac{x}{d_o}\right] \qquad .001 \leq M \leq 0.8 \qquad \text{(IV-13)}$$

Reference: Ivanov[17] as cited by Abramovich[20]

$$\left(\frac{y-y^+}{d_o}\right) = 5.5\,M^{1.175}\left(\frac{x-x^+}{d_o}\right)^{2.175} \qquad 0.01 \leq M \leq 0.028,\ \alpha_o = 0 \qquad \text{(IV-14)}$$

where y^+/d and x^+/d denote the end of a zone of establishment. Values vary somewhat with M but are of the order of one or less.

Reference: Keffer and Barnes[26] as cited by Field et. al.

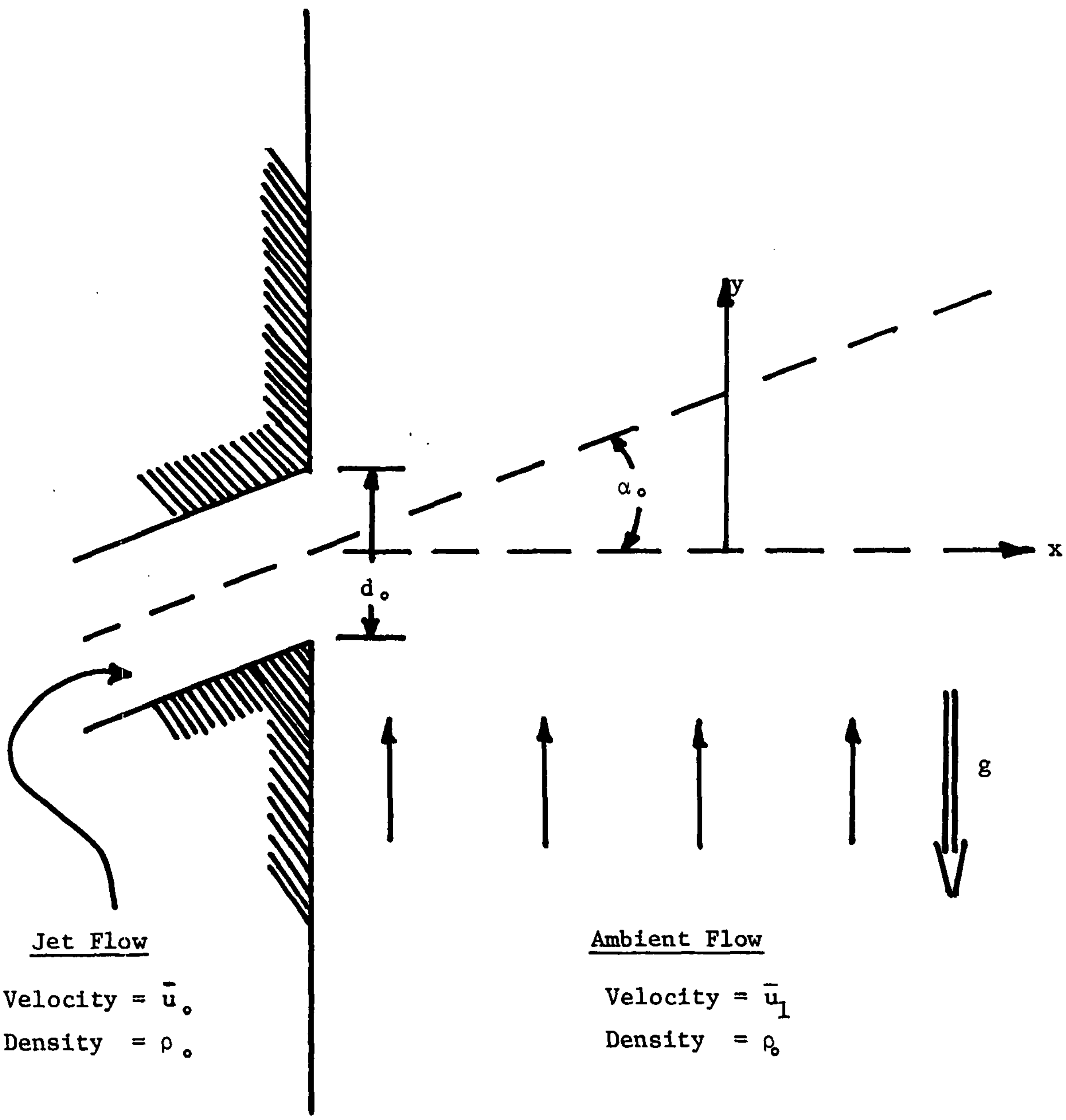


FIGURE IV-8 COORDINATE SYSTEM FOR ROUND JET IN CROSSFLOW

Abramovich[20] derived an analytical relation of the form:

$$(\frac{x}{d_o}) = 14.4 \ \frac{1}{MC_x})^{1/2} \ \log \ [1 + 0.1 \ (\frac{y}{d_o}) \ (1 + \ 1 + 20 \ \frac{d_o}{y}] \qquad \text{(IV-15)}$$

where C_n is an effective drag coefficient relating drag on the jet to the momentum flux in the external flow.

A simplified treatment of jet behavior in crossflow with temperature effects was presented by Davis[16] in 1937. The crossflow effect was introduced by the assumption that tha jet rapidly acquired a velocity component equal to the crossflow velocity. The jet was then seen to follow a path corresponding to vector addition of the crossflow velocity to the jet centerline velocity (the latter being calculated using a simplified velocity decay law by Tollmien[27]). Temperature effects were introduced as being reflected in increases in jet velocity due to expansion of the cold nozzle fluid (initially at T_o) after mixing with the hot furnace gases (at T_s). Davis' final equation for the deflection (y) is given by:

$$y = \frac{u_1 x(x + 4d_o \cos \alpha_o)}{2a_1 d_o \bar{u}_o \cos^2 \alpha_o} [\frac{T_o}{T_s}]^{1/3} + \tan \alpha_o \qquad \text{(IV-16)}$$

where "a_2" is a constant depending on nozzle geometry (1.68 for round jets and 3.15 for long, narrow plane jets). The many rough assumptions in Davis' analysis (some of which have been shown to be in error) would indicate that its use should be discouraged. Comparison of calculated trajectories with observed flame contours, however, (Figure IV-4) suggests it may have some general value. Interpretation of the meaning of the general agreement between calculated jet trajectory and flame contour as shown in Figure IV-4 is difficult, however, and use of the Davis equation in incinerator applications is uncertain.

Patrick[24] reported the trajectory of the jet axis (defined as the maximum concentration) to be

$$(\frac{y}{d_o} = 1.0 \ M^{1.25} \ (\frac{x}{d_o})^{2.94} \qquad \text{(IV-17)}$$

Figure IV-9 shows a plot of the velocity axis [Equation (IV-12)] and the concentration axis [Equation (IV-17)] for a value of M = 20. The concentration axis shows a larger deflection than does the velocity axis. This is probably due in part to the assymetry of the external flow around the partially deflected jet. Also, recent calculations by Tatom[15] for plane jets suggest that under crossflow conditions, streamlines of ambient fluid can be expected to cross the jet velocity axis.

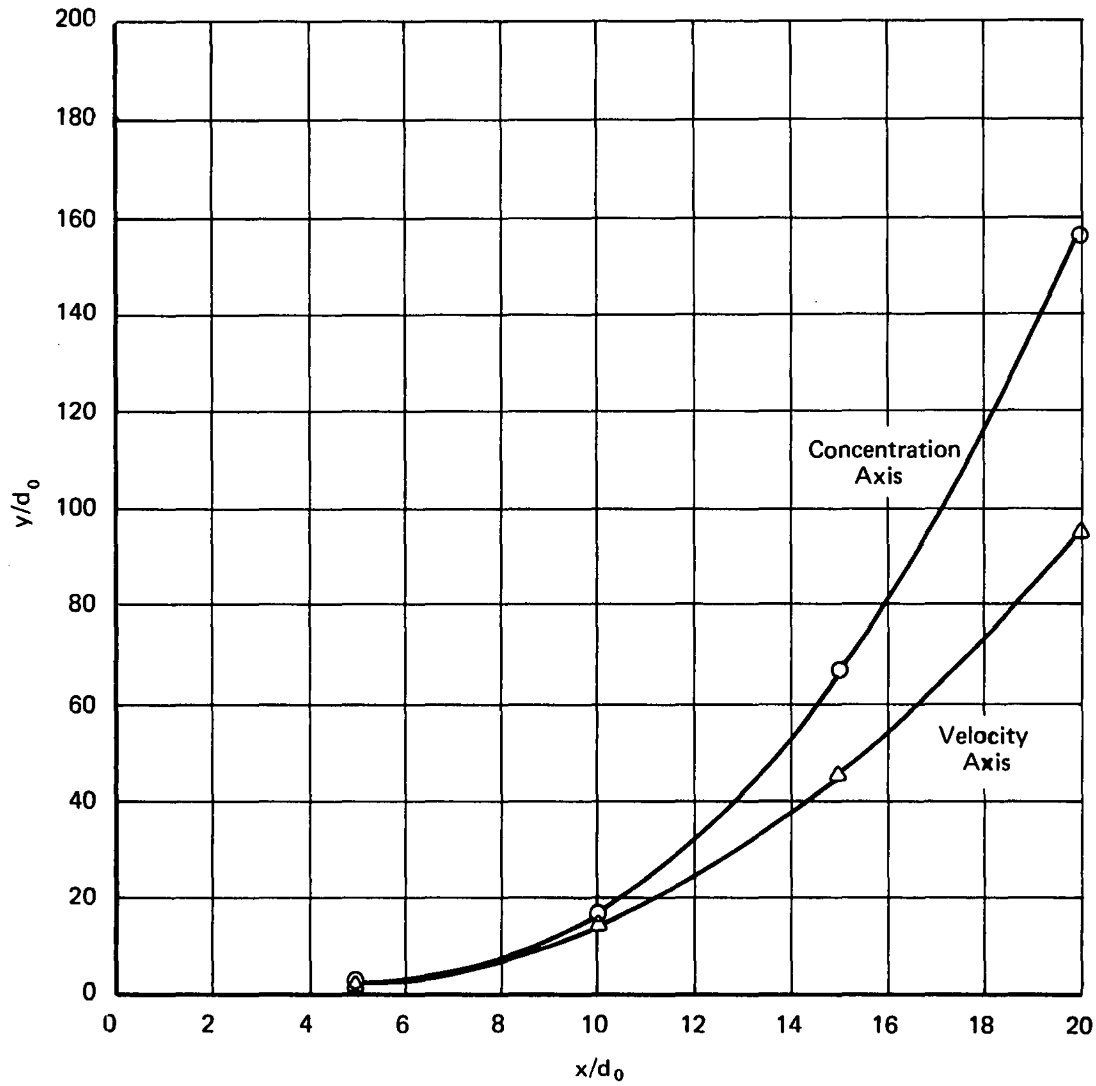


FIGURE IV–9 TRAJECTORY OF CONCENTRATION AND VELOCITY AXES FOR JETS IN CROSS FLOW (–DATA OF PATRICK[24]–) M = 0.05

For our purposes, we are interested in jets which penetrate reasonably far into the crossflow (i.e., those which have a relatively high velocity relative to the crossflow). The empirical equations of Patrick[24] [Equation (IV-11)] and Ivanov[17] [Equation (IV-13)] were developed from data which satisfy this condition. Figures IV-10 and IV-11 show comparisons of these two equations for values of M of 0.001 ($\bar{u}_o/u_1 = 30$) and M = 0.01 ($\bar{u}_o/u_1 = 10$). Ivanov's expression predicts higher deflections at large (x/d_o, particularly at M = 0.01.

Ivanov[17] also investigated the effect of the spacing between jets in a linear array on jet trajectory. He measured the trajectories of jets under conditions where M = 0.01 at spacings of 16, 8 and 4 jet diameters. His results are shown in Figure IV-12 along with the trajectory of a single jet (infinite spacing). The data show that reducing the spacing between jets causes greater deflection of the jets. As spacing is reduced, the jets tend to merge into a curtain. The blocking effect of the curtain impedes the flow of external fluid around the jets and increases the effective deflecting force of the external fluid. The increase in deflection is most notable as s/d_o is reduced from 16 to 8. Above $s/d_o = 16$, the merging of the jets apparently occurs sufficiently far from the nozzle mouth to have little effect on the external flow. At $s/d_o = 8$, the jet merger apparently takes place sufficiently close to the nozzle mouth that further reduction in spacing has little added effect.

Earlier data (Abramovich[20]) on water jets colored with dye issuing into a confined, cross-flowing stream was correlated in terms of jet penetration distance. The penetration distance, L_j, was defined as the distance between the axis of the jet moving parallel to the flow and the plane containing the nozzle mouth. The axis was defined as being equidistant from the visible boundaries of the dyed jet. The resulting correlations was

$$\frac{L_j}{d_o} = k \frac{\bar{u}_o}{u_1} \qquad \text{(IV-18)}$$

where k is a coefficient depending on the angle of attack and the shape of the nozzle. Defining the angle of attack (β) as the angle between the jet and the crossflow velocity vectors, ([90 - α_o] in the terminology shown in Figure IV-8), the recommended values of k are:

For β = 90°, for round and square nozzles; k = 1.5

For β = 90°, for rectangular nozzles; k = 1.8

For β = 120°, for all nozzles; k = 1.85

Figure IV-13 shows a comparison of Ivanov's correlation [Equation (IV-13)] with the jet penetration correlation [Equation (IV-18)], for M = 0.001 and M = 0.01. Equation (IV-18) predicts a smaller jet penetration than does Equation (IV-13). There are two possible explanations for this discrepancy.

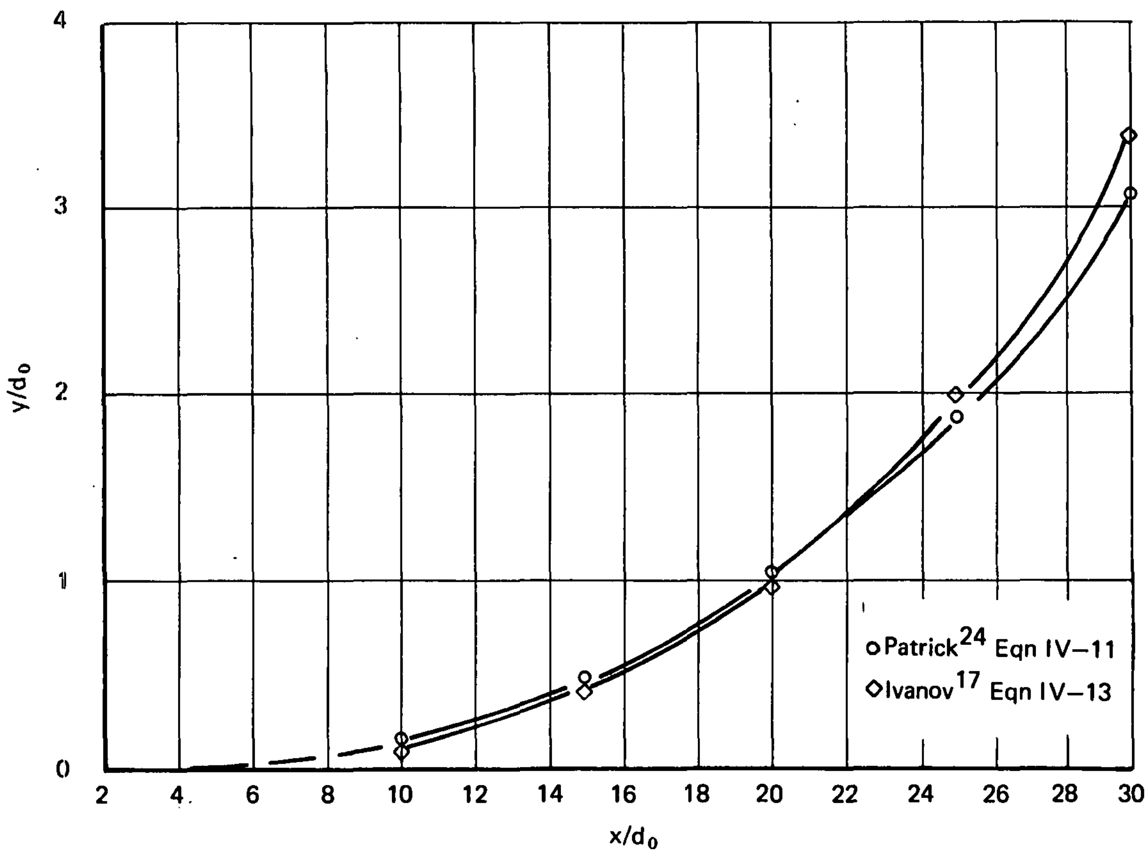


FIGURE IV–10 COMPARISON OF TRAJECTORIES AT M= 0.001 ($\bar{u}_0/u_1$ = 31.6) FOR JETS IN CROSSFLOW

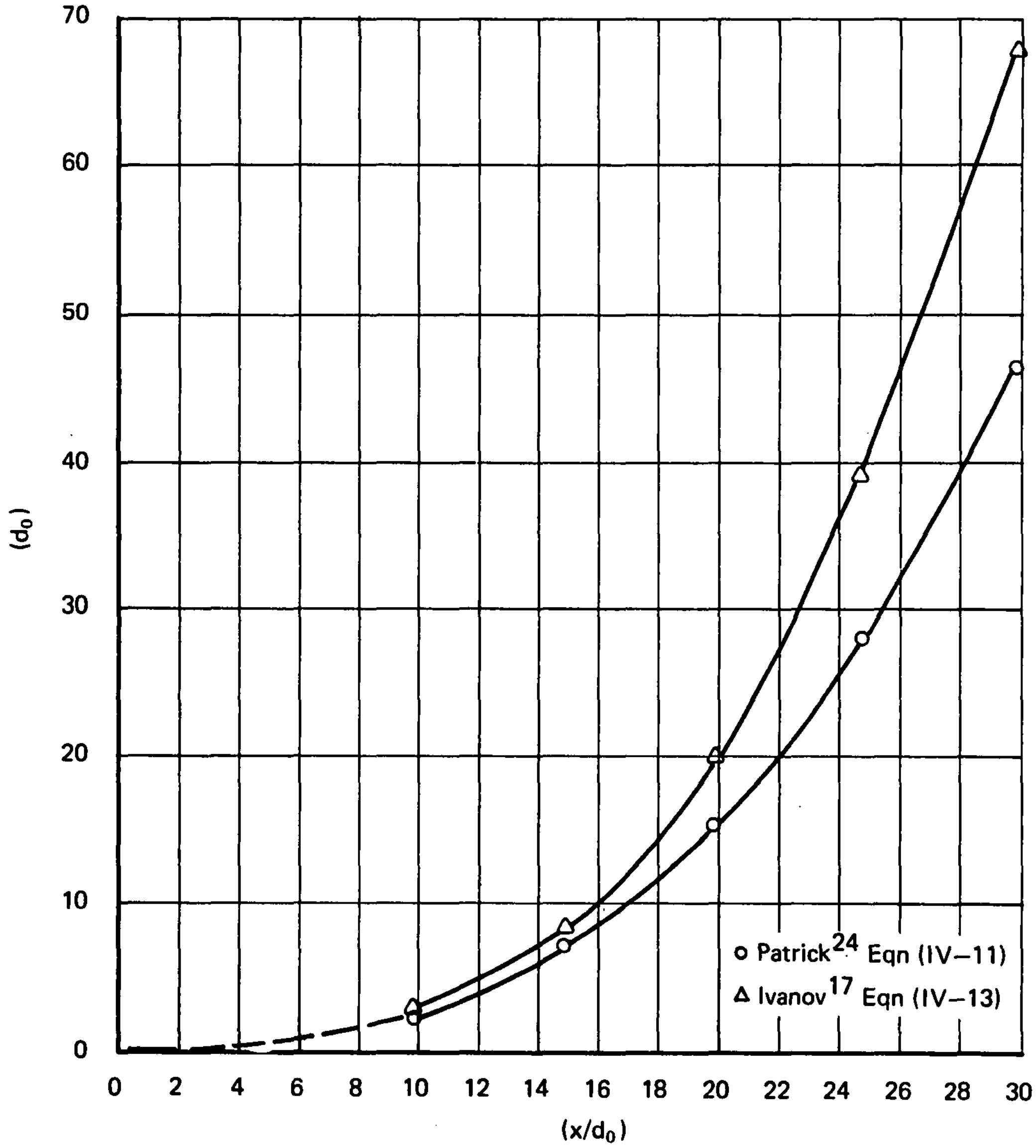


FIGURE IV–11 COMPARISON OF TRAJECTORIES AT M = 0.01 ($\bar{u}_0/u_1$ = 10) FOR JETS IN CROSS FLOW

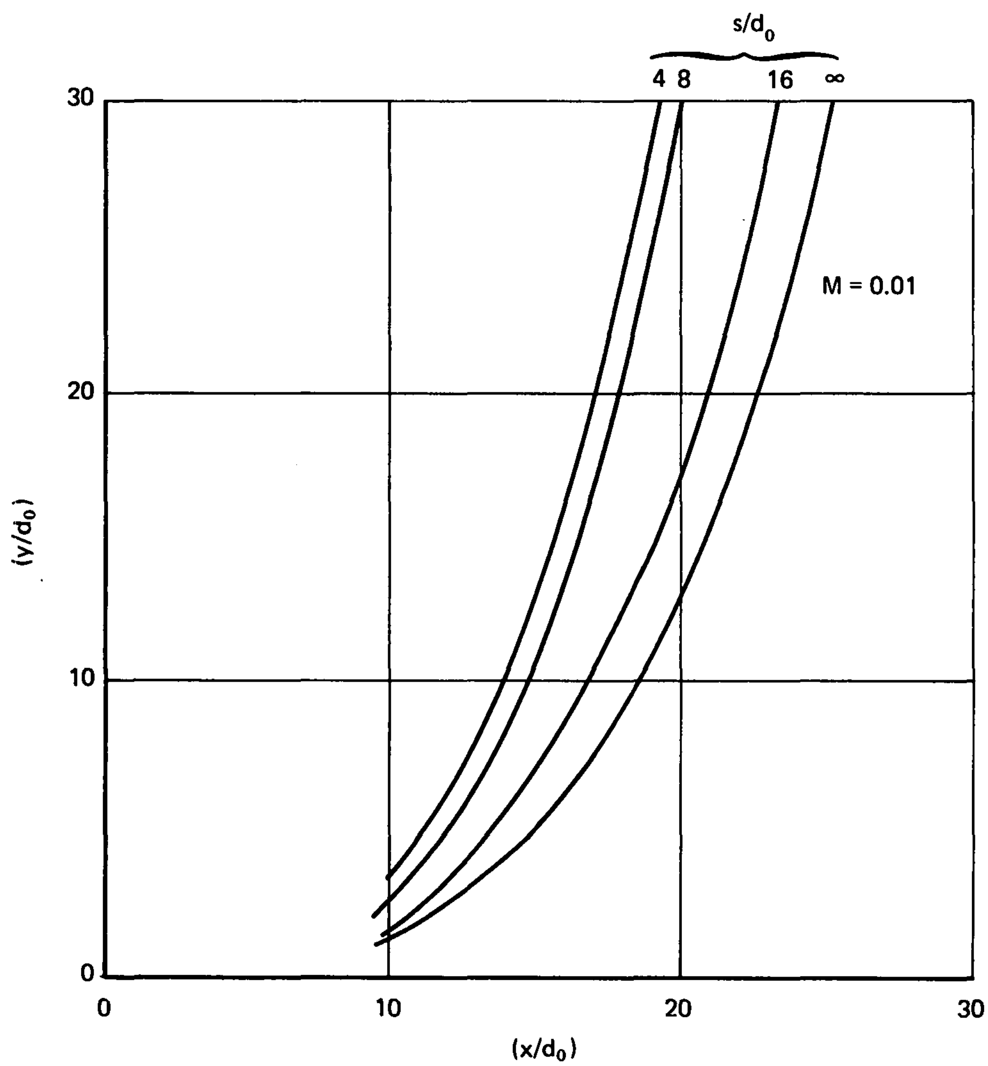


FIGURE IV–12 EFFECT OF JET SPACING ON TRAJECTORY FOR JETS IN CROSS FLOW (AFTER IVANOV[17])

a. The data on which Equation (IV-13) was based do not extend to large values of x/d_o, and extrapolation of the data may be in error.

b. The data on which Equation (IV-18) was based were taken in a confined crossflow in which the lateral dimension (normal to both the jet axis and the crossflow) was sufficiently small to interfere with normal jet spreading. The jet effectively filled the cross-section in the lateral dimension, behaving like a series of jets at low spacing.

Ivanov's data (Figure IV-12) show that penetration is reduced at lower spacing. The penetration given in Equation (IV-13) for M = 0.01 was reduced by 25% (see the dotted trajectory marked $s/d_o = 4$, M = 0.01 in Figure IV-13). Agreement between this adjusted trajectory and the penetration given by Equation (IV-18) is better.

4. BUOYANCY AND CROSSFLOW

When a cold air jet is introduced tnto a crossflowing combustion chamber, both buoyancy and crossflow forces act simultaneously on the jet.

Abramovich[20] reports the results of experiments conducted by injecting cold jets into a hot crossflow. Temperature ratios of as much as 3 to 1 were used (corresponding to the jet fluid having a density three times that of the crossflowing fluid), with the values of M in the range of 0.045 to 0.5. The normal crossflow trajectory equation correlated the data when the value of M was computed using actual fluid densities. From these data, Abramovich concluded that buoyancy effects could be neglected, other than as density differences were incorporated into the crossflow parameter M.

The same conclusion was drawn by Ivanov[17] who injected hot jets into a cold crossflow. The ratio of temperature (and density) between jet and ambient fluids was 1.9 and M ranged from 0.005 to 0.02. The geometry of the tests was not clearly stated by either Abramovich or Ivanov. It appears that the buoyancy force acted in the same direction as the crossflow force in Ivanov's tests (i.e., a hot jet discharging into an upflow).

Application of these conclusions to incinerator design practice, however, is subject to question because of the large geometrical scale-up involved. Physical reasoning suggests that the ratio of buoyant force to drag force acting on a non-isothermal jet in crossflow depends on scale. The buoyant force (B) is a body force and is, therefore, proportional to jet volume. The drag force (D) exerted by the crossflow has the characteristics of a surface force and is, therefore, proportional to the effective cylindrical area of the jet. Therefore, the ratio of buoyant force to drag force is proportional to jet diameter. A simple analysis, discussed in Section D, gives:

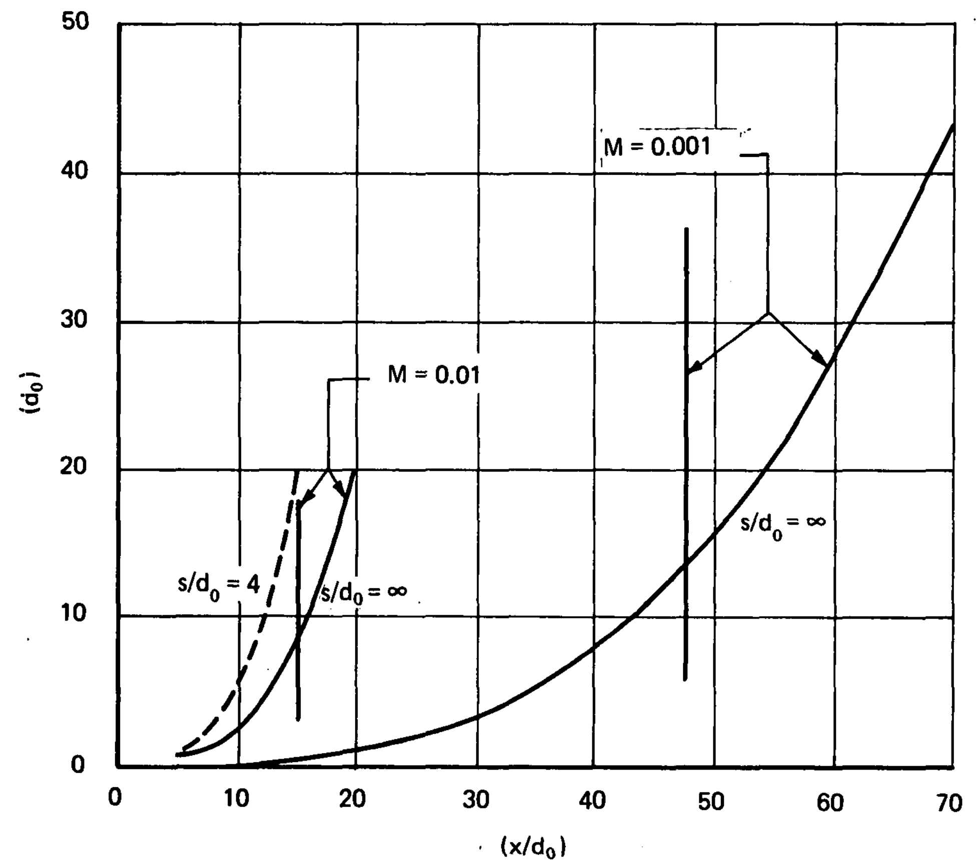


FIGURE IV–13 COMPARISON OF IVANOV'S TRAJECTORY CORRELATION[17] (Egⁿ IV–13) WITH JET PENETRATION CORRELATION (Egⁿ IV–18)

$$\frac{B}{D} = \frac{\pi}{2C_x} \left(\frac{gd_o}{u_1^2}\right) \left(1 - \frac{\rho_o}{\rho_1}\right) \qquad \text{(IV-19)}$$

where C is the effective drag coefficient.

The value of the effective drag coefficient (C_x) is believed to be in the range of 1 to 4; analysis given in Section D suggests that 4 is the better value.

Typical values of the physical parameters in Ivanov's experiments[17] are:

d_o = 5 to 20 mm (0.0164 to 0.063 ft.)

$\bar{u}_o/u_1$ = 10 to 20

$T_o/T_1 = \rho_a/\rho_o = 2$

$\rho_o \bar{u}_o^2/\rho_a u_1^2$ = 100 to 200

u_1 = 3.68 to 4.16 m/sec (12 to 13.6 ft/sec)

The maximum value of B/D results from the maximum value of nozzle diameter d and the minimum value of crossflow velocity u_1. Substitution of these values into Equation (IV-19) yields a force ratio of 0.0026 which indicates that the crossflow effect completely dominated the buoyant effect on the small jets used in Ivanov's tests. Therefore, we speculate that his results (no buoyancy effect) could be anticipated under his test conditions.

In incinerator applications, jet diameters in the neighborhood of 4 inches are contemplated, along with crossflow velocities in the order of 2 to 5 feet per second. For a 4 inch jet in a 2 feet per second crossflow with the same 1.91 density ratio as in Ivanov's tests,

$$\frac{B}{D} = \frac{(3.14)}{(2)(4)} \; \frac{(32.2)(0.033)}{(2)^2} \; \left(1 - \frac{1}{1.91}\right)$$

$$= 0.50$$

which suggests that the buoyant and crossflow forces are of the same order of magnitude.

5. DESIGN METHODS

The correlations given above provide the basic tools for the analysis of jet behavior in real furnace environments. In general, the correlations are based in theory and corroborated with data. Translation of behavioral relationships into designs can be approached in two ways:

- Detailed analysis of the actions and interactions of each component of the system under design, "building" an understanding of system behavior from an understanding of its parts; and

- Assembly of generalized correlations into "rules-of-thumb" and the like which show applicability to a number of systems similar to the device in question.

The jet design correlations above, when coupled with the bed burning and chamber flow analyses, are supportive of the first approach. As such, they are broadly applicable but their use makes demands upon the designer for data and understanding which he may not possess.

Approaches to a "rule-of-thumb", generalized method for overfire air jet design have been proposed by Ivanov[17] and by Bituminous Coal Research, Inc.[28] Although these design guides were developed for coal-fired boilers, they are presented here as an indication of approaches successful in other applications. Their applicability to incineration systems, however, has not been shown.

a. <u>The Method of Ivanov</u>--Ivanov conducted a number of experiments in a non-combusting model furnace to determine the effects of various jet configurations on the temperature profiles above a burning coal bed. He concluded that:

  - It is preferable to position overfire jets in the front wall of the furnace rather than in the rear wall.

  - Close spacing of the jets is desirable in order that the jets form an effective curtain. Above this curtain, a rotary motion of the gases is induced, which contributes greatly to the mixing process.

  - If maximum temperatures occur near the center of the grate, rather than near the front, the design depth of penetration of the jet should be increased by 5%.

  - Slightly better mixing is obtained if jets are fired from one wall, rather than if the same flow is divided between jets on opposite walls. This applies whether the opposing sets of jets are directly opposed, staggered but on the same horizontal level, or on different horizontal levels.

  - A given level of mixing is achieved at lower power cost and with introduction of less air if small diameter jets are used rather than large ones.

For conditions which gave good mixing, Ivanov computed the jet penetrations from Equation (IV-20).

$$L_j = d_o k \left(\frac{\bar{u}_o}{u_1}\right) \qquad \text{(IV-20)}$$

and normalized the values obtained with respect to L_T, the axial length of the model furnace.

Using the values of k given on page IV-25 to compute L_j he correlated h/L_T with the relative jet spacings "s/d_o" (where s is the center-to-center jet spacing and d_o is the jet diameter) and obtained the following values:

(s/d_o)	Front Arch Furnace (L_j/L_T)	Rectangular Furnace (L_j/L_T)
4	0.90	0.80
5	0.95	0.90
6	1.10	1.0

This correlation is the basis for his design method.

Ivanov's design method is as follows:

(1) Nozzles should be located not less than 3 nor more than 6 1/2 feet above the fuel bed.

(2) The angle of inclination of the jets is determined by aiming the jets at a point on the grate 4-6 1/2 feet from its far end. Jets fired from the underside of a front arch may be angled downward as much as 50° from the horizontal, if the fuel bed is not disturbed by the resulting jet.

(3) The relative jet spacing should be in the range of s/d equals 4 to 5.

(4) The velocity of gases in the furnace at the cross-section where the jets are located is computed from known overall air rates and grate areas and corrected for temperature. This velocity is the crossflow velocity u_1 and the density is ρ_a.

(5) The jet velocity is set by the capability of the overfire fan, but should always be 200 feet per second for cold jets and 230 feet per second for heated jets. He assumes a fan outlet pressure of about 14 in w.g. and computes the jet velocity from:

$$\bar{u}_o = \sqrt{\frac{2gP}{(1.2)\rho_o}} \qquad \text{(IV-21)}$$

where ρ_o is the density of the nozzle fluid.

(6) The required jet diameter is computed from Equation (IV-22):

$$d_o = \frac{L_j}{k\frac{\bar{u}_o}{u_1}\sqrt{\frac{\rho_o}{\rho_a}}} \qquad \text{(IV-22)}$$

where k is 1.6 for s/d_o = 4-5, and L_j is taken to be a factor times L_T, the axial length of the furnace. the factors were given on page IV-33 and range from 0.8 to 1.10.

(7) The number of nozzles in the row (N) is then calculated from the furnace width (B) and the jet spacing (s) according to:

$$N = \frac{B-4s}{s} \qquad \text{(IV-23)}$$

(8) The required fan capacity is then computed from:

$$Q = N\,\frac{\pi d_o^{\,2}}{4}\,\bar{u}_o \qquad \text{(IV-24)}$$

b. The Bituminous Coal Research (BCR) Method--The National Coal Association has published a design handbook for "Layout and Application of Overfire Jets for Smoke Control", based on work by BCR.[28]

The NCA recommends:

- Side wall placement;
- Location of nozzles about 18 inches above the fuel bed in modern furnaces and from 9 to 12 inches above the bed in older, small furnaces;
- Introduction of from 10% to 30% theoretical air via jets depending on whether the smoke formed is "light" or "heavy."

The design method is as follows:

(1) Read the required volume of air (cfm per lb coal burned per hour) from a table, given the heating value of the coal and whether the smoke is light, moderate or heavy. Compute the air requirement in cfm.

(2) Decide where nozzles will be located (front, side, or back wall).

(3) Read the number of nozzles required from a table, given the dimension of the wall on which the jets are to be located and the penetration distance (equal to the axial dimension of the furnace).

(4) Compute the air requirement per nozzle by dividing the result of Step 1 by the result of Step 3.

(5) Read nozzle diameter and required fan pressure from graphs, given the air requirement per nozzle (Step 4) and the penetration distance (Step 3).

(6) Determine duct size from a nomograph given total air requirement (Step 1).

The design criteria on which this method is based are not readily apparent. Examination of the tables and graphs included in the references shows the following relationships.

- The number of jets is approximately proportional to the length of the furnace wall where the jets are installed and approximately inversely proportional to the penetration distance.

- The penetration distance appears to be defined as that distance required to reduce the velocity of a jet, issuing into a quiescent chamber, to 8 feet per second.

Several qualitative statements can be made. First, the cross-flow velocity does not enter explicitly into the design method. Second, working out several examples shows that relative spacings (s/d) of up to ten or more result. This is at odds with Ivanov's finding that spacings of 4 to 5 jet diameters are optimal.

General Discussion

The design methods cited above apply to furnaces burning coal or shale, and are generally used in boiler design. These applications are characterized by:

- A uniform and predictable fuel supply which burns in a regular and repeatable pattern along the grate;

- Use of high heating value fuel (in the range of 10,000 to 15,000 Btu/lb), with low moisture and ash content;

- The desirability of minimizing excess air so that high combustion temperatures and high heat recovery efficiency can be obtained;

- Relatively low combustion volume per Btu/hr capacity.

In contrast, incinerators are characterized by:

- A variable and generally unpredictable fuel supply; the composition and moisture content very seasonally in a somewhat predictable manner and hourly (as fired) in an unpredictable manner;

- Use of low heating value fuel (4450 Btu/lb average as fired), with relatively high ash (20%) and moisture (28%) content[1];

- No general requirement for high combustion gas temperature, except in heat recovering incinerators;

- Relatively large combustion volumes per Btu/hr capacity.

In both cases, complete fuel burnout is desirable and combustion gas temperatures must be kept below the point where slagging or damage to the refractory occurs. Both types of units have fly ash problems, although the incinerator problem is more severe since relatively large pieces of unburned paper can be lifted into the combustion volume.

The differences in characteristics place different requirements on the overfire jets. Jet systems in incinerators must contend with:

- A shifting combustion profile caused by variations in the upflow gas temperature, composition and velocity, and in the moisture content and composition of the fired refuse;

- Large pieces of partially burned refuse in the combustion volume;

- Large combustion volumes per Btu/hr. which increases difficulty of mixing the combustion gases.

In meeting these conditions, minimization of excess air introduced in the jets is not as important as in heat recovering boilers. The principal factors which mitigate for low excess air in incinerators are draft limitations, higher costs of air pollution control, fan and stack equipment, power costs and the general requirement that the overfire air not quench the combustion reaction. Although these factors are important, realization of complete combustion of pollutants,materials survival and inhibition of slagging are predominant concerns.

D. CONTRIBUTIONS TO THE ART

The published methods for designing overfire jet systems are directed toward coal-fired boilers which fire a well-defined fuel and are generally more square in cross-section than are incinerators. In addition, the design methods ignore the buoyancy effects associated with the introduction of cold jets into the combustion space. The buoyancy effect was neglected, however, on the basis of laboratory data, where the jet diameter was sufficiently small that buoyancy effects would be expected to be negligible. In larger scale systems, this buoyancy effect can be important.

The variable nature of refuse and the bed-burning processes on the temperature and concentration profiles within the incinerator suggests that overfire jet systems in incinerators should have sufficient flexibility to meet the moment-to-moment variations in the location of combustion volume segments which require overfire oxygen or turbulence.

The high aspect ratio of incinerators (large grate length to width ratio) takes the design of end wall jet systems into a region of jet behavior where relatively little data exists. Few measurements have been made of jet behavior at large (several hundred) values of normalized axial distance (x/d).

In building on what is known in the design of overfire air jet systems for incinerators, we are cognizant of the required system flexibility. Our studies of axial temperature and concentration profiles above the fuel bed and the nature of the flow patterns in the combustion space have enabled us to estimate the degree of variability to be expected.

Accumulation of additional experimental data to quantify the behavior of jets at large distances from the nozzle mouth and under conditions where buoyant effects might be significant was beyond the scope of this study. We have attacked the buoyancy problem analytically in an attempt to define those sets of conditions where buoyant effects become important and have run some simple qualitative experiments to shed further light on the problem. The results of these studies are discussed in the following sections.

1. MATHEMATICAL MODELING OF COMBINED BUOYANT AND CROSSFLOW EFFECTS

Figure IV-8 defines the geometry on which our evaluation is based. This first step in analysis is concerned only with jet behavior very near the nozzle mouth. The objective is to establish the relative magnitude of the buoyant and drag forces acting on the jet.

The mathematical model is based on a segment of the jet dx in length. The buoyant force (B) on this differential volume element is:

$$dB = g(\rho_a - \rho_o) \left(\frac{\pi d_o^2}{4}\right) dx \qquad \text{(IV-25)}$$

The drag force (D) is computed by analogy with the way in which the drag on a solid cylinder is computed,

$$dD = (\frac{1}{2}\rho_a u_1^{\,2})\ C_x\ (d_o)(dx) \qquad \text{(IV-26)}$$

where C_x is the drag coefficient.

The ratio of these forces F is:

$$F = \frac{dB}{dD} = (\frac{\pi}{2C_x})\ (\frac{gd_o}{u_1^{\,2}})\ (\frac{\rho_o - \rho_a}{\rho_a}) \qquad \text{(IV-27)}$$

The significance of this relationship is that:

If F > 1, buoyant forces predominate (the cold jet sinks).

If F < 1, drag forces predominate (the jet is "blown away").

Comparison of this type of drag force model with experimentally measured deflections of jets in crossflow suggests that the value of C_x lies in the range of 3 to 5. Using Abramovich's correlation of the deflection data[20] yields $C_x = 4.75$. Therefore, Equation (IV-3) becomes:

$$F = 0.33\ (\frac{gd_o}{u_1^{\,2}})\ (\frac{\rho_o - \rho_a}{\rho_a}) \qquad \text{(IV-28)}$$

Equation (IV-28), with F = 1 defines the functional relationship between the crossflow velocity (u_1) and jet diameter (d_o) for given furnace gases and jet fluid properties for which buoyant and drag forces are equal. A plot of this function (Figure IV-14) divides the d_o-u_1 space into two areas of buoyant and drag domination. In Figure IV-14, the temperatures of the gas and jet fluids are taken to be 2000°F and 100°F, respectively. Differences in molecular weights are neglected.

Figure IV-14 shows, for example, that with a jet diameter of 3 inches, drag forces predominate if the crossflow velocity is greater than 3 feet per second. This plot serves as a rough design tool. The region to the left and above the line should be recognized as a design regime where the predominance of buoyant forces may cause the jet to "fall into" the bed and disturb the fuel distribution or create hot spots. As one moves away from the line down and to the right, drag forces become increasingly predominant and the design assumptions of Ivanov more applicable.

In an attempt to compute the trajectory of a jet subjected to both drag and buoyant forces, we expanded the simple model to include the effects of spreading and change in direction which the jet experiences in crossflow.

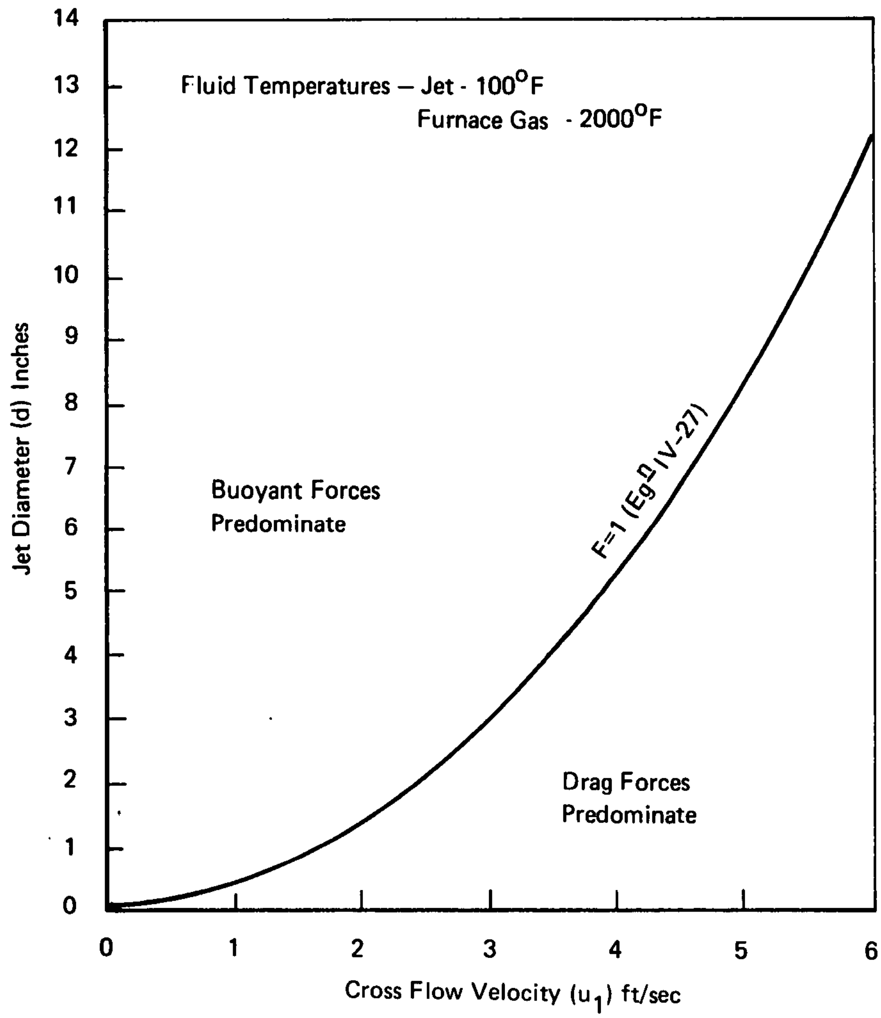


FIGURE IV–14 RELATIVE IMPORTANCE OF BUOYANCY AND DRAG

This mathematical model is based on conservation of momentum in the x and y directions. In the axial (x) direction, this amounts to simply the conservation of the momentum in the jet at the nozzle mouth. In the crossflow (y) direction, the effects of both buoyant and crossflow forces on the initial momentum are included.

The complete derivation of the model is given in Appendix E; the resulting trajectory of the jet centerline is given by:

$$\left(\frac{y}{d_\circ}\right) = \left(\frac{x}{d_\circ}\right) \tan \alpha_\circ + \left(\frac{2.25\ C_x}{\pi \cos\ \alpha_\circ}\right) \left(\frac{\rho_a u_1^2}{\rho_\circ u_\circ^2}\right) \left(\frac{x}{d_\circ}\right)^2 + \left[\left(\frac{c\ C\ \gamma}{3\pi\ \cos\alpha_\circ}\right) \left(\frac{\rho_a u_1^2}{\rho_\circ u_\circ^2}\right) - \left(\frac{\gamma^3}{19.2}\right) \left(\frac{gd_\circ}{u_\circ^2}\right) \left(\frac{\rho_\circ - \rho_a}{\rho_\circ}\right)\right] \left(\frac{x}{d_\circ}\right)^3 \qquad \text{(IV-29)}$$

where: C_x is the nominal drag coefficient relating crossflow drag on the jet to systems parameters.

c is the rate of spread of the jet in the lateral dimension (h): where $h = 2.25\ d + c\ell$.

ℓ is the path length along the jet axis.

γ is equal to sec β, where β is the angle between the jet centerline and the vertical; γ is equal to csc $\alpha_\circ$ at the nozzle mouth and increases without limit as the jet becomes parallel to the crossflow.

For the case where $\alpha_\circ = 0$ (the jet is introduced normal to the crossflowing stream), Equation (IV-29) reduces to:

$$\left(\frac{y}{d}\right) = \left(\frac{2.25\ C_x}{\pi}\right) \left(\frac{\rho_a u_1^2}{\rho_\circ u_\circ^2}\right) \left(\frac{x}{d_\circ}\right)^2 + \left[\left(\frac{c\ C_x\ \gamma}{3\pi}\right) \left(\frac{\rho_a u_1^2}{\rho_\circ u_\circ^2}\right) - \left(\frac{\gamma^3}{19.2}\right) \left(\frac{gd_\circ}{u_\circ^2}\right) \left(\frac{\rho_\circ - \rho_a}{\rho_\circ}\right)\right] \left(\frac{x}{d_\circ}\right)^3 \qquad \text{(IV-30a)}$$

In order to evaluate Equation (IV-30a) for a particular case, average values for γ, c and C_x are required. We are first concerned with behavior when the jet is close to the horizontal, in order to establish whether the drag or buoyant forces predominate. In this region, $\alpha \cong \alpha_\circ$, and $\gamma = 1$.

For a rectangular jet introduced into a quiescent volume, c = 0.22. Recent data[29,24] suggest that the spreading rate for a deflected jet is greater than for a straight jet. Based on these data, we estimate c = 0.32.

As before, we assume that $C_x = 4.75$.

Substitution of these values into Equation (IV-30a) yields:

$$(Y) = M\left\{[3.4X^2 + [(0.16) - 0.052 \left(\frac{gd_o}{u_1^2}\right)\left(\frac{\rho_o - \rho_a}{\rho_a}\right)]\ X^3]\right\} \qquad \text{(IV-31)}$$

where: $Y = y/d_o$
$X = x/d_o$
$M = \left(\frac{\rho_a u_1^2}{\rho_o \bar{u}_o^2}\right)$

This model does not allow for simple comparison of drag and buoyant forces due to the presence of both X^2 and X^3 terms. The coefficient of X^2 is always positive (since it arises from the crossflow drag term). The coefficient of X^3 may be either positive or negative depending on the value of the group $\left(\frac{gd_o}{u_1^2}\right)\left(\frac{\rho_o - \rho_a}{\rho_o}\right)$ which has the same form as occurred in the simple model. If the coefficient of X^3 is negative, the slope of the trajectory will become negative at sufficiently large values of X; the more negative the coefficient, the smaller the value of X at which this occurs.

The efficiency of the model is difficult to evaluate because of the assumptions inherent in the derivation and the empirical parameter relationships which are used. The drag force concept is somewhat artificial at best in this application, and is increasingly suspect as the jet is deflected and distorted in the crossflow. The empirical relation between jet spread and axial distance from the jet mouth is only applicable for some 6 or more jet diameters from the jet mouth. Experimental data is needed to test the model, and to serve as a basis for development of reliable design correlations.

2. QUANTITATIVE MODEL EXPERIMENTS

In order to shed light on the behavior of buoyant jets in crossflow, we conducted experiments using the apparatus shown in Figure D-1 in Appendix D. A jet of cold nitrogen gas at essentially the boiling point of nitrogen (139°R) was injected horizontally into the test section. Air at room temperature was blown upward in the test section so that the buoyant and crossflow forces on the jet were in opposition. Water vapor condensed when the cold jet mixed with the humid room air, allowing the jet path to be observed. A grid of strings was arranged on 6" centers to allow measurement of jet deflection. A more detailed discussion of the test equipment is given in Appendix D.

Runs were made using two nozzle diameters at several combinations of jet and crossflow velocities. Conditions for the runs are shown in Table IV-2, along with the calculated values of the buoyancy/drag force ratio [Equation (IV-28)].

TABLE IV-2

CONDITIONS FOR BUOYANCY-CROSSFLOW MODEL TESTS

Run No.	Nozzle Diam. (in)	Crossflow Velocity (fps)	Jet Velocity (fps)	F*
1	0.469	1.31	40	0.66
2	0.469	1.31	8.1	0.66
3	1.063	1.31	2.5	1.49
4	1.063	1.31	1.3	1.49
5	1.063	2.10	2.5	0.58

* Force ratio as defined in Equation (IV-4)

Figures IV-15 to IV-19 show the jet trajectory for each of the runs. In runs 1, 2, and 5, in which the buoyancy/drag ratio was less than unity, the jet shows no tendency to drop. In runs 3 and 4, where the ratio was about 1.5, buoyancy forces cause the jet to drop. While these tests are not sufficiently comprehensive to prove the validity of the force ratio parameter in determining when buoyancy effects are important, the data do support the contention that $F \gtrsim 1$ is a valid criterion to identify conditions when jet sinking can be anticipated.

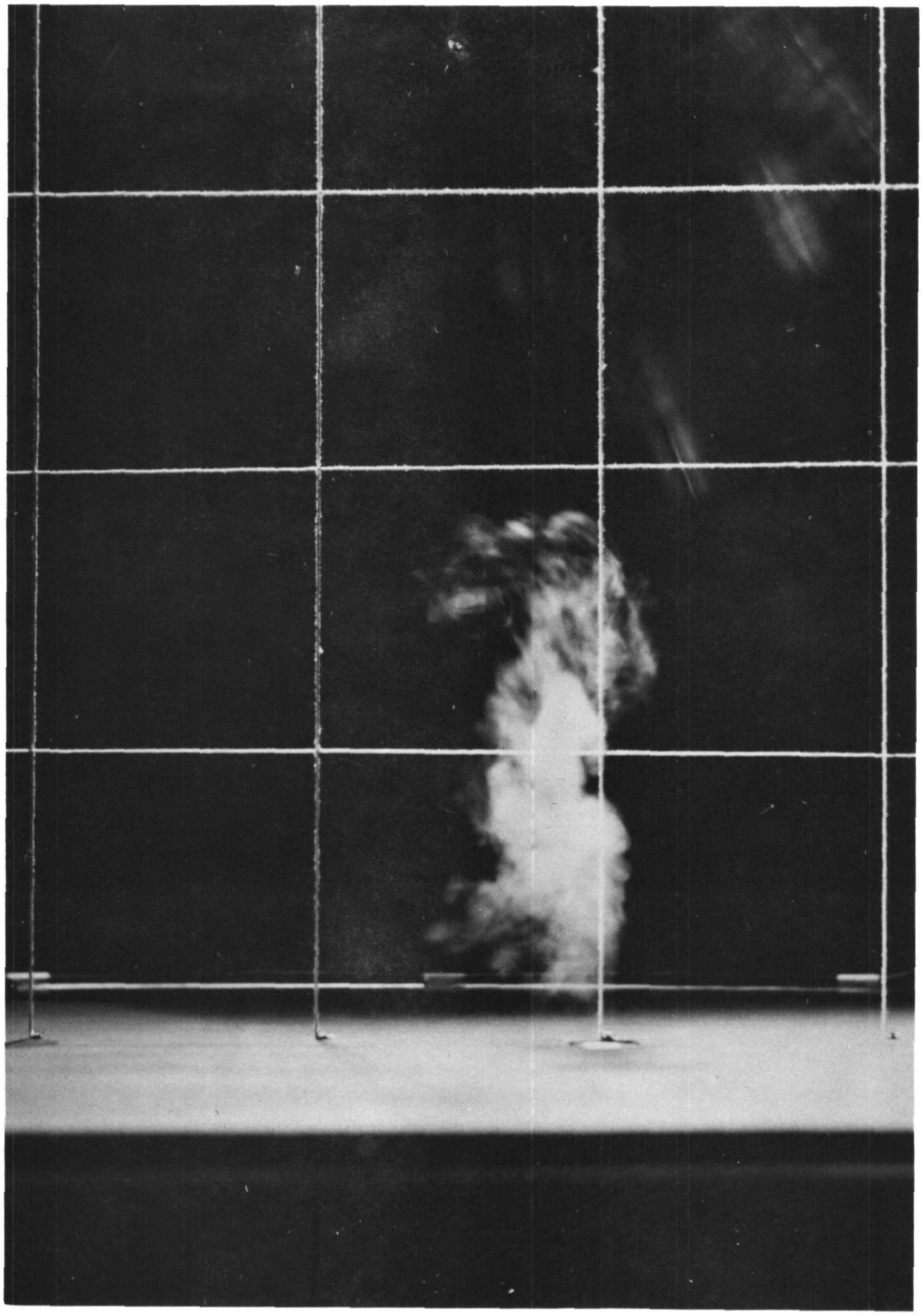


FIGURE IV–15 PHOTOGRAPH SHOWING OBSERVATIONS FOR TEST 1

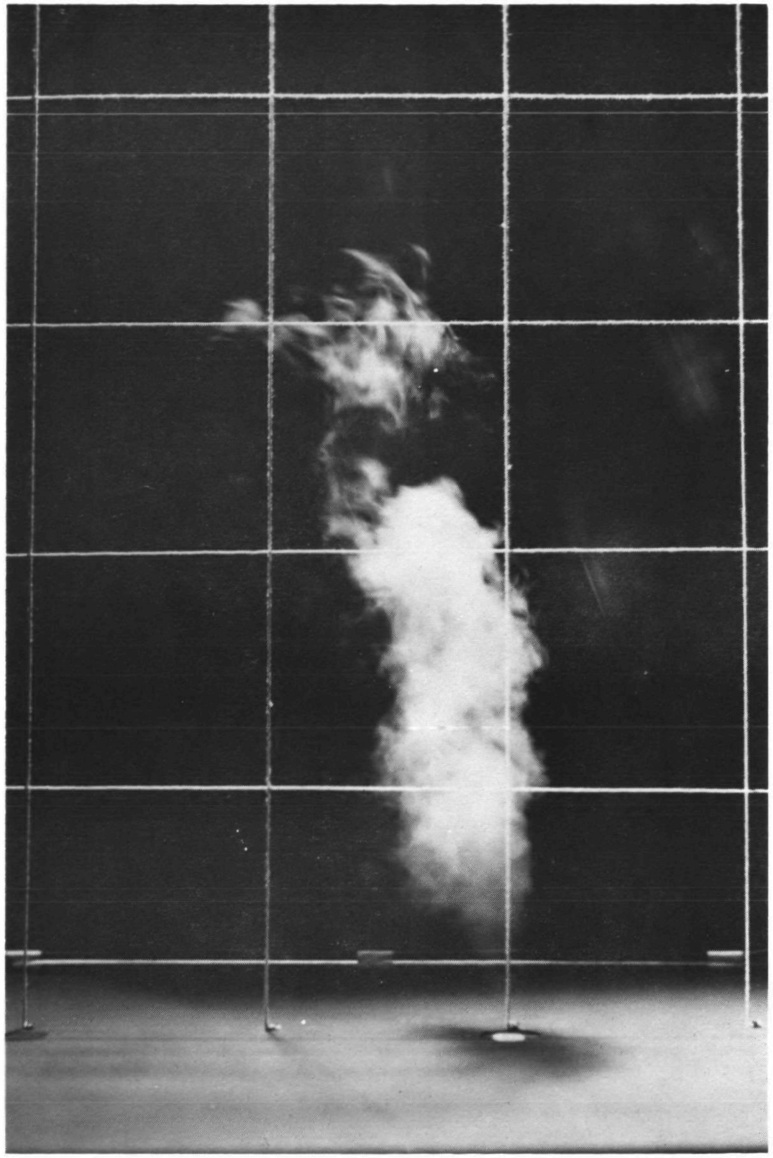


FIGURE IV–16 PHOTOGRAPH SHOWING OBSERVATIONS FOR TEST 2

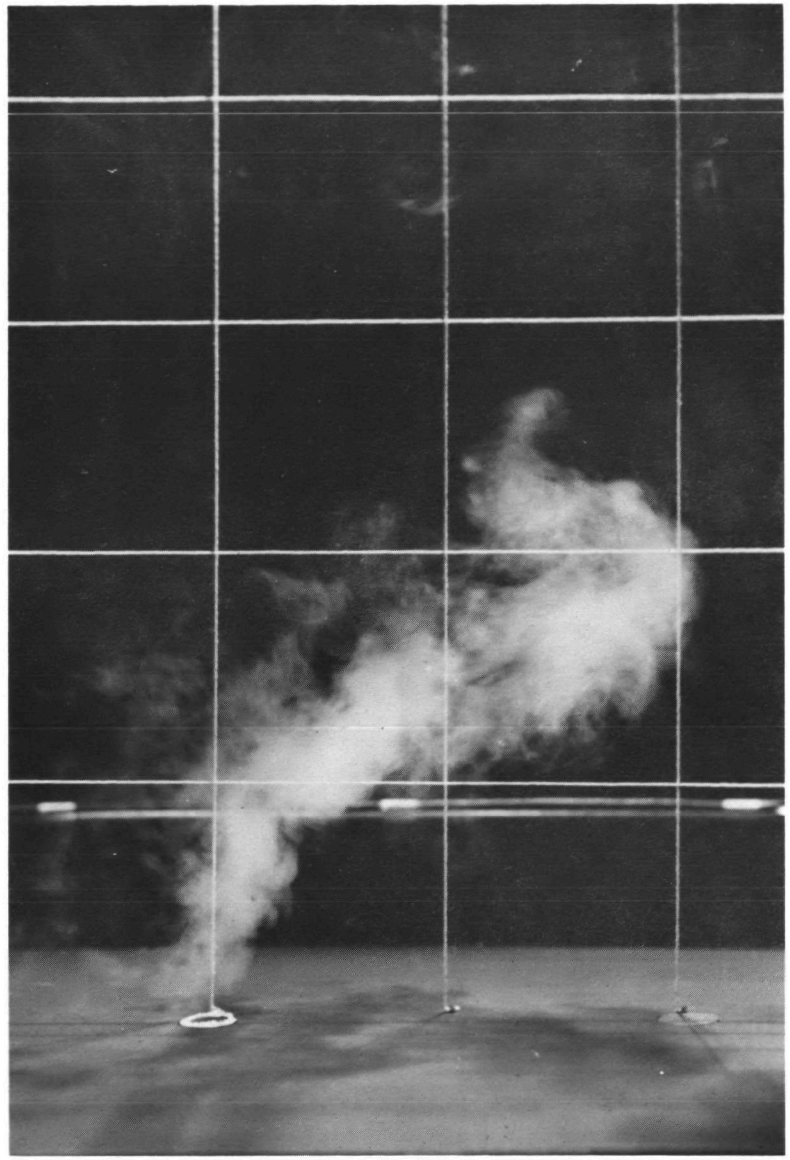


FIGURE IV–17 PHOTOGRAPH SHOWING OBSERVATIONS FOR TEST 3

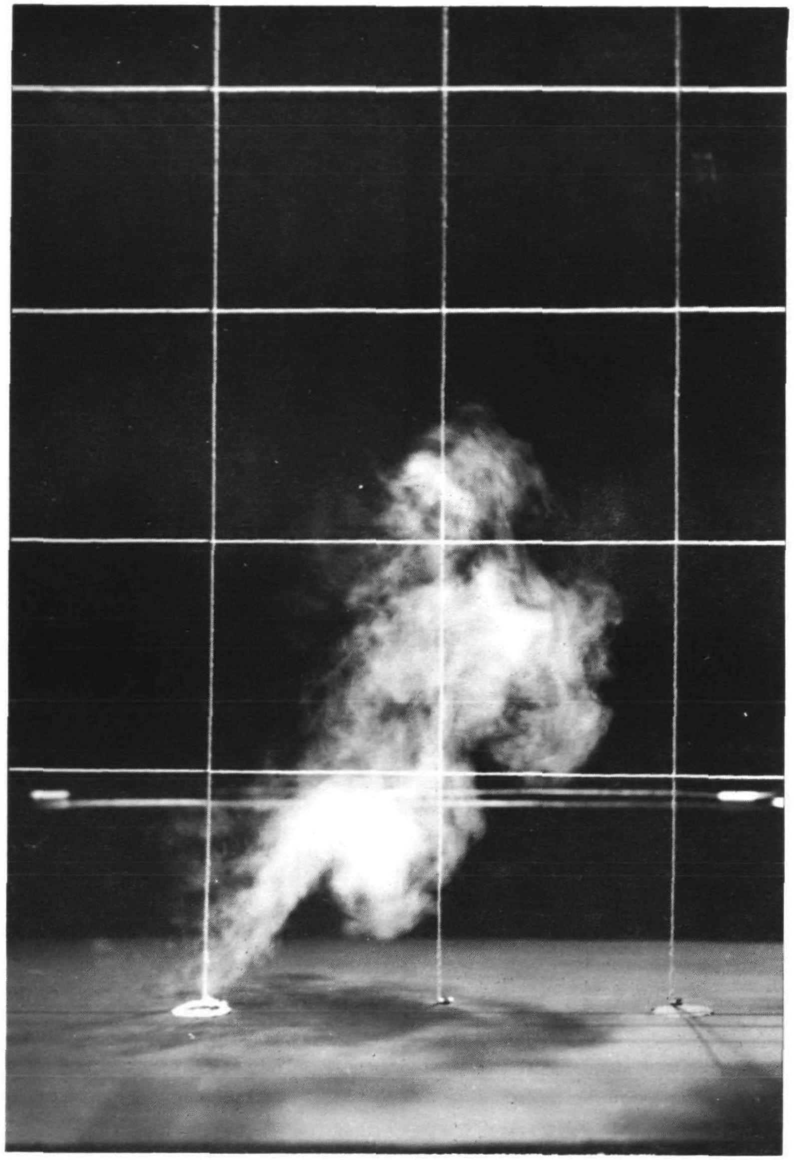


FIGURE IV–18 PHOTOGRAPH SHOWING OBSERVATIONS FOR TEST 4

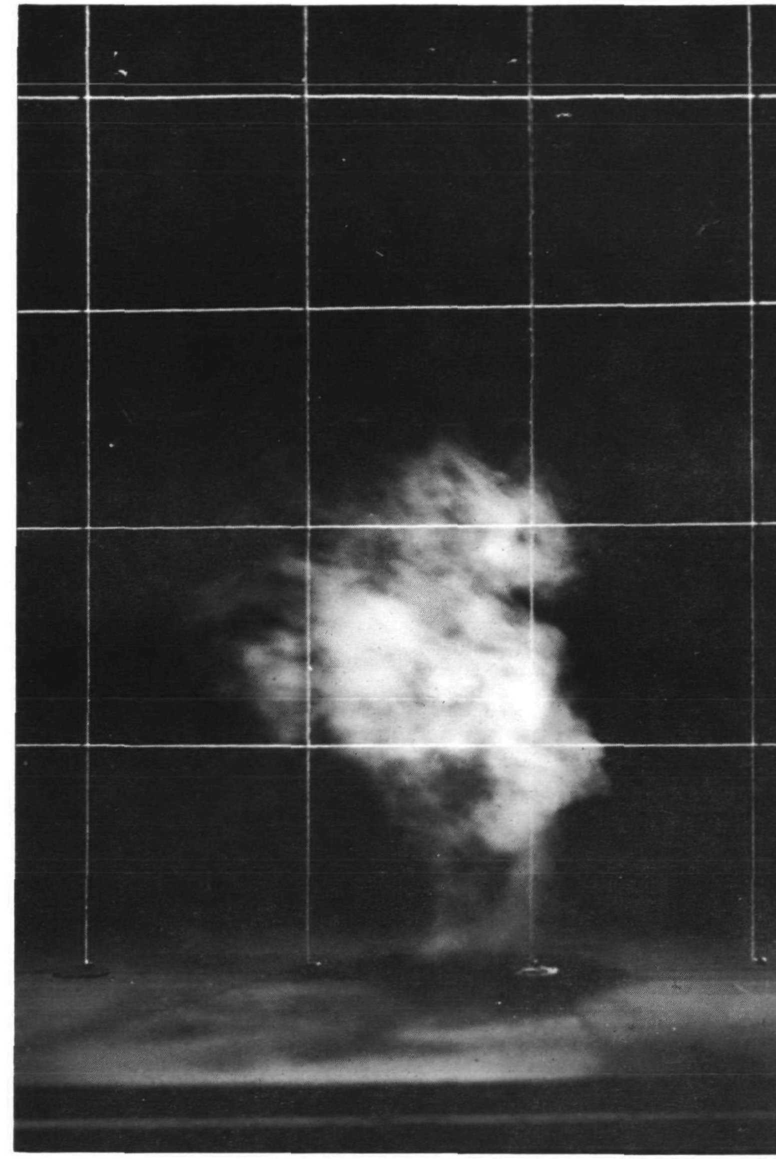


FIGURE IV–19 PHOTOGRAPH SHOWING OBSERVATIONS FOR TEST 5

3. COMBUSTION EFFECTS

The analysis in Chapter II indicated that under all conditions the flow of air through a refuse or char bed will produce gases containing unburned combustible. Although bypassing (channeling) of air through the bed can provide some of the needed oxidant to the fuel-rich gases, the data by Kaiser referenced in Chapter II indicate that the gases, on the average, remain fuel-rich. As a consequence, it should be expected that some combustion will occur as the oxygen-bearing overfire air jet penetrates the hot gas flow rising from the bed. This results in a so-called "inverted flame" where a jet of oxidant discharges and burns in a fuel-rich environment.

This phenomenon has been observed in both coal- and refuse-burning practice where impingement of a jet on the opposite sidewall has resulted in refractory overheating and slagging contributing to premature wall failure. In another case, a jet of air moving beneath a long arch over the discharge grate of an incinerator furnace yielded temperatures of over 2500°F in the brickwork. Because of the potential importance of this combustion effect, a simplified mathematical model of jet behavior under these conditions was developed and the effect of the pertinent variables was explored.

The analysis makes use of jet concentration correlations describing the axial ($\bar{c}_m$) and radial ($\bar{c}$) weight concentration of nozzle fluid as functions of the distance from the nozzle plane (x) and the radial dimension (r). The concentration at the nozzle is $\bar{c}_o$ and the nozzle diameter is d_o. The ambient (T_1), nozzle fluid (T_o), and mixture (T_m) temperatures are those prior to combustion. For non-combusting jets, in the absence of crossflow and buoyancy, these variables are related by:

$$\bar{c}_m = 5\,\bar{c}_o \left(\frac{T_1}{T_o}\right)^{1/2} \left(\frac{d_o}{x}\right) \qquad \text{(IV-32)}$$

$$\bar{c} = \bar{c}_m \exp\left(-57.5 \left[\frac{r}{x}\right]^2\right) \qquad \text{(IV-33)}$$

Assuming equal and constant specific heats for the nozzle and ambient fluid, an energy balance yields:

$$T_m = T_1 + 5 \left(\frac{T_1}{T_o}\right)^{1/2} (T_o - T_1) \frac{d_o}{x} \exp\left(-57.5 \left[\frac{r}{x}\right]^2\right) \qquad \text{(IV-34)}$$

Using Equation (IV-34), we can calculate the mixture temperature. Then, by comparison with an assumed minimum ignition temperature (T_F) (say 1100°F), it can be determined whether or not combustion will occur. (Note that this shows "quenching" of combustion in the cold core of the jet.)

From the analysis method described in Chapter II, the oxygen demand (Λ pounds of oxygen per pound of ambient fluid) and the heat of combustion (H_c Btu per pound of oxygen reacting) of the furnace gases can be estimated. Defining Ω as the concentration of nozzle fluid in the mixture relative to the nozzle concentration ($\bar{c}/\bar{c}_o$), we find that if $T_m \geq T_F$ and if Λ $(1-\Omega)$ $-\Omega\ \bar{c}_o \geq 0$, combustion will occur to the extent of the available oxygen releasing $H_o \Omega \bar{c}_o / \Lambda$ Btu per pound of mixture. The resulting gas temperature (T_c) is given by:

$$T_c = T_R + \frac{1}{cp} \left[(1 - \Omega)\ (T_1 - T_R)\ cp + Hy\bar{c}_o/\Lambda \right] \qquad \text{(IV-35)}$$

Where T_R = the reference temperature for enthalpy (say 60°F) and cp = the average specific heat of the gases between the reference temperature and T_c.

For the oxygen-rich case, if $T_m \geq T_F$ and if $\Lambda(1 - \Omega) - \Omega c_o < 0$, combustion will occur to the extent of the available fuel releasing H_c $(1 - \Omega)$ Btu per pound of mixture. The resulting gas temperature is given by:

$$T_c = T_R + \frac{1}{cp} \left[(1 - \Omega)\ (T_1 - T_R)\ cp + H_c\ (1 - \Omega) \right] \qquad \text{(IV-36)}$$

Calculation of the radial profiles of temperature according to the above was carried out and the results plotted in Figures IV-20 to IV-22 at various distances from the nozzle plane. The depressed temperature (<1100°F) near the discharge plane is the sub-ignition temperature or "quenched" region. The peaks of temperature found along a radial temperature "traverse" identify the stoichiometric point. The peak temperature (about 2570°F) reflects the assumptions:

T_1 = 2460°R (2000°F)

H_c = 370 Btu/lb ambient

Λ = 0.0545 lb O_2/lb ambient

$\bar{c}_o$ = 0.23 lb O_2/lb nozzle fluid (air)

cp = 0.31 Btu/lb °F relative to 60°F

Note that only absolute temperatures are to be substituted in the equations.

It can be seen that a hot zone is rapidly developed with a diameter of about one foot and that this zone endures, for large jets, for distances which approximate the widths of many incinerator furnaces (6-10 feet). This problem is greatly reduced as the jet diameter is decreased, thus adding additional incentive to the use of small-diameter jets.

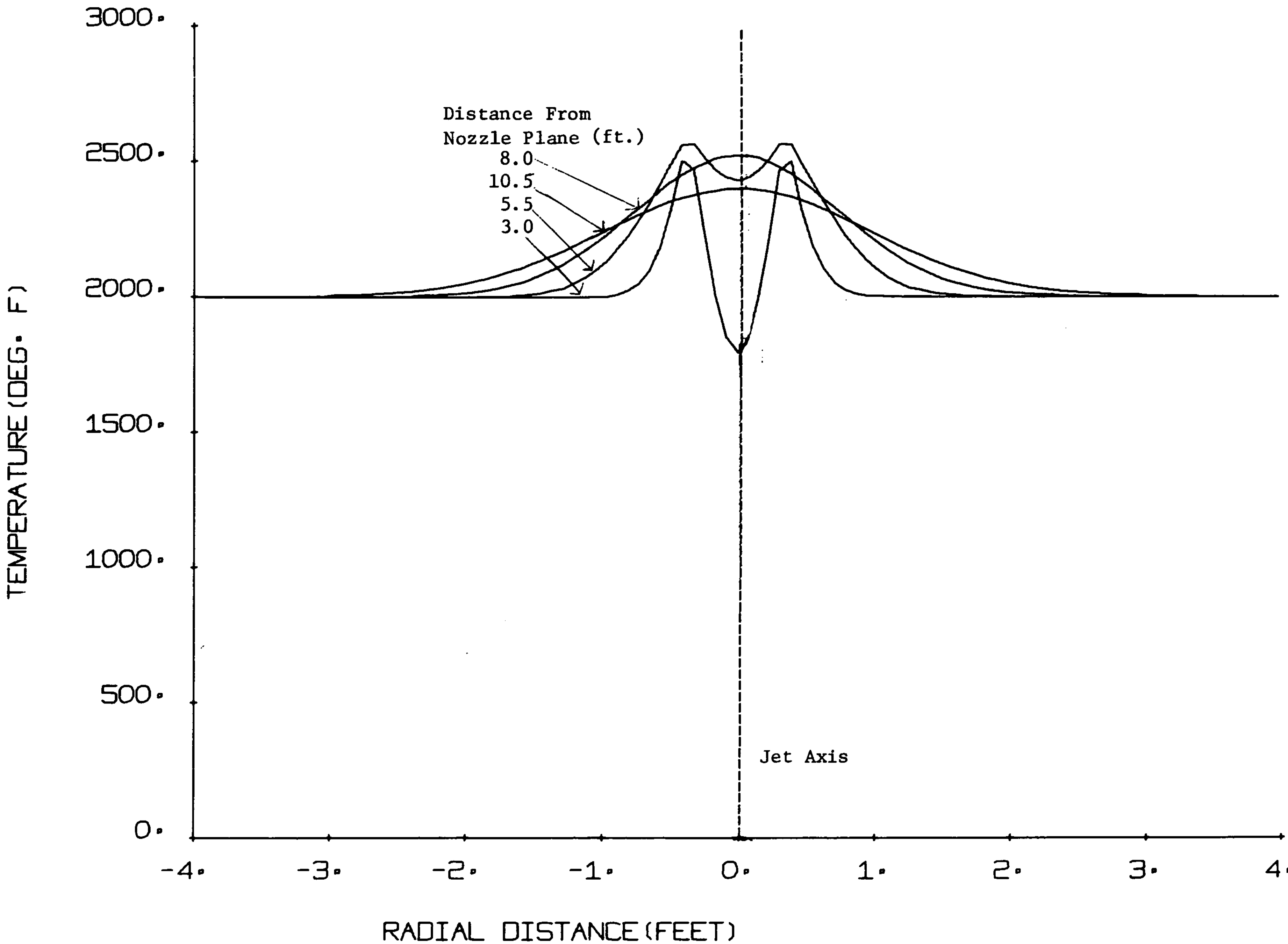


FIGURE IV-20 TEMPERATURE VERSUS RADIAL DISTANCE (Jet Diameter = 1.5")

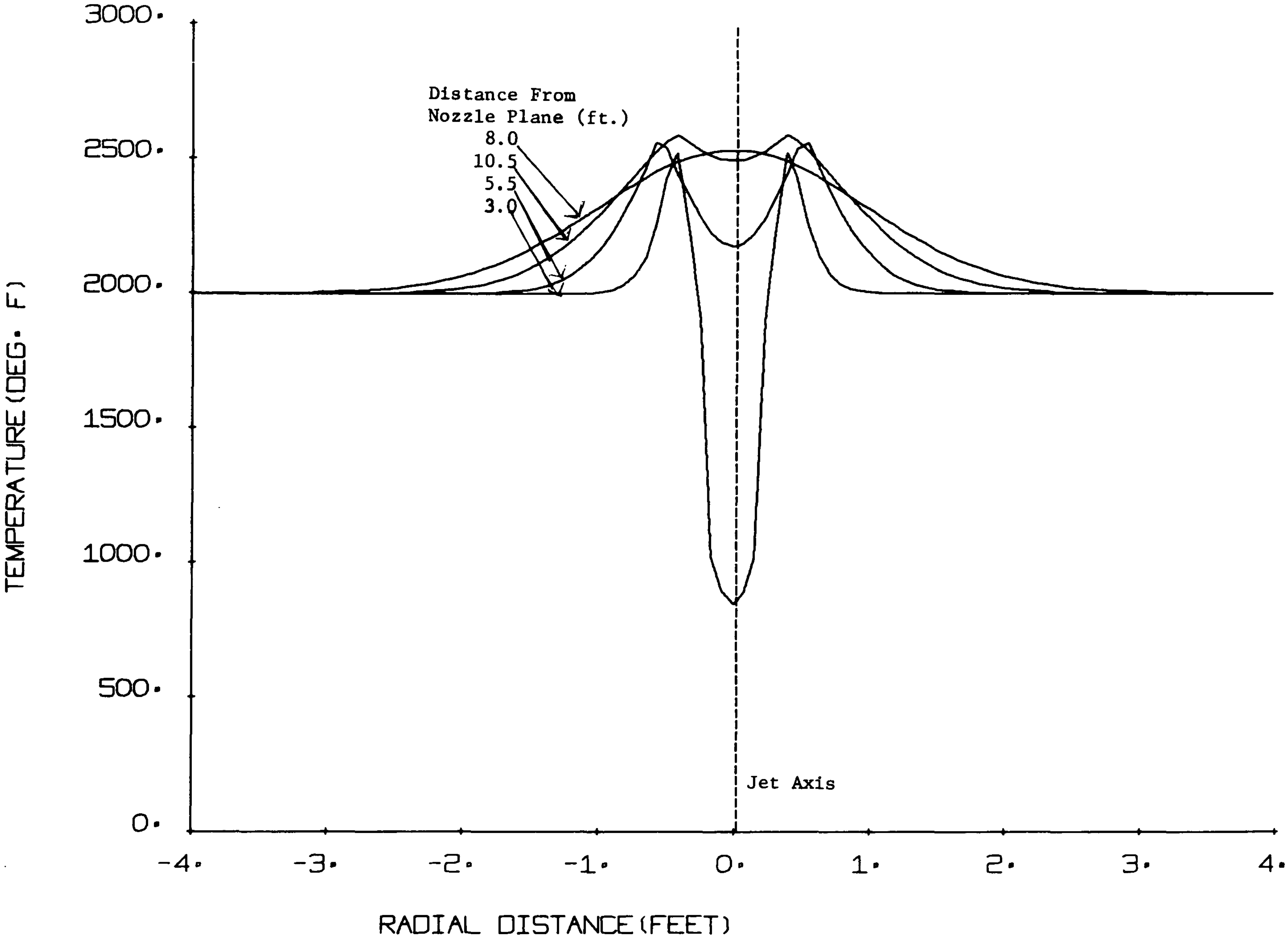


FIGURE IV–21 TEMPERATURE VERSUS RADIAL DISTANCE (Jet Diameter = 2")

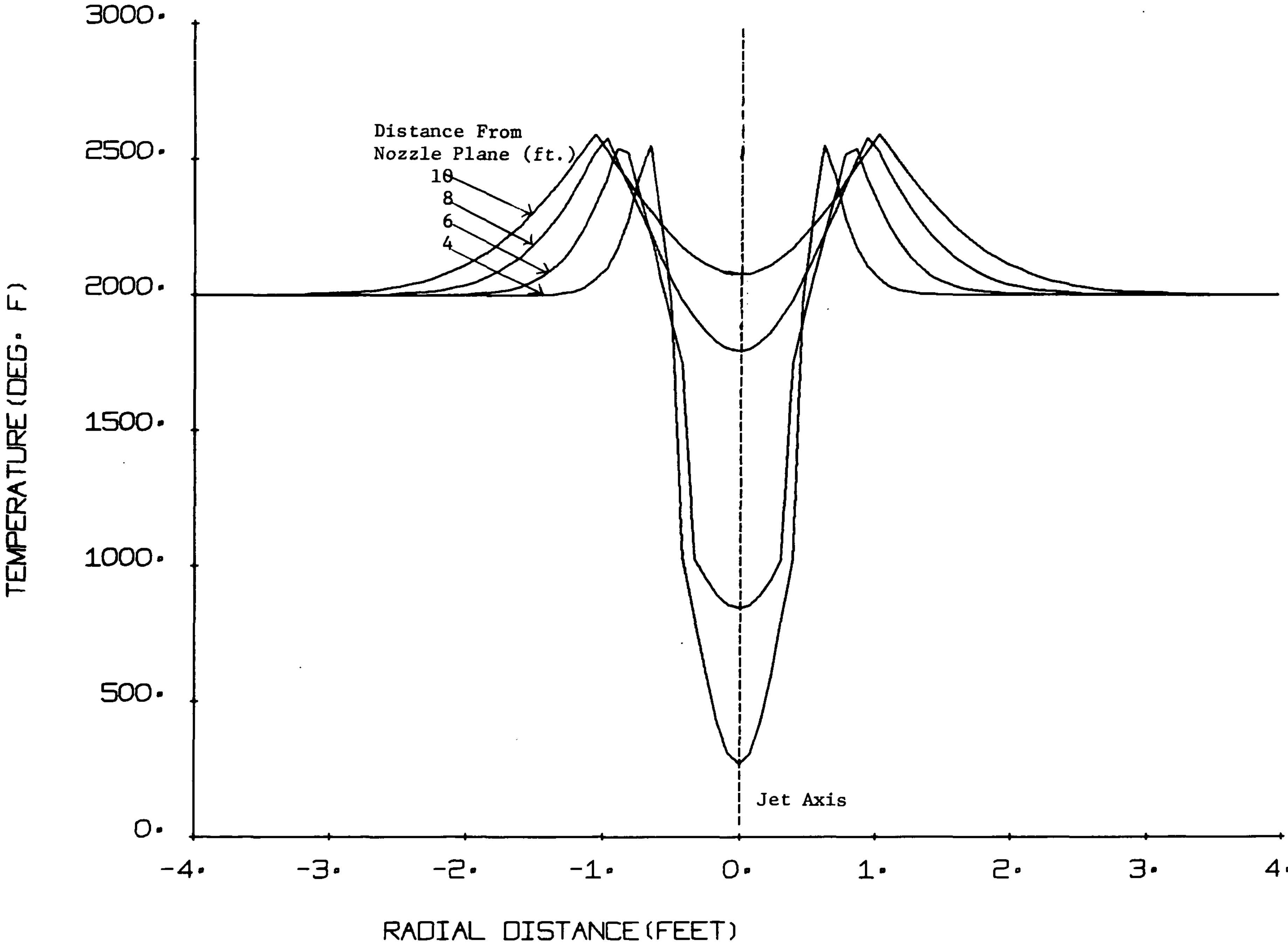


FIGURE IV–22 TEMPERATURE VERSUS RADIAL DISTANCE (Jet Diameter = 4")

The above provides confirmation of the occurrence of "blow torch" effects from air jets and shows that temperatures near 2500°F such as have been experienced with jets could be anticipated. Clearly, however, the utility of the above is only to provide perspective as to the nature of air jet behavior in incinerators. No allowance is made, for example, of crossflow effects which are known to increase jet entrainment rates and thus "shorten" the inverted flame described by the analysis. The method, therefore, can be expected to produce a conservative result.

The correlations of axial concentration by Patrick[24] provide a means to estimate the effect of crossflow in shortening the distance to the point of completion of the combustion reactions. From Equation (IV-32) an analysis readily shows the distance from the nozzle plane to the point where the gas on the axis is at a stoichiometric ratio to be given by:

$$\frac{x}{d_o} = 5 \left(\frac{T_1}{T_o}\right)^{1/2} \left(\frac{\bar{c}_o}{\Lambda} + 1\right) \qquad \text{(IV-37)}$$

Patrick found the centerline concentration to vary along the jet path length "ℓ" in crossflow according to:

$$\frac{\bar{c}_o}{\bar{c}_m} = \left[\left(\frac{\ell}{d_o}\right) \exp\left(7.8\, M^{1/2} - 1.856\right)\right]^{1.18} \qquad \text{(IV-38)}$$

and, for no crossflow according to

$$\frac{\bar{c}_o}{\bar{c}_m} = 0.112 \left(\frac{x}{d_o}\right)^{1.18} \qquad \text{(IV-39)}$$

Equating (IV-38) and (IV-39) establishes the relationship between the centerline distance "x" for non-crossflow which corresponds to the same concentration ratio as for a jet in crossflow which has traveled over a path length "s."

$$\frac{x}{d_o} = 1.42 \left(\frac{\ell}{d_o}\right) \exp\left(7.8\, M^{1/2} - 1.856\right) \qquad \text{(IV-40)}$$

The path length can be easily calculated by numerical integration of Equation (IV-41) which also can be derived from Patrick's trajectory relationships:

$$\left(\frac{\ell}{d_o}\right) = \int_o^{x/d_o} \left[9\, M^{2.55} \left(\frac{x}{d_o}\right)^4 + 1\right]^{0.5} d\left(\frac{x}{d_o}\right) \qquad \text{(IV-41)}$$

Therefore, to find the distance from the nozzle plane to the point where such peak (stoichiometric) temperatures will be obtained on the centerline, the non-crossflow distance is calculated from Equation (IV-37); the resulting value is substituted into Equation (IV-40) to yield the crossflow path length at an equivalent degree of mixing; and the integration given in Equation (IV-41) is carried out to define the dimensionless distance x/d integration limit which causes the integral to assume the value of the calculated path length. This latter x/d value corresponds to the horizontal distance from the nozzle to the plane where peak temperatures exist. The vertical displacement of the jet in this plane may then be calculated by substitution into Equation (IV-42).

$$\left(\frac{y}{d_o}\right) = M^{1.28} \left(\frac{x}{d_o}\right)^3 \qquad \text{(IV-42)}$$

The analysis shows that jet temperatures can be considerably elevated by combustion effects. Therefore, when jet operation is desired in regimes where buoyancy analysis (neglecting combustion) suggest jet drop would be important, these effects could provide counterbalancing jet temperature increases. The complete analysis suggesting the degree to which the buoyancy/drag criteria could be slackened by consideration of the combined effects of buoyancy and crossflow with combustion, however, was beyond the scope of this analysis effort.

4. TENTATIVE INCINERATOR OVERFIRE AIR JET DESIGN METHOD

Although little data is available to give specific guidance for incinerator overfire air jet design, the prior art and the studies carried out under this contract provide the basis for tentative guidelines and a design methodology. Experiments anticipated in subsequent phases of this program (Chapter VI) will be helpful in strengthening these arguments.

The basic parameters to be selected in design of an overfire air jet system are:

- The diameter (d_o) and number (N) of the jets to be used:
- The placement of the jets;
- The quantity of air to be overfired (Q_T)

Related but not independent variables are the jet velocity ($\bar{u}_o$), and the head requirements for the overfire air fan (P).

The tentative design method is based on that of Ivanov[17] which was discussed in Section C-5. It is important in using this method to compare the values of d_o and u_1 obtained with Figure IV-14 to determine if buoyant forces might be important. Values should fall in the "drag forces predominate" region.

The basic equations on which the design method is based are as follows:

Air Flow Relation

$$Q_T = 47\ Nd_o^2\ \overline{u}_o \qquad \text{(IV-43)}$$

where: Q_T is the overfire air rate (CFM)

N is the number of jets

d_o is the jet diameter (feet)

$\overline{u}_o$ is the jet velocity (fps)

Ivanov's Penetration Equation

$$\frac{L_j}{d_o} = 1.6\ (\frac{\overline{u}_o}{u_1}) \sqrt{\frac{\rho_o}{\rho_1}} \qquad \text{(IV-44)}$$

where: L_j is the desired jet penetration (feet)

u_1 is the estimated crossflow velocity in the incinerator (fps) calculated by the methods of Chapters II and III

ρ_o and ρ_1 are the jet and crossflow densities, respectively.

Jet Spacing Equation

$$L + SNd_o \qquad \text{(IV-45)}$$

where: L is the length of furnace wall on which the jets are to be placed (feet), and

S is the desired value of jet spacing measured in jet diameters.

Inherent in Equation (IV-45) is the assumption that the jets are placed in a single line. Ivanov recommends that S be in the range of 4 to 5, although values as low as 3 can probably be used without invalidating the penetration equation [Equation (IV-44)].

The form of these equations sheds some light on the options open to the systems designer. For a given set of furnace conditions, Equation (IV-44) can be rearranged to give:

$$d_o\overline{u}_o = \text{constant} = B_1 \qquad \text{(IV-46)}$$

The product of jet diameter and velocity is fixed by furnace conditions.

Substitution of Equations (IV-45) and (IV-46) into Equation (IV-43) yields:

$$Q_T/L = \frac{0.326B_1}{S} \qquad \text{(IV-47)}$$

The air flow per length of wall is fixed by furnace conditions, except inasmuch as S can vary from 3 to 5.

The design method can best be illustrated by example. Consider an incinerator with the following properties:

Capacity - 250 TPD
Total Stoichiometric Air Req. - 15,000 cfm
Length of Wall for Jet Placement (L) - 15 ft
Desired Depth of Penetration (L_1) - 8 ft
Crossflow Velocity (u_1) - 4 fps

Jet and furnace temperatures of 100°F (560°R) and 2000°F (2460°R), respectively, so that:

$$\sqrt{\frac{\rho_o}{\rho_a}} = \sqrt{\frac{T_1}{T_o}} = \sqrt{\frac{2460}{560}} = 2.10$$

Step 1. Compute the product $d_o\bar{u}_o$ from Equation (IV-44).

$$d_o\bar{u}_o = hu_1 \left(\frac{1}{1.6}\right) \frac{\rho_a}{\rho_o} \qquad \text{(IV-48)}$$

$$= \frac{(96)(4)}{(1.6)(2.10)} = 171.4$$

Step 2. Select value of d_o.

Figure IV-14 shows that for u_1 = 4 fps, any value of d_o below about 5 inches will allow drag forces to predominate over buoyant forces. Select d_o = 3 inches.

Step 3. Compute $\bar{u}_o$ from Equation (IV-48).

$$\bar{u}_o = \frac{171.4}{3} = 57.1 \text{ fps}$$

The pressure requirement for the overfire air fan depends mainly on $\bar{u}_o$. Equation (IV-49) may be used to calculate the required velocity head (H) as a function of $\bar{u}_o$.

$$H = 1.2 \left[\frac{\rho_o \bar{u}_o^2}{2}\right] \qquad \text{(IV-49)}$$

For $\bar{u}_o$ = 57.1 fps, the required head is about 1 inch of water plus ducting and nozzle losses.

Step 4. Compute N from Equation (IV-45).

Setting S = 4, Equation (IV-45) yields:

$$N = \frac{180}{(4)(3)} = 15$$

Step 5. Compute Q_T from Equation (IV-43).

$$Q_T = (0.327)(15)(3)^3(57.1) = 2520 \text{ cfm}$$

Step 6. Compare Q_T with the theoretical air requirement.

The theoretical air requirement is 15,000 cfm, so that the overfire air requirement is [(2520/15,000)](100) = 17% of theoretical. The soundness of this value can be checked by comparison with the air requirements defined by the bed burning process (Chapter II) and by reference to experience. It is worthy to note, however, that few data exist to allow confident valuation of the performance of existing plants with respect to combustible pollutant emissions and the design and operating parameters of the overfire air systems.

The amount of overfire air can be increased or decreased within limits without seriously affecting the performance of the jets by changing the jet spacing parameter ("S" in Equation (IV-45) within the range of 3 to 5; by modification of the assumed value for "L"; or by using opposed jet placement which prevents impingement of jets on the opposite wall.

E. SUMMARY

The behavior of cold, oxygen-rich jets in a crossflow of hot, fuel-rich furnace gases is not perfectly understood. Basic experimental and theoretical work showing the interactions of buoyant and crossflow effects in the presence of combustion is lacking. However, our experimental and analytical work, together with previous work reported in the literature, allows a semi-quantitative analysis of these effects which provides a basis for the design of overfire jet systems.

Specifically, the prior art and the original work described above support the following design objectives:

1. Determination of the conditions where buoyancy effects are important relative to crossflow effects: Using the criterion developed, jets can be designed to pass close above the refuse bed without danger of disturbing the bed.

2. Approximation of the temperature profiles in a combusting jet as a function of jet parameters and axial distance from the nozzle mouth: These profiles show how jet design influences the probability of impinging the opposite wall with a hot jet.

3. Presentation of mathematical correlations and behavioral descriptions of jet characteristics under conditions to be expected in incinerators (drawing on the results presented in Chapters II and III to describe the furnace environment): With this understanding, the jet designer can gain insight and anticipate interactions between the jet and the furnace.

4. Presentation of a procedure for designing overfire jet systems: This procedure incorporates the criteria listed under 1. and 2. above, both of which mitigate toward small diameter jets, with the general procedure of Ivanov.

The analysis methods for jet behavior presented in this chapter, when coupled with the results of bed and flow analysis, give the incinerator designer a new and powerful tool for the control of combustible pollutant emissions. He can now estimate how much air is needed, where, and the extent to which the incinerator flow will perturb the jet behavior. To be sure, the correlation between realized performance (reduced combustible emissions) and optimum placement/design/operation of jets is not known. It is reasonable to assume, however, that some measure of improvement must be associated with more strategic jet operation.

Beyond the correlations and design approaches presented here, it can be seen that these analytical tools provide the basis for control systems for incinerators. Knowledge of the relationships between air demand, refuse properties and burnout is the cornerstone of control logic. Also, understanding of the furnace dynamics assists in the placement of instrumentation and control sensors and in the interpretation of their output.

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CHAPTER V

DESIGN METHODS FOR INCINERATOR OVERFIRE AIR SYSTEMS

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DESIGN METHODS FOR INCINERATOR OVERFIRE AIR SYSTEMS

In the previous chapter, it was cautioned (Section IV-C-5) that use of the "rule-of-thumb" approaches presented there for the design of incinerator overfire air jet systems could result in the installation of ineffective devices. Uncertainty in this matter arises primarily from the unknown risk arising from the use in refuse-burning applications of generalizations developed from coal-burning experience. Differences between refuse and coal (as reflected in heat release distributions, typical furnace shapes used, fuel chemistry, bed uniformity and so forth) could be expected to influence the desired design characteristics of the overfire air systems.

This chapter presents an illustration of a more tailored design approach which draws heavily upon the analysis methodology presented in Chapters II, III, and IV. The assumptions explicitly and implicitly incorporated into these analysis methods are admittedly speculative and they require experimental verification. It would appear, however, that development of a design approach along lines which allow input of parameters unique to the system and refuse in question would provide the vehicle for better new plant design optimization; for more rapid resolution of operating problems; and for the evaluation of alternatives in plant upgrading.

A. STATEMENT OF THE PROBLEM

It is required to evaluate the design requirements of the overfire air system for a triple traveling grate boiler-type incinerator furnace with a capacity of 250 tons per 24-hour day. Each grate is approximately 15-feet long. The furnace is 8-feet wide. Its general configuration is shown in Figure III-1. Because of the use of silicon carbide sidewall construction along the grate line, the air will be introduced 3 feet above the top of the refuse bed. Experience has shown that an average refuse residence time of 45 minutes will be required under most circumstances, resulting in an average grate speed of 60 feet per hour (1 foot per minute) and an average initial grate loading of 40 pounds of refuse per square foot.

The refuse to be burned in the unit ranges in free-moisture content between 0 and 30%. ("n" varies from 5/6 to 1.75 in the "refuse compound" $C(H_2O)_n$ discussed in Chapter II.)

B. EVALUATION OF SYSTEM OPERATING CHARACTERISTICS

1. BED PROCESSES

In analyzing the bed processes according to the methods presented in Chapter II, it is necessary to specify the carbonization characteristics of the refuse (the fraction gasified), estimate the temperature of the gases leaving the bed, and estimate the heat losses or gains experienced across the plane at the top of the bed plane. Based on the observations of workers in the field and some pyrolysis data, let us assume that 80% of the carbon is gasified. Further, let us assume that the gases leave the bed at 2000°F and that the average heat loss by the bed to the waterwalls is 56,000 Btu/mole of oxygen passing through the grate. (It should be noted that, although these values were assumed for the trial calculations below, a broader range of values should be tested in actual system designs to explore more fully the spectrum of possible situations.)

Using the technique of analysis presented in Chapter II, the gas compositions leaving the bed can be calculated. Using an assumed distribution of undergrate air flow rates (e.g., Table V-1), the flow rate of each gaseous compound (CO, CO_2, O_2, N_2, H_2O) entering the furnace volume can then be calculated for the gasification and char burnout regions. Further, if some estimated fraction of the total undergrate air flow is assumed to bypass the bed due to channeling affects (Table V-2), the effect of this secondary air on the combustion gas composition and temperature can be calculated. These composition estimates may then be used to identify the oxygen (overfire air) requirement to effect gas-phase burnout.

For these calculations, the equilibrium constants for the water-gas shift and the carbon-CO-CO_2 reaction at a temperature T (°R) may be calculated from the following relationships:

a. Water-Gas Shift Equilibrium

$$kw = \frac{[CO_2][H_2]}{[H_2O][CO]}$$

$$\log_{10} kw = \frac{3000}{T} - 1.587$$

b. CO-CO_2 Equilibrium

$$kc = \frac{[CO_2]}{[CO]^2} \text{ atm}^{-1}$$

$$\log_{10} kc = \frac{15169}{T} - 8.821$$

The calculation technique also allows estimates to be made of the length of the gasification and char burnout zones. This is accomplished by consideration of the initial quantity of refuse on the grate and the mass loss rate per

TABLE V-1

TOTAL UNDERGRATE AIR FLOW RATES
(scfm/square foot of grate)

	Grate		
Case	1	2	3
A	10	50	25
B	10	70	25
C	10	100	25

TABLE V-2

FRACTION OF TOTAL UNDERGRATE AIR BYPASSING BED

Grate	% ByPassing
1	10
2	15
3	20

foot of grate travel due to gasification of char arising from the undergrate air flow. [Note that the results illustrated in Table II-4 are in quantities (Btu, moles, etc.) per mole of oxygen passing through the grate.] Such calculations will show the effects of refuse composition (specifically inerts and moisture) and undergrate air flow on residue burn-out. Alternatively, by holding burnout constant, the variation of system capacity (tons per day) with these parameters could be estimated.

The results of a series of calculations using the parameters given in Tables V-1 and V-2 are shown in Table V-3. The composition of the gases over the char region on grates 2 and 3 are given in Table V-4.

The strong relationship between gas composition, overfire air needs along the grate, refuse characteristics and undergrate air flow can be seen readily. Also, since gas temperatures, especially in the char region, can be considerably elevated above those normally considered "safe" to prevent slagging, it may be desirable to estimate the quantity of overfire air needed to meet both the combustion requirements of the off-gases and also to cool the gas mixture.

Although it must be emphasized that results of the type shown in Tables V-3 and V-4 are not rigorous, they will be helpful to identify conditions to be expected and the system response to refuse and operating practice changes. For example:

- Dry refuse yields more CO in the pyrolysis region than wet refuse;

- More CO is emitted in the char-burning zone than elsewhere in the furnace and will present a large overfire air demand; and

- The underfire air requirement to achieve burnout increases as the refuse moisture content increases.

Changes in the heat loss or gain term in the analysis, based on estimates of the radiative heat transfer within the system, would allow evaluation of the effect of uncooled combustion chambers (refractory systems). Also, the effects of differential grate speeds and other alternatives in undergrate air flow should be readily explored. It should be noted that although the computations are straightforward, they rapidly become tedious and a simple computer program was prepared to produce the values shown.

The calculation results of interest in overfire air jet design are primarily the air requirements shown for the gases leaving each of the grates in the gasification and char burnout zones. The air requirements in Table V-3 are given in standard (70°F, 1 atm) cubic feet per minute per square foot of grate area. Therefore, the total overfire air requirements can be calculated in view of the 8-foot furnace width and the calculated zone length. In trial 2, for example, a minimum of 28 to 36 scfm are required over the last half of the first grate, from 160 to 240 scfm over the full length of the second grate, and 100 to 150 scfm over the first half of the third grate.

TABLE V-3

THEORETICAL BED COMBUSTION CHARACTERISTICS

Trial	Refuse Moisture	Grate Airflow Case	Burning Zone Lengths (a,b) Gasification From	Gasification To	Char From	Char To	Gasification Zones Gas Composition (Vol. %) Excluding Bypass CO	CO_2	H_2	N_2	H_2O	Grate 1 - Gasification Zone Gas Composition (Vol. %) Including Bypass (c) CO	CO_2	H_2	N_2	O_2	H_2O	Final Temperature °F(e)	Overfire Air Req'd (d) SCFM/ft² w/o Bypass	w/Bypass
1	0.0%	A	1-8	3-8	3-9	9.8%	14.6	10.5	9.7	48.7	16.4	12.4	11.4	7.9	51.4	0.0	16.9	2324	8.4	7.4
2	15.0%	B	1-8	2-15	2-15	3-8	7.7	12.7	5.9	51.0	22.7	5.8	13.4	4.2	53.6	0.0	22.8	2327	4.5	3.5
3	21.4%	A	1-8	3-13	3-14	14.0%	5.6	13.2	4.7	51.7	24.9	3.8	14.0	2.9	54.4	0.0	24.9	2329	3.3	2.3
4	21.4%	B	1-8	2-15	2-15	3-8	"	"	"	"	"	"	"	"	"	"	"	"	"	"
5	25.0%	A	1-8	3-15	-	16.7%	1.9	14.1	1.7	53.2	29.2	0.2	14.8	0.2	55.8	0.0	29.0	2332	1.1	0.1
6	25.0%	B	1-8	3-3	3-4	3-11	"	"	"	"	"	"	"	"	"	"	"	"	"	"
7	25.0%	C	1-8	2-12	2-13	2-14	"	"	"	"	"	"	"	"	"	"	"	"	"	"
8	29.2%	B	1-8	3-4	3-5	3-10	0.1	14.4	0.1	53.9	31.5	0.0	13.4	0.0	55.8	1.4	29.3	2012	0.0	0.0

Trial	Grate 2 - Gasification Zone Gas Composition (Vol. %) Including Bypass (c) CO	CO_2	H_2	N_2	O_2	H_2O	Final Temperature °F (e)	Overfire Air Req'd (d) SCFM/ft² w/o Bypass	w/Bypass	Grate 3 - Gasification Zone Gas Composition (Vol. %) Including Bypass (c) CO	CO_2	H_2	N_2	O_2	H_2O	Final Temperature °F (e)	Overfire Air Req'd (d) SCFM/ft² w/o Bypass	w/Bypass	Char Zone Overfire Air Req'd (d) SCFM/ft² Grate 2 w/o BP	Grate 2 w/BP	Grate 3 w/o BP	Grate 3 w/BP
1	11.2	11.9	6.9	52.8	0.0	17.1	2499	39.8	32.3	9.9	12.4	5.9	54.3	0.0	17.4	2682	18.7	13.7	N/A	N/A	18.2	13.2
2	4.8	13.9	3.4	55.0	0.0	22.9	2503	29.9	19.4	-	-	-	-	-	-	-	-	-	54.1	43.7	18.2	13.2
3	2.8	14.5	2.1	55.8	0.0	24.9	2505	15.7	8.1	1.7	15.0	1.2	57.2	0.0	24.9	2691	7.4	2.4	N/A	N/A	18.2	13.2
4	"	"	"	"	"	"	"	21.9	11.4	-	-	-	-	-	-	-	-	-	54.1	43.7	18.2	13.2
5	0.0	14.7	0.0	57.6	0.0	28.4	2360	5.3	0.0	0.0	13.8	0.0	57.8	1.5	26.8	2347	2.5	0.0	N/A	N/A	N/A	N/A
6	"	"	"	"	"	"	"	7.4	0.0	"	"	"	"	"	"	"	2.5	0.0	N/A	N/A	18.2	13.2
7	"	"	"	"	"	"	"	10.6	0.0	-	-	-	-	-	-	-	-	-	77.4	62.5	N/A	N/A
8	0.0	12.9	0.0	56.7	2.2	28.2	2012	0.3	0.0	0.0	12.3	0.0	57.7	3.0	27.0	2012	0.1	0.0	N/A	N/A	18.2	13.2

Notes:

(a) Assumes no burning on first 9 feet; all distances in feet.

(b) From 2-7 to 3-5 should be interpreted as: from 7 feet down Grate 2 to 5 feet down Grate 3; percent (if given) denotes % unburned carbon in total residue when burnout is incomplete.

(c) Assumes combustion with available oxygen and re-adjustment of equilibrium.

(d) Units are SCFM of air per foot of grate per foot of furnace width for stoichiometric combustion.

(e) Gas temperature after combustion with bypass gases.

TABLE V-4

THEORETICAL CHAR REGION OFF-GAS CHARACTERISTICS

Component	Gas Composition (Vol. %) Excluding Bypass	Gas Composition (Vol. %) Including Bypass (a)	
		Grate 2 (b)	Grate 3 (c)
CO	32.1	21.0	17.5
CO_2	1.6	6.3	7.8
O_2	0.0	0.0	0.0
N_2	66.3	72.0	74.7
Temperature (a)		2700°F	2900°F

(a) Assumes complete burning with available oxygen.

(b) Assumes 15% of the total undergrate air bypasses the bed.

(c) Assumes 20% of the total undergrate air bypasses the bed.

The range of air requirements mentioned in the preceding paragraph arises from consideration of the effect of bypassed undergrate air on the air requirement of the gases leaving the grate. The validity of the bypass or channel-burning concept rests on the data of Kaiser mentioned in Chapter II and on observations of incinerator burning where cracks may be clearly seen from time to time in the burning mass. The fact, however, that by-passing occurs on a random basis suggests that there may be times when the full undergrate air supply is provided to the bed. Under such circumstances, a greater quantity of overfire air would be required (calculated by dividing the indicated "without bypass" air requirement by the quantity: $\frac{100-\%\ \text{Bypassed}}{100}$).

Whatever the relevance of the bypass question when applied to the gasification zone, it is clear that under almost all circumstances, substantial amounts of carbon monoxide will be formed and released to the overfire volume in the char zone. Since the assumptions used in calculating the gas compositions assumed only carbon was present in the bed, it can be noted that the air requirement is related only to the underfire air rate and thus can be made smaller (but never zero since some air flow is needed for grate cooling) in direct proportion to the underfire air rate. Since the char zone is almost invariably located on the burnout grate, this emphasizes the desirability of minimizing the undergrate air flow in this area consistent with obtaining complete burnout. Studies on particulate emission also suggest the desirability of maintaining low underfire air rates in this area to minimize ash entrainment. It is clear, however,(Case A), that reducing the underfire air rate too much will result in incomplete burnout.

2. FURNACE ENVIRONMENT

Based on the calculation method described above, the composition, temperature and mass flow rate of gases entering the furnace enclosure can be estimated. The next step in analysis of the system is to evaluate flow patterns and mixing. The procedures described in Chapter III are directly applicable. It may be advantageous, however, to consider a three zone flow analysis (gasification, char burnout and ash cooling) and to carry out, using numerical techniques, an analysis of the (curving) streamline patterns throughout the furnace enclosure. From such a more detailed evaluation, the mixing of oxygen-containing gases from (under some conditions) the gasification zone and from the ash-cooling zones could be estimated. Such an analysis would suggest the minimum requirements for overfire air systems, would identify velocity distributions, and would affect the convection boiler design (regarding tube spacing and protection from erosion), and would provide a basis for estimation of overfire air jet crossflow effects.

Of particular importance is estimation of the velocity environment with which overfire air jets will interact. As suggested in Chapter III, the hot gases rising from the gasification and char burnout zones are accelerated in the hydrostatic pressure field created by the slow-moving cold gases in the discharge grate region. The velocity of the gases may be estimated by the following rationale.

Assuming that the behavior of the bed off-gases of density ρ_h in rising from the top of the bed at an elevation z_1 to another elevation z_2 and experiencing a pressure change from P_1 to P_2 may be described by the Bernoulli equation:

$$\text{Total Head} = \text{Constant} = \frac{Pg_c}{\rho_h g} + \frac{u^2}{2g} + z \qquad \text{(V-1)}$$

where g is the acceleration of gravity.

Therefore, from elevation 1 to 2, the hot gas behavior may be described by:

$$\frac{u_2^{\,2}}{2g} = \frac{u_1^{\,2}}{2g} + \frac{(P_1 - P_2)g_c}{\rho_h g} + (z_1 - z_2) \qquad \text{(V-2)}$$

In the cold gases of density ρ_c, it is assumed that velocity changes are small, such that:

$$\frac{P_1 g_c}{\rho_c g} + z_1 = \frac{P_2 g_c}{\rho_c g} + z_2 \qquad \text{(V-3)}$$

Combining these equations and in recognition of the reciprocal relationship between temperature and density, the following results:

$$u_2^{\,2} = u_1^{\,2} + \frac{2g}{g_c}(z_2 - z_1)\left(\frac{T_H}{T_C} - 1\right) \qquad \text{(V-4)}$$

where T_H and T_C are the temperatures (°R) of the hot and cold gases, respectively.

Using Equation (V-4) and substituting the temperatures calculated above, hot gas velocity may be estimated as a function of the height over the bed. The results of such calculations are shown in Table V-5. It is of interest to note the rapid increase in velocity over the first few feet of rise. This has particular significance in jet trajectory calculations where it can be seen that the velocity environment traversed by the jet is strongly related to the elevation and is quite different from that calculated without consideration of this effort.

3. JET BEHAVIOR

The evaluation of overfire air jet behavior can be readily carried out using the correlations presented in Chapter IV. With these correlations, trajectory estimates can be prepared which incorporate crossflow and buoyancy effects (using values such as those in Tables V-3, V-4 and V-5). The combustion processes associated with the jet can also be evaluated using the gas composition data. In establishing the flow rate requirements, attention will be required concerning the length of grate needing air and the effect of overfire air in escalating gas temperatures and the possible need for excess air to provide cooling.

TABLE V-5

THEORETICAL GAS VELOCITIES ABOVE REFUSE BEDS (ft/sec)

Feet Above Bed	Gasification Zone Grate 1				Gasification Zone Grate 2				Gasification Zone Grate 3			
Trial	0	2	4	6	0	2	4	6	0	2	4	6
1	1.3	10.7	15.1	18.5	7.0	13.3	17.5	20.8	3.6	12.5	17.3	21.0
2	1.3	10.9	15.4	18.8	9.4	14.8	18.8	22.0	-	-	-	-
3*												
4	1.3	10.7	15.1	18.5	9.2	14.6	18.5	21.7	-	-	-	-
5*												
6	1.2	10.7	15.1	18.5	8.5	13.8	17.5	20.6	3.0	11.1	15.5	18.8
7	1.2	10.7	15.1	18.5	12.3	16.4	19.6	22.4	3.0	11.1	15.5	18.8
8	1.1	9.4	13.2	16.2	7.6	12.0	15.2	17.8	2.7	9.7	13.4	16.3

Feet Above Bed	Char Zone Grate 2				Char Zone Grate 3			
Trial	0	2	4	6	0	2	4	6
1	-	-	-	-	3.3	13.4	18.7	22.7
2	8.7	15.0	19.4	22.9	3.3	13.4	18.7	22.7
3*								
4	-	-	-	-	3.3	13.4	18.7	22.7
5*								
6	-	-	-	-	3.3	13.4	18.7	22.7
7	-	-	-	-	3.3	13.4	18.7	22.7
8	-	-	-	-	3.3	13.4	18.7	22.7

* Negligible cold gas zone in these trials. Analytical model does not apply.

CHAPTER VI

TEST PLAN

CHAPTER VI

TEST PLAN

A. INTRODUCTION

Because of the complexity of the incineration process, design approaches based on analytical predictions of behavior require experimental demonstration. This chapter presents a test plan for demonstration of the effectiveness of design methods using the analysis methods presented in Chapters II, III and IV. We recommend that such tests be performed in full-scale plants rather than in pilot plant or laboratory units. In view of the complexity of the process, scaling laws for extrapolation to full-scale are uncertain.

The objectives of the study are to:

- Determine the effect of operating and design parameters on combustible pollutant emissions by quantitative assessment of the performance of a selected incinerator;
- Evaluate the effectiveness of a number of alternatives in jet mixing systems to effect burnout of these combustible pollutants within the primary combustion chamber; and
- Develop guidelines for incinerator design and operation based on gathered data which identifies promising combustible pollutant control techniques.

These objectives would be met by efforts including the following steps:

a. Conceptual Design--Prior to the implementation of a test program at an incinerator, a review of current combustion theory is necessary to delineate the areas where improved design can be effected. Such information is available from this report. The pertinent analyses and conclusions are presented below.

b. Preparation for Tests--The first step in implementing such a study is to negotiate with an appropriate municipality to make an incinerator facility available. Then, final detailed construction plans for jet mixing systems can be prepared and appropriate equipment purchased and installed at the incinerator for use in the test program. Tests should then be carried out at the incinerator site to define the flow of air and refuse to the system and install measuring instruments to monitor these flows during subsequent tests.

c. Uncontrolled Emission Rates--The second step is to carry out a series of tests approximating seven (7) test days to determine the combustible pollutant emissions (both gaseous and particulate) typical of the selected incinerator as they relate to feed rate and undergrate air flow and distribution. These data will be analyzed to suggest the sources of combustible pollutants within the system; to identify any relationships between the emission rates of the various pollutants one to another and to incinerator operating conditions; and to identify through prolonged testing the test duration necessary to obtain meaningful average values of measured quantities.

d. Incinerator Behavior--The third step is to conduct a series of experiments to characterize the behavior of the burning refuse bed and furnace flow dynamics. Sampling of bed off-gas as a function of position would identify the need and location for overfire air. Also, the data would be useful in corroborating the bed and flow behavioral analyses. Ten (10) test days and some laboratory flow modeling studies is suggested as a target for these tests.

e. Mixing Experiments--The fourth step is to conduct a series of tests using steam and air injected into the overfire combustion volume and determine the consequent effects on combustible pollutant emissions. The effect of injection angle, discharge velocity and mass flow will be determined for both air and steam injection. In the course of this program, a cross-furnace stationary probe will be evaluated. A target of fourteen (14) test days is suggested for these tests.

f. Design and Operating Guidelines--The final step is to analyze the data collected in the test program above. From this analysis would result:

- Guidelines for incinerator operations to minimize combustible pollutant emissions for existing plants; and
- Design guidelines for modification of existing plants and for new construction to enable the design of systems with minimum combustible pollutant emissions.

A detailed discussion of these steps follows,including necessary implementation plans and technical considerations.

B. TECHNICAL DISCUSSION

1. CONCEPTUAL DESIGN

From this report, several conclusions can be drawn as to the type of incinerator to be used, the location and characterization of the jets, and data acquisition scope and methods.

a. The Incinerator

There are more than 500 incinerator furnaces now operating in the United States. Even within one plant, strict duplication is seldom found. This fact emphasizes the need for generality in the results of any design correlation development effort, and also highlights the difficulty in finding a truly representative incineration system. It is our feeling that an incinerator for testing should meet the following criteria: it should be of relatively recent construction, laid out such as to facilitate modification and testing, and should be representative of the type of incinerator presently enjoying the greatest "popularity." Although a few batch-feed systems are being built throughout the nation, the majority of new furnaces are of the continuous-feed variety, utilizing multiple mechanical grates to move the refuse through the primary furnace enclosure.

For a given incinerator, there are a limited number of variables which may be controlled. These include the grate speed (refuse feed rate), the undergrate air supply (total mass flow and its distribution to the various undergrate plenums), and the flow dynamics of the overfire air system. The refuse composition and heating value is largely uncontrollable, reflecting the daily receipts, but could be modified to reduce heating value by metered addition of water, either to the pit (preferred) or to the charging chute.

The undergrate air supply and distribution is important from two standpoints: the undergrate air flow largely defines the burning rate within the system and is also instrumental in controlling grate surface temperatures. Undergrate air rate is also important as it affects the formation of residue clinkers by forcing temperatures within the bed to ranges where the residue materials exceed their sticking temperatures. Typically, incinerators are operated with total undergrate air flows corresponding to an overall 50% excess air. Often, however, the distribution of the air is not such that its introduction corresponds with the air demand of the bed. In most cases, much of the air passes up through the discharge grate where it is relatively ineffective for combustion.

b. The Jet System

Prior to the actual design of the overfire air and steam injection systems, two initial design details must be determined: the location of the jets and the proposed jet variables.

Two conceptual pictures have been suggested to identify the location within an incinerator furnace where combustible materials are formed.

- In the pyrolysis region of the furnace (the first few feet of the second grate) the combustion air demand exceeds the air supply and incomplete combustion occurs. The gas flows are such that the combustibles would be expected to be found at the top of the primary furnace and breeching. Combustion of the fuel-rich gases could be effected by overfire air injected at high velocities directly over the pyrolysis region or by overfire air directed from elsewhere in the furnace which is caused to enter the rising stream of pyrolysis gases. These approaches are represented in the sidewall and roof jets suggested for these tests and illustrated in Figure VI-1.

- In the colder regions of the furnace (near the discharge end of the third grate) burning occurs with such an excess of air that combustion is quenched. In this case, we would expect to find the combustible pollutants in the cooler gases at the bottom of the outlet breeching of the primary chamber. If this region is, indeed, the source of a sizable fraction of the combustible pollutants emitted (probably more associated with carbon monoxide and hydrocarbon emission than soot), means are needed to reduce the flow rate of cooling air and to effect mixing of the remaining off-gases with the hot gases generated elsewhere in the furnace. The former aspect of combustible pollutant control can be effected directly by reduction of the undergrate air flow to the minimum required for grate cooling. The second aspect could involve the use of steam jets (injecting mixing energies rather than additional air), directed in such a manner as to prohibit escape of the cool gases along the bottom of the furnace and flues and to encourage their mixture with the hot gas stream. This is suggested and is also indicated in Figure VI-1.

In order to provide a rough estimate of the relative importance of the pyrolysis region and discharge region hypotheses for combustion pollutant generation, a limited number of tests (reported in Appendix F) were carried out. These tests show higher carbon monoxide concentrations in the cooler (lower) regions of the furnace breeching and almost none in the hotter (upper) regions, suggesting the dominant importance of the discharge region as a source of combustible pollutants. However, the fact that at the time of the tests the furnace was being operated with a dry commercial refuse (offering near optimal combustion characteristics) biases the situation so that, for the moment, the hypothesis that the pyrolysis zone can be an important source of these pollutants should be retained. Indeed, smoking in the pyrolysis region is common when the refuse is wet. Also, no tests were made of hydrocarbons or combustible particulate loadings, and thus the source of these pollutants (anticipated to be the pyrolysis region) was not established.

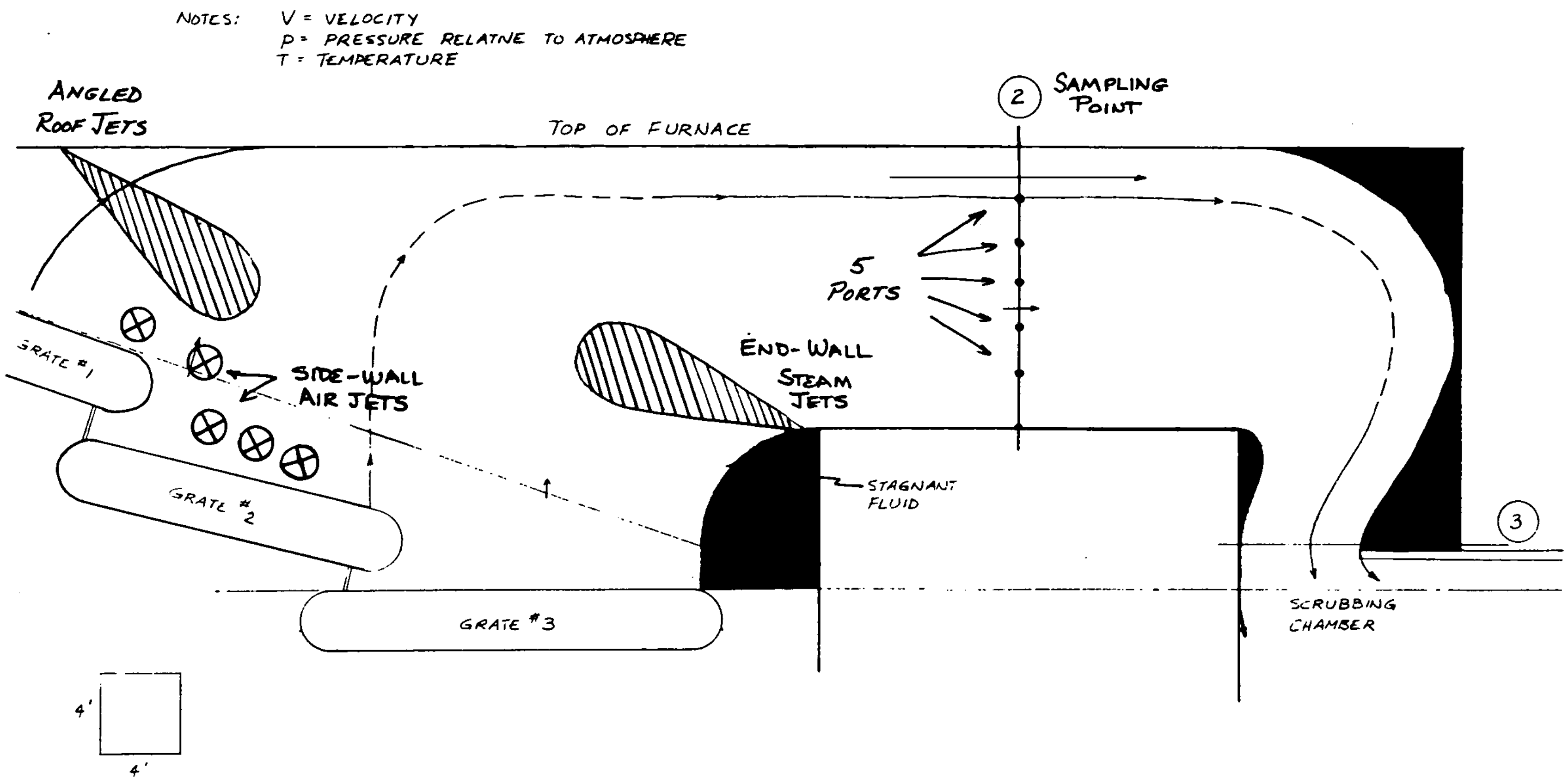

FIGURE VI– 1 SCHEMATIC DIAGRAM: TWO-FLUID MODEL OF INCINERATOR FURNACE FLOW

Following selection of the jet location, consideration must be given to the dynamic flow parameters which should be associated with the jets: discharge velocity, mass flow rate and injection direction. With respect to the velocity, care must be taken for the sidewall jet system that impingement on the opposite wall does not occur. Because overfire combustion is anticipated for air jets discharging into the pyrolysis region, temperatures within the jet could exceed 2500° F. Impingement of such high-temperature flows directly on refractory can lead to slagging, fluxing and rapid refractory degradation. As a consequence, one approach to jet operation should include opposed jets (Figure VI-2a) where high jet discharge velocities can be used, but the jets located on the opposite side of the furnace act to prohibit direct wall impingement. One might anticipate problems in exact opposition of these systems; and, as a consequence, some jet deflection into the bed may occur. As this could cause the entrainment of fly ash material, thus disadvantageously increasing the particulate loading in the flue gases, this effect must be monitored.

An alternative to opposed jets involves an interlacing of the jets (Figure VI-2b). For this approach, care must be given to the selection of jet operating characteristics to avoid penetration distances greater than the furnace width.

The velocity and mass flow rate for the jets should be selected on the basis of the anticipated air requirement and the expected jet penetration and flow path. Since some uncertainty exists in these flow paths (see Chapter IV), careful observation of the system in operation and provision of a wide range of possible operating characteristics (as reflected in the fan pressure and volume flow capability) is necessary.

c. Data Acquisition Scope and Methods

In order to provide a meaningful indication as to the need for and effectiveness of jet mixing systems, a location must be found which provides a valid measure of the effect of combustion chamber mixing on combustible pollutant emission rate.

As a minimum, sampling ports should be installed in the outlet breeching of the primary chamber. A disadvantage of measurements limited to the breeching area is that the conditions of gas composition and temperature and velocity found in the breeching represent the integrated effect of a number of interacting processes and forces within the furnace which are not explicitly revealed in point measurements taken in the breeching. To partially offset this limitation, we recommend that the majority of tests use a profile method of measurement where the vertical gradients of temperature, composition and velocity of the gas flow would be measured. Although the relationship between the fluid sampled at, say, the upper region of the breeching and a specific region within the furnace cannot be established with high assurance, some inferences can be drawn. It is also possible that the data from a cross-furnace stationary (CFS) probe may make it possible to better relate breeching profiles to incinerator furnace enclosure flow behavior.

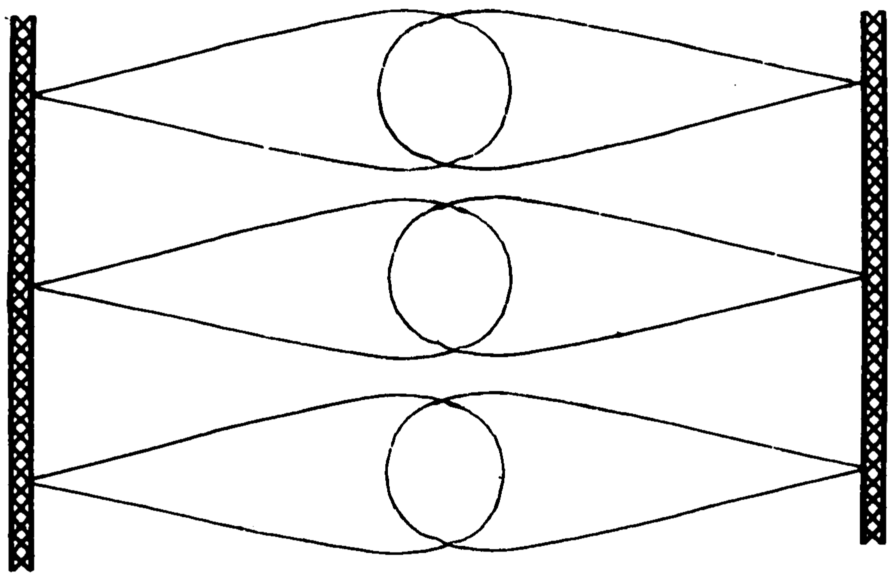

FIGURE VI-2a **OPPOSED JETS**

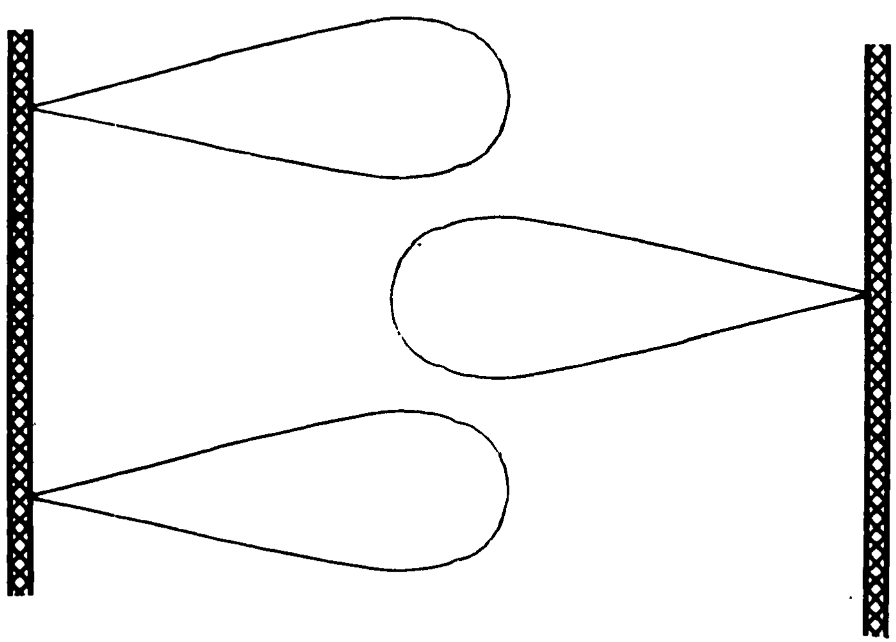

FIGURE VI-2b **INTERLACING JETS**

The variables of interest include gas temperature, velocity and composition. Detailed data of this sort will enable calculation of energy and material balances about the primary furnace, suggest the sources (pyrolysis or discharge grate areas) of combustible and mineral pollutants, and permit the estimation of furnace emission rates of air pollutant species. Initially, measurements will be made of:

- Combustion Gases (CO, CO_2, O_2, N_2, H_2, H_2O)
- Hydrocarbons
- Mineral Particulate
- Combustible Particulate

In regions of high CO concentration, the $CO/CO_2/N_2/O_2$ measurements can be made on a batch basis with a manual or electric Orsat. In regions of low CO concentration, non-dispersive infrared methods are necessary. The latter measurement method (which we expect to be necessary in the breeching area) will give continuous CO concentrations for one probe or, by use of a sequenced solenoid valving arrangement, will give CO readings at multiple elevations at frequent intervals.

Water vapor determinations can be made by metering condensate and gas flow rate in areas of high moisture content and by wet/dry bulb means for lower moisture.

Hydrogen determinations (important in studying the water-gas reaction equilibria) can be made on a batch basis using a gas chromatograph.

Hydrocarbon determinations can be made using flame ionization methods.

Particulate matter determinations (dry dust only) can be made with filters alone (excluding the impingers associated with the full EPA sampling train).

Probes for the test program should be water-cooled to enhance their survival in the high-temperature environment and, importantly, to quickly quench combustion reactions of the gases, aerosols and particulate. Ideally, the probe should also incorporate a thermocouple junction, shielded from radiation and immersed in the in-flowing gas stream. Also, and especially for tests in the breeching area, the probe should be equipped to determine the impact and static pressure (velocity) of the gases in the sampling region.

In addition to the sample gases, data acquired during the tests should include a variety of quantitative and qualitative data regarding the system operating characteristics. These would include:

- All forced airflows into the furnace (underfire air under each grate, overfire air or steam jet flows);

- Grate speed (as it indicates refuse feed rate);

- Readings of standard furnace instruments (draft gauges, thermocouples, etc.);

- Commentary on the appearance of the feed refuse (dry or wet; fluffed or compact; unusually high concentrations of leaves, plastics, metal; etc.);

- Commentary on the appearance of the main flame (shape, end point along grate, length);

- Commentary on point along grate where burn-out of refuse is complete and qualitative evaluations of the combustible content of the residue;

- Commentary on the appearance (penetration, trajectory, shape) of observable jet flows.

2. PREPARATION FOR TESTS

The initial groundwork for the tests includes the selection of an adequate incinerator and the preparation of the final shop drawings of the overfire air and steam injection systems. Following the formalization of the arrangement between the contractor, EPA and the incinerator authorities, the contracting for the installation of air and steam-handling equipment and for refractory modification is possible.

The pre-test effort would also include the installation and calibration of appropriate flow-measuring devices to enable monitoring of all forced air flows both above and below the grate. In addition, the sampling probes would be designed and analytical techniques will be tested appropriate to the special needs of the program.

3. UNCONTROLLED EMISSION RATES

It will be necessary to provide a reference emission baseline prior to the conduct of tests on the effectiveness of overfire air systems. For these, and in subsequent tests, the majority of the emission determinations could be made in the breeching leading out of the primary combustion chamber (Figure VI-1).

In this phase of the program, the effects on emissions of two operating variables would be determined: the underfire air flow rate and distribution and the grate speed (refuse feed rate). The underfire air distributions tentatively suggested for tests on a typical triple-grate continuous feed incinerator configuration are shown in Table VI-1.

TABLE VI-1

SET POINTS FOR GRATE AIR FLOW STUDIES

Grate	Condition	Air Flow (percent of stoichiometric) I	II	III	IV
1		0	0	0	0
2		75	50	113	75
3		25	50	37	75
Total Undergrate Air		100	100	150	150

In order to avoid the possibility of overheating the grate, experiments should not be carried out at underfire air rates less than that equivalent to stoichiometric rate. We would suggest that the total undergrate air flow should be varied from stoichiometric to 50% excess air. Also, the distribution of undergrate air should be modified. In order to assure a continuing refuse disposal capacity, the refuse feed rate (grate speed) should not be reduced below 75% of the rated furnace capacity.

This range of parameters should give a strong indication of the effects of variation of these parameters on pollutant emission rates. In tests on the effects of overfire air on combustible pollutant emissions, a subset of three of the conditions shown in Table VI-1 and the two suggested grate speeds should be used. These conditions would be chosen in part to reflect "normal operating practice", best operating practice and "undesirable" conditions in that combustible pollutant emission rates may be high, but other system characteristics (e.g., feed rate) may be desirable. Measurements to be taken in the breeching would include profiles of the following variables:

- Gas velocity;
- Gas temperature;
- Carbon monoxide, hydrocarbon and hydrogen concentrations;
- CO_2, O_2 and moisture concentrations; and
- Particulate emission rate (reported separately as combustible and mineral particulate emission rates).

The results sought during the seven (7) test days which we feel appropriate for this series would include data illustrating the vertical profile of pollutants and their response to changes in firing rate and undergrate air flow rate and distribution and data illustrating the timewise variation of pollutant emissions. From these data, the following would be sought:

- Postulates regarding the sources of combustible pollutants within the furnace system (based on the distribution of the pollutants in the different gas temperature and flow regions);
- Relationships between the emission rates of the various combustible pollutants to simplify subsequent testing and analytical procedures;
- Perspectives which will identify the minimum test duration needed to obtain valid average values of the fluctuating pollutant emission rates;
- Reference points in system performance against which to judge the effectiveness of mixing systems; and
- Relationships between the emission rate of the various pollutants and the operating characteristics of the incinerator.

4. INCINERATOR BEHAVIOR

It is appropriate to study incinerator behavior both for verification of the analytical tools developed in Chapters II, III and IV and for information of the jet locations, air rates and injection velocities proposed for the demonstration tests. Specifically, we recommend sampling of the space over and along the bed to provide composition and temperature data to test the bed burning model.

In addition, it may be of interest to study the flow patterns in the furnace. The data from these tests would be useful in checking the analysis given in Chapter III and in interpreting the data on incinerator behavior. Paralleling the study of incinerator gas flow, modeling studies (in the laboratory) could be of value in providing a tool for application in future incinerator design efforts.

a. Confirmation of the Bed Burning Model

The bed burning model verification requires data on gas compositions, estimates of heat flux across the top of the bed and temperature measurements of the leaving gases. The gases rising from the bed should be sampled to determine:

- Temperature;
- Concentration of CO_2, O_2, N_2, H_2, H_2O and CO; and
- Particulate loadings.

The sampling should be done at various positions along the length of the bed and the results correlated with data on the gross refuse composition $[C(H_2O)_n]$ as measured by analysis of the flue gases at the breeching to the secondary chamber.

Particulate loading measurements would be determined as a function of position and undergrate air flow.

Flux to the top of the bed could be estimated by pyrometric evaluation of refractory wall temperature and subsequent radiative heat transfer analysis. An approximate analytical method should be developed for estimation of this quantity in the general case. This method could be checked by the measured wall temperatures.

This experimental program and analysis effort would seek to provide the following:

- Data to show the source of particulate and gaseous combustible emissions along the grate to suggest means to reduce emission rate.
- Confirmation of the water-gas shift equilibrium hypothesis.
- Development of a method for estimating heat flux to the refuse surface.
- Confirmation of the ability of the bed-burning model to predict gas compositions and burnout.
- A test of the theories proposed in the past as to the relationship between underfire air rate and particulate emissions.

b. Measurement of Gas Velocities

There is little data on furnace gas flow. The analytical models that are discussed in Chapter III implied certain velocity levels and profiles in the overfire region. To assess the validity of these models, we suggest measurements of gas flow velocity in the incinerator. Photographic techniques, as described by Lavrov[3], could be used for this purpose. The technique uses high-speed photographs of the hot gas flow field, to determine velocities by analysis of the trajectories of incandescent particles. In the case of an incinerator, it may be possible to observe particles rising from the refuse bed. Alternatively, it may be desirable to inject fine powders of magnesium or other material to obtain a more controlled source of particles. The observations would be made through holes in the sidewalls or ceiling of the furnace at points where other measurements are planned.

c. Scale Modeling of Incinerator Flow

A second way to assess the validity of the analyses of the gas flow is to compare its results to observations of flow in a laboratory scale model. In the course of the experimental study of jets in a crossflow described in Chapter IV, we developed techniques for simulating flow fields with large temperature (hence, density) gradients. Specifically, room temperature air can be utilized to simulate the hottest gases in an incinerator if nitrogen vapor near its boiling point (-320°F) is used to simulate the coolest gases. Water vapor in the air condenses at points in the simulated flow where the temperature is below the dew point so that the mixing region between the cold nitrogen vapor and the room air is clearly visible and can be photographed.

We suggest construction of a scale model of the incinerator (desk-top size) and simulation of the furnace flow using this technique. Two or more zones of flow should be utilized. This laboratory approach has the advantage that input velocities and flow rates can be accurately determined and alternate geometries can be studied. The flow field can be photographed and relatively simple temperature measurements can be made to obtain an indication of the degree of mixing. The model could be used to assess the validity of the analysis, for example, by comparing observed widths of zones of flow in the breeching area with those calculated. This work could also provide a new approach for simulating furnace flow that could be applied to various types of configurations.

As a result of these efforts, the following would be sought:

- A bringing together of an analytical technique, a low-cost laboratory technique and data on the full-scale device to enable evaluation of incinerator flow dynamics.
- Confirmation of the theoretical analysis of gas acceleration and mixing in incinerators.
- Development of a method for study of alternate chamber geometries which allows simulation of the important buoyancy effects.

5. MIXING EXPERIMENTS

The experimental tests outlined above would document a baseline of combustible pollutant emissions against which to judge the effectiveness of the mixing tests described here. As shown in Figure VI-2, the incinerator would be equipped with three systems for overfire volume mixing; a roof-entry air jet system, a sidewall-entry air jet system, and a discharge end-wall steam jet system. In the fourteen (14) test days envisioned for this part of the program, these jet systems would be operated and their net effect on combustible pollutant emission rates determined by profile measurements in the breeching. Also, a cross-furnace probe could be evaluated for use in the follow-on test program described below.

The test matrix presented for the baseline series of measurements (Table VI-1) indicates eight combinations of grate underfire air rate and distribution and grate speed which are of interest. From among those eight conditions and in light of the baseline tests, two or three should be selected for use in the study of jet effects. The overall conditions within the refuse bed will likely be controlled by the underfire air and grate speed parameters. Since the bed takes long times to reach an equilibrium state, the plant should be operated at a constant underfire air rate and distribution and grate speed over an entire test day. We anticipate, however, that the overfire jets will not strongly affect the bed processes but will rapidly (within seconds) shift the pollutant emission characteristics to reflect new values corresponding to equilibrium in the overfire combustion and mixing processes. As a consequence, we speculate that a large number of jet effect tests can be run in a given day.

a. Sidewall Jet Tests--The experiments with sidewall jets would fall into two categories. As shown in Figure VI-2, the sidewall jet system would be comprised of a total of ten jets arranged in two banks of five on either side of the furnace. The jets would be located directly opposite one another. Three jets on either side can then be operated in an opposed jet arrangement and a three-and-two combination can be used for interlacing jets (Figure VI-3). The sidewall jets will be operating in a region of the furnace where we would anticipate high release rates of pyrolysis products and thus the jets would be expected to have a great effect on the production of soot (combustible particulate) within the furnace.

b. Roof Jet Tests--The roof jets would operate using a feed-end roof jet bank modified to inject the air at an angle toward the discharge end of the furnace. As for the sidewall jets, roof jets would add air to the stream of pyrolysis products rising from the bed and have the advantage in comparison to the sidewall systems that hot spots arising from impingement on the walls should not be experienced.

c. Steam Jet Tests--The steam jets operating from the discharge end-wall of the incinerator would serve to induce mixing and control flow patterns in the cooler, air-rich gas arising from the grate near the discharge end of the furnace. It has been found that under some conditions of operation, a substantial fraction of the carbon monoxide appears to arise in this region of the furnace. Thus, by inducing mixing of this cold gas with the hotter gases generated towards the feed end of the furnace, higher temperatures and CO burnout can be anticipated.

In addition to the tests of the jet systems as determined by measurement of the furnace output at the breeching, we would also suggest use of a cross-furnace stationary (CFS) probe (Figure VI-3). Determinations of the temperature and gas composition distribution within the furnace would be of great interest and value. If, as described below, it appears valuable to study the furnace environment in more detail, we suggest that use of a grid pattern of CFS probes, such as the one to be tested, would be a desirable approach. Therefore, we suggest use of a prototype of the CFS probe to gather data during the jet evaluations. The probe will provide for the measurement of temperature, for gas sampling (non-isokinetic) in the furnace and may be suitable for obtaining gas velocity data.

6. DESIGN AND OPERATING GUIDELINES

The data developed above would provide estimates of the effectiveness of overfire mixing in the reduction of combustible pollutant emissions. In addition, information would be derived on the correlation between the emission rates of combustible and mineral particulate as they relate to incinerator operating parameters (feed rate and undergrate air flow). To the extent justified by the data, generalizable conclusions and recommendations can be drawn from the experimental results which will constitute guidelines for incinerator operations and design to minimize combustible pollutant emissions. These guidelines would find use in the operation of existing plants, would suggest approaches to plant modification and would be helpful in the development of designs for new plants. It should be recognized, however, that the data taken at the outlet breeching of the furnace represent the integrated result of interactions of the fuel bed, furnace enclosure flows and jet flows. To an extent, it may be possible to infer the combustion chamber dynamics from the profiles determined in the outlet beeeching. The development of definitive and highly credible design correlations, however, would rest on data taken throughout the furnace enclosure. These tests could be part of a subsequent program, although they would make extensive use of the cross-furnace stationary probe and cantilever probe sampling techniques developed in the course of this effort.

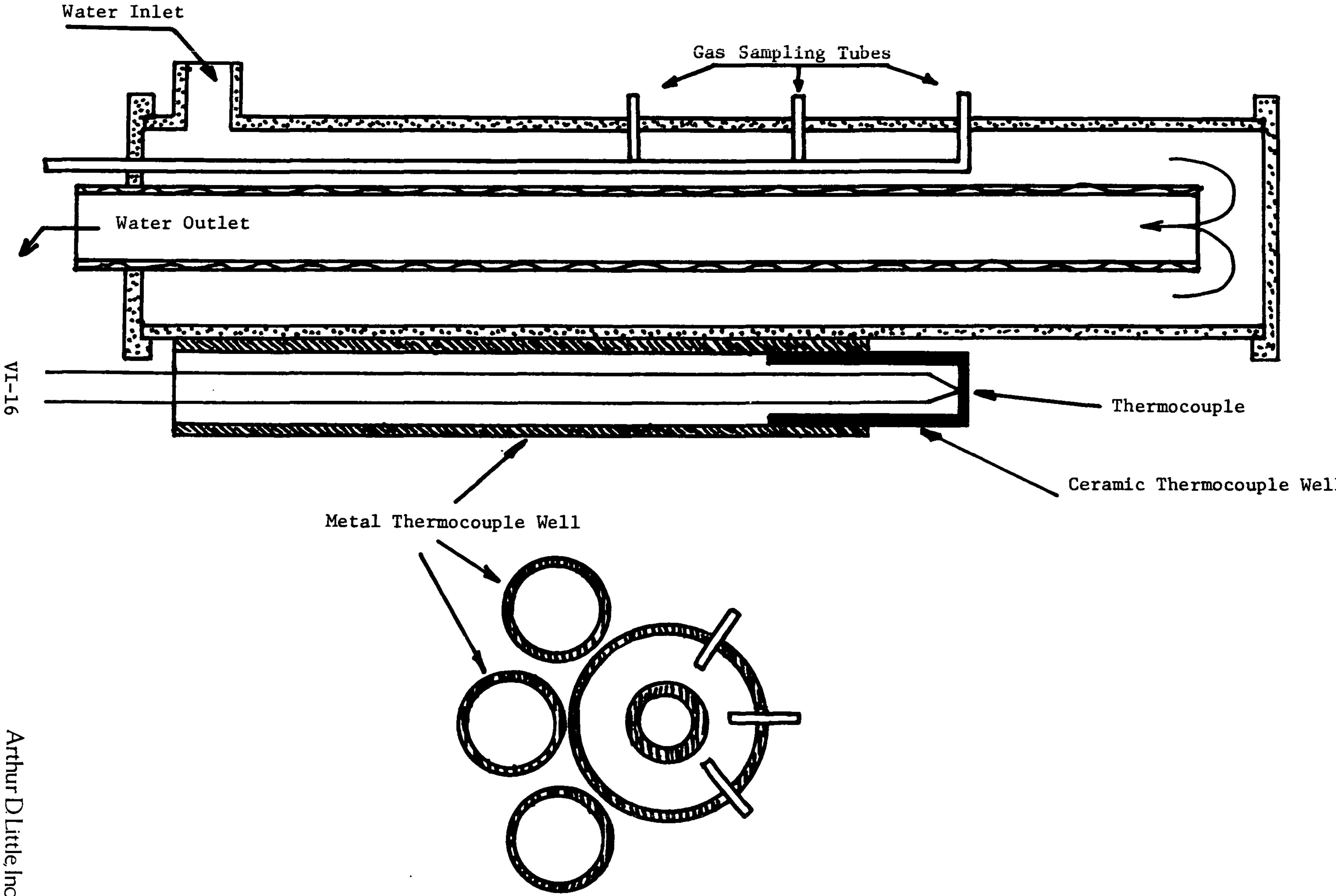

FIGURE VI–3 CROSS-FURNACE STATIONARY (CFS) PROBE

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CHAPTER VII

BIBLIOGRAPHY

CHAPTER VII

BIBLIOGRAPHY

Chapter VII is made up of two parts: (A.) a bibliography prepared as part of the Arthur D. Little, Inc. study of incineration overfire mixing; and (B.) abstracts of the foreign articles appearing in the bibliography. The abstracts were, for the most part, prepared by a member of the ADL project team. In cases where they have been taken from CHEMICAL ABSTRACTS or ENGINEERING INDEX, the source has been noted. Reference 90 has not been abstracted as it was a major source for this report and has been cited extensively.

The bibliography is a compilation of selected references from specific subject areas. These subject areas include:

The comprehensive literature search conducted for this project covered the years 1901 through 1956. A check in more recent literature indicated that the technology we sought information on was not covered after the early 50's. Domestic technical and government sources were reviewed as well as foreign technical literature. The most pertinent sources of information were found to be ENGINEERING INDEX and a listing of Bureau of Mines publications.

The following sources were covered:

CHEMICAL ABSTRACTS	1907 thorugh 1956
ENGINEERING INDEX	1901 through 1953
INDUSTRIAL ARTS INDEX	1934 through 1954

Additional sources were "International Symposium on Combustion," numbers 1 through 12 and "List of Publications Issued by the Bureau of Mines, July 1, 1910 to January 1, 1960." Bibliographies of all articles reviewed were scanned for pertinent references and extensive use was made of the ADL and MIT library holdings.

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B. ABSTRACTS OF FOREIGN ARTICLES

3. MIT WELCHEM UNTERDRUCK WIRD DIE ZWEITLUFT DEM FEUERRAUM AM ZWECKMAESSIGSTEN ZUGEFUEHRT? (At What Pressure Should Secondary Air Be Supplied to a Furnace?)
   Anon
   Wärme 61 (5), 97 (29 January 1938)

   The pressure at which secondary air should be supplied to combustion chamber is briefly discussed. Results are given of investigations carried out on a sectional boiler with 600 sq.m. heating surface equipped with a zone controlled force draft stoker. It is shown that nozzles are much more effective than simple wall openings. [Eng. Index, 131 (1938)]

14. UBER DIE WIRKUNGSWEISE VON RUSSBLASERN (On the Way in Which Soot Blowers Work)
    K. Cleve and R. Muller
    Arch. Warmewirtsch. u. Dampfkesselw. 21 (1), 17-9 (1940)

    For many years soot blowers have been used for the operation of boilers in order to avoid slag depositions and fly ash agglomerations on the heating area. However, very little is known about the flow and pressure history of the medium which is blown - mostly overheated vapor - after it has left the soot blowers. In this paper some tests are reported which explain these conditions. In addition, the highest possible jet velocity which is permissible without endangering the sidewalls is briefly discussed.

20. DIE TECHNIK DER ZUFUERUNG VON VERBRENNUNGSLUFT (The Technique of Supplying Combustion Air)
    R. Fehling
    Arch. Warmewirtsch. u. Dampfkesselw. 11 (4), 119-23 (1930)

    Thermal and aerodynamic conditions of combustion air admission are discussed; nature of turbulence, its influence on combustion, and relation between air admission and heat transfer in furnace. [Eng. Index, 189 (1930)]

24. ZWEITLUFTZUFUHRUNG BEI ROSTFEUERUNGEN (Secondary Air for Grate Firing)
Wilhelm Gumz
Feuerungstech. 23 (11), 123-4 (1935)

Among the topics discussed are the necessity for secondary air for larger grate firings, preconditions as related to the effect of secondary air, amount and spatial distribution of the air, choice of pressure and temperature and its influence on the penetration, advantages of secondary air.

25. ZWEITLUFTZUFUHRUNG. DIE DUSENANORDNUNG UNTER BESONDERER BERUCKSICHTIGUNG DES SYSTEMS BADER (Secondary Air Supply. Jet Arrangement with Special Regard to the Bader System)
Wilhelm Gumz
Feuerungstech. 30 (2), 32-6 (15 February 1942)

The advantages and implications of a secondary air supply are discussed along with the amount of secondary to be used, the influence of the pressure, influence of the jet diameter and the local arrangement of the secondary air jets. The Bader system is described. Comments are made on the introduction of secondary air immediately over the coke layer and introduction through a layer regulator which is built in the form of a hollow tube.

27. VERSUDHE AN FLAMMROHR-INNENFEUERUNGEN MIT ZWEITLUFTZUFUHRUNG (Experiments with Secondary Air Supply in a Flame Tube Internal Furnace)
H. Janissen
Wärme 62 (12), 205-7 (25 March 1939)

The smoke and soot formation, which in recent years has become noticeable again by the increased demand from the present boilers, occurs particularly often if flue boilers [cornwall boilers] with normal plane grate internal firings are used. To abolish this shortcoming, several firms produce special provisions for secondary air supply and offer them to be built into such firings. Experiments with such installations yield a survey of the way of their operation and the possible savings if the required preconditions are given. It must be pointed out, however, that before the installation, appropriate measurements have to show whether an improvement can be achieved by secondary air supply.

29. THEORETISCHES UBER ZWEITLUFTZUFUHR BEI ROSTFEUERUNGEN
(Theory of Secondary Air Supply for Grate Firing)
P. Koessler
Arch. Warmewirtsch. u. Dampfkesselw. 19 (6), 153-6 (1938)

For some time, the secondary air supply for grate firings has received much attention; however, it seems that the possible firing techniques which are noted are not always clear. Therefore, the author treats here the problems of secondary air supply according to processes in the combustion chamber. In a second paper [Reference No. 30], the author outlines the state of research and the problems which are still to be solved.

30. ZWECKMABIGE ZWEITLUFTZUFUHR BEI FEUERUNGEN- STAND DER FORSCHUNG UND ZUKUNFTIGE AUFGABEN (Appropriate Secondary Air Supply in Firing. State of Research and Future Problems)
P. Koessler
Arch. Warmewirtsch. u. Dampfkesselw. 19 (7), 169-73 (1938)

Secondary air is well apt to mix the gases in the combustion chamber. By its application it is possible to achieve a far-reaching improvement in the combustion process on the traveling grate and to better adjust the firing to fuel and load. With a suitable secondary air arrangement, it is certainly possible to decrease the height of the combustion chamber and increase the combustion chamber output. Sufficient energy of the secondary air jets is the main requirement for a favorable effect. The amount of secondary air and exit velocity can be guessed from the known experimental results and they can be calculated approximately from load, type of fuel, and primary air supply. Secondary air amounts of about 10% of the total air and velocities from 50 meters per second upward have been shown to be effective. The influence of the height of the nozzle installation above the grate has not been investigated sufficiently. However, it is senseless to install the nozzles very high. A downward direction of the nozzles is advisable although here too, except from the theory of Gumz, no exact investigations have been made. It is advisable to install the nozzles in the front or rear wall of the combustion chamber [not the sidewall]. Nozzles which blow against one another, especially in a dislocated arrangement, are undoubtedly good.

36. ZWEITLUFTZUFUHRUNG ZUM EINHALTEN DES WIRTSCHAFTLICHEN CO_2-GEHALTS DER RAUDHGASE (Secondary Air Supply to Keep an Economic Co_2 Content of the Off-Gases)
Obering. Mortensen
Wärme 60 (49), 799-801 (4 December 1937)

A new firing technique from Scandanavia is introduced and experimental results from several plans are given. The new technique consists of dividing the combustion process into two combustion processes, in which the first stage is a partial combustion with partial gasification and the complete combustion takes place in the second stage by the use of overfire air.

39. SULLE POSSIBILITA DI MIGLIORARE IL RENDIMENTO DELLE CALDAIE INSUFFLANDO ARIA SUPPLEMENTARE SOPRA IL COMBUSTIBILE (On the Possibility of Improving Efficiency of the Boiler by Blowing Supplementary Air Over the Fuel)
Antonio Rasi
Energia termica 7, 202-6 (1939)

This paper reports some results obtained in combustion tests carried out at the Instituto di Fisica Tecnica of the University of Padova. The boiler used was provided with a regulator for controlling forced overfire air. The advantages of its use are discussed.

42. GASBLANDNINGENS INFLYTANDE PA FORBRANNINGSHASTIGHETIN I FLAMMOR (The Influence of Gas Mixing on Combustion Velocity in Flames)
John Rydberg
Feuerungstech. 30 (11), 257-9 (15 November 1942)

From his investigations, Rydberg concludes that a high energy consumption does not guarantee a good mixing result. It is better to mix the secondary air into the combustion chamber through many different nozzles in order to promote the mixing if the minimum secondary air pressure and velocity can be kept. It is particularly advantageous if the gas flow can be narrowed down simultaneously by the addition of secondary air. Thereby, the flame length can be shortened, a condition which otherwise can only be obtained by a larger amount of excess air.

43. WANDERROSTFEUERUNG MIT WIRBELLUFTZUFUHRUNG (Traveling Grate with Turbulent Air Supply)
W. Schultes
Arch. Warmewirtsch. u. Dampfkesselw. 16 (5), 117-8 (1935)

A modern form of secondary air supply for traveling grate firing is described. It uses sharp, narrow aerojets of high penetrating power for the whirling of combustion gases. This method offers advantages for the elimination of smoke, soot, and unburned gases.

57. DIE WIRKUNGSWEISE VON WIRBELLUFTDUSEN (How Jets Promote Turbulence)
Karl Cleve
Feuerungstech. 25 (11), 317-22, 362 (1937)

The state of the knowledge is given of the characteristics of air which is introduced by means of "vortex air jets" into the combustion chamber. New experiments are described which centralize velocity, and the expansion of the jet after the exit from the air nozzle is discussed. The experimental results have been applied and a proposal made for an appropriate supply of turbulent air.

58. K VOPROSU O RASCHETE OSTROGO DUT'YA (On Overfire Air Jet Calculations)
I. K. Nairmark
Sovet. Kotloturbostroenie 7, 253-7 (1939)

The article gives formulae for calculating the trajectory of an overfire air jet of a fixed size, the initial speed of the jet and the direction of the nozzle. The trajectory of the jet curls when there is a variation of specific gravity of the air blast and hot gases, and also due to the action of furnace gases on the jet stream.

67. BERICHT UBER UNTERSUCHUNGEN ZUR AUSGEBILDETEN TURBULENZ (Report of Investigation of Developed Turbulence)
L. Prandtl
Z. angew. Math. u. Mech. 5, 136-9 (1925)

In this article the author describes a model for the turbulent flow near the wall. As an extension of the well-known model of the turbulent mixing, the author outlines the procedure to calculate free turbulent flow. The term free turbulent flow is used for flows without bordering walls. An outline is given only for stationary flows. The solution of the fundamental differential equation will be published later from Dr. Tollmien.

68. DER EINFLUSS DES MISCHVORGANGS AUF DIE VERBRENNUNG VON GAS UND LUFT IN FEUERUNGEN. I. THEORETISCHE VORBEMERKUNGEN (The Influence of the Mixing Process on the Combustion of Gases, Fuel and Air in Furnaces. I. Introductory Theoretical Remarks)
Kurt Rummel
Arch. Eisenhuttenw. 10 (11), 505-10 (May 1937)

The different states of combustion and degrees of combustion are explained and the importance of the combustion mechanism is stressed. Visual flame and gas analysis are normally insufficient to describe the degree of combustion. Therefore, the state of combustion is described by the figures L-bar for the amount of air, G-bar for the amount of unburned gas, R-bar for the amount of reacted gas. Using these figures, the areas of the flame are described.

69. DER EINFLUSS DES MISCHVORGANGS AUF DIE VERBRENNUNG VON GAS UND LUFT IN FEUERUNGEN TEIL. II. VERSUCHE AN DER BRENNERSTRECKE (The Influence of the Mixing Process on the Combustion of Gases, Fuel and Air in Furnaces. II. Experiments with the Firing)
Von Kurt Rummel
Arch. Eisenhuttenw. 11 (1), 19-30 (July 1937)

From previously described experiments, it can be derived that:

a. The velocity of the mechanical mixing is decisive for the combustion in normal industrial furnaces. Whenever gas and air mix, they burn with a velocity which is infinitely greater than the one obtained normally.

b. As a result of this, the influence of the temperature on the velocity of the combustion in the furnace should not be shown between 900 and 1300°. Probably, there is no influence of the temperature even in larger temperature limits, for example, between 600° and the dissociation temperature. At high temperatures we have a stronger expansion of the burnt gases which requires an accordingly increased space for the mixing.

c. Design of the body of the furnace has an influence on the shape of the space in which the mixing processes take place.

d. Space and time required for combustion decrease with an increase in the difference of the velocity of the parallel jets of a burner with gas and air. This is due to the increased turbulence at the boundary of gas and air.

e. Space and time required for combustion increase with a decrease of the exit velocity of gas and air from the burner as less energy is available for the mixing process.

f. Excessive air lessens the combustion of the gases with respect to space and time during the whole combustion process, not only at its end.

g. The space in which the combustion of gas and air takes place, that is, the space in which air, gas and burnt gas can be simultaneously found in the analysis, forms a cone which starts at the mouth of the burner. Excessive gas or air diverge from this burner.

h. The buoyancy forces are very small in a vivid jet of given specific weight in the body of the furnace in which the temperature differences are on the order of a few hundred degrees.

i. If an air jet lies on top of a gas jet of lower specific weight and the difference of the specific weight counteracts the buoyancy forces, this appearance can be used to guide the flame. The difference of the specific densities causes the stronger mixing and shortens the space and time required for combustion. The composition of the "atmosphere" is extremely different as long as the mixing of gas and air is incomplete. It should be determined whether flat burners should be used instead of round burners in cases where the reducing or oxidizing atmosphere is warranted at particular points of the furnace.

70. DER EINFLUSS DES MISCHVORGANGS AUF DIE VERBRENNUNG VON GAS UND LUFT IN FEUERUNGEN TEIL. III. MODELLVERSUCHE UBER DIE MISCHUNG VON GAS- UND LUFTSTRAHLEN (The Influence of the Mixing Process on the Combustion of Gases, Fuel and Air in Furnaces. III. Model Attempt for the Mixing of Gas and Air Jets)
Von Kurt Rummel
Arch. Eisenhuttenw. 11 (2), 67-80 (August); (3), 113-23 (September); (4), 163-81 (October 1937)

This paper deals with the following topics:

Similarity of the Models - the relationship between the mixing factors and the composition of the gas and air atmosphere; The Experimental Plan - the influence of the dimensions, the load, the velocities, the jet direction and repulsion, the influence for burners of the design of the SM-furnaces, before the entrance to the combustion chamber, of the dead corners of the combustion chamber, etc.

71. DER EINFLUSS DES MISCHVORGANGS AUF DIE VERBRENNUNG VON GAS UND LUFT IN FEUERUNGEN TEIL. IV. NACHPRUFUNG DER ERGEBNISSE DER MODELLVERSUCHE (The Influence of the Mixing Process on the Combustion of Gases, Fuel and Air in Furnaces. IV. Summary of the Results of the Furnace Investigations)
Von Kurt Rummel
Arch. Eisenhuttenw. 11 (5), 215-24 (November 1937)

The theory which was developed in References 68-70 was examined in a furnace and found to be correct in practice. Admittedly, the judgment of the practical cases was often difficult. Only fundamental cases could be examined in this work. Most often, however, numerous influences work together and oppose one another in industrial furnaces. The aerodynamics of firing is the result of complicated relations. However, in addition to the investigations of fundamental cases, the work has shown the applicability of the model attempts and represents, therefore, an important mean. The investigators design a simple burner for gas which allows, in a wide range of furnace conditions, the quick adjustment of the flame length and of the atmosphere over the furnace bottom.

72. LAMINARE STRAHLAUSBREITUNG (Laminar Jet Expansion)
H. Schlichting
Z. angew. Math. u. Mech. 13, 260-3 (1933)

Theoretical mathematical analysis of laminar spread of fluid jet, issuing from small rectangular or circular orifice, within surrounding fluid. [Eng. Index, 479 (1933)]

77. BERECHNUNG TURBULENTER AUSBREITUNGSVORGANGE (Calculation of Turbulent Expansion Processes)
Walter Tollmien
Z. angew. Math. u. Mech. 6 (6), 468-78 (1926)

The article treats the turbulent mixing of the homogeneous air jet with its surrounding air.

81. UBER DIE STROMUNGSVORGANGE IM FREIEN LUFTSTRAHL (On the Flow Processes of Free Air Jets)
W. Zimm
VDI-Forsch. Gebiete Ingenieurw. 11 (234), 5-35 (1921)

A free air jet is investigated whose velocity is low enough so that the compressibility of the air does not have to be regarded. The expected flow processes are derived in a theoretical way from an expanding nozzle whose outer walls are increased and taken away in the process of transition to a free jet. The "secondary air movement" occurs as a characteristic appearance of the free flow. Description is made of the instruments used in the experiment to produce the air jet and to measure the velocity. (The pitot stagnation instrument and electrical heat wire instrument are from Professor Weber). For precise measuring of the flow in the downstream region, only the sensitive heat wire instrument is used which is calibrated with a pitot-tube and which is continuously re-examined. Experimental investigation of the turbulent jet region goes up to 3.5m actual width and up to .1m/sec velocities. The analysis and plotting of the achieved experimental results with respect to the flow process, the participating amounts of air, and the kinetic energy were disturbed by the secondary air which influences the measurements in the outer boundaries by vortexes. The experimental results prove the theoretical assumption of a secondary air movement in the case of a free air jet. The amounts of secondary air increase with an increasing distance from the center and an increased velocity of the primary air and discharge the kinetic energy of the original jet. The initial energy is transferred with considerable losses onto the secondary air jet. This one represents then with its single elements a new primary gas which continues its energy dissipation into the space. The general validity and the broadening of the appearances, which have been found for a jet with a circular cross-section, is now considered. The picture, which has been found in the framework of these investigations for the flow processes in a free unhindered expanding air jet of low velocity can be valuable for the purposes of the technique and can be regarded as a preparatory work for the complete analysis of the free air jet.

102. GRAPHISCHES BERECHNUNGSVERFAHREN FUR MEHRSTUFIGE DAMPFSTRAHLAPPARATE (Graphical Method for Computing Multiple Steam-Jet Installations)
Ladislaus Zimmermann
Chem.-Ing.-Tech. 25 (11), 665-71 (1953)

A comprehensive treatise containing nomographs relating important operating variables. Examples are given for a four and five-stage unit. [Chem. Abstracts 48, 1736 (1954)]

119. BETRIEBSERFAHRUNGEN MIT DEM NEUEN TURBINENROST (Operating Experiences with New Turbine Grate)
Anon
Arch. Warmewirtsch. 6 (11), 299-301 (1925)

Describes new furnace and grate design for uniform air distribution over the entire grate; results show that this furnace is very suitable for low grade fuels. [Eng. Index, 84 (1925)]

122. FEUERUNGSTECHNIK DES WANDERROSTES EINFLUSSE VON BRENNSTOFF, LUFT UND ROSTBAUART (The Firing Technique of the Traveling Grate. The Impact of Fuel, Air and Grate Design)
Karl Beck
Arch. Warmewirtsch. u. Dampfkesselw. 20 (4), 93-8 (1939)

This paper is written mainly for men from power stations who want to achieve a clearer view of the present state-of-the-art of combustion processes on traveling grates and the influences of varying plant conditions. Factors which are mainly regarded while designing the grate will not be considered here. The article discusses some findings of A.R. Mayer and is, in addition, a very good introduction to the operation of the traveling grate.

127. ROSTWIDERSTAND VERSCHIEDENER KOHLENSORTEN (Combustion Resistance of Various Types of Coal)
Roman Dawidowski
Z. oberschles. bert- u. huttenmann. Ver. Katowice 65 (10), 660-7 (October); 728-34 (November 1926)

Discussed flow of air of combustion in fuel bed for various kinds of coal; adaptability of grates to coals; gas friction in fuel beds; velocity of flow, results of experiments. [Eng. Index, 82 (1926)]

131. PHYSIKALISCHE THEORIE DER VERBRENNUNG (Physical Theory of Combustion)
W.H. Fritsch
Wärme 60, 749-57 (13 November); 768-73 (20 November 1937)

Factors influencing the efficiency and rate of combustion in a restricted combustion space, e.g., a boiler furnace, are discussed. The velocity of the chemical reactions involved is such that the time necessary for their completion is only about 0.001% of that actually taken. The observed velocity must therefore be determined by physical factors, e.g., by the rate of mixing of combustible with air. The mechanism of air penetration through a fuel bed and of the mixing of gases or liquids by turbulence is discussed in relation to data afforded by experiments with models. The introduction of secondary air is not an effective method of improving the mixing of gases in a combustion space. It is concluded that more efficient combustion in a boiler furnace is to be effected only by radical changes in furnace-grate design, thus permitting better mixing of combustible gases. By overcoming the physical resistance to combustion, much higher rates of heat release per hr. per cu. m. of combustion space could be obtained. [Chem. Abstracts 33, 2310 (1939)]

137. DER VERBRENNUNGSVORGANG IN DER WANDERROSTFEUERUNG (The Process of Combustion with Chain Grates)
W. Gumz
Feuerungstech. 24, 10-2 (1936)

Meier investigated the combustion process on a traveling grate. Experiments were made in a normal industrial furnace. Thus, the mistakes or side influences of small lab instrumentation could be avoided. The grate was operated in the zoneless way and secondary air was added in various small amounts. Brown coals were used . In the dissertation, the percentages of coal, volatile compounds and ash have been determined and measurements of the CO_2 concentration in the combustion chamber have been made. It was concluded that the speed of combustion and pyrolysis mainly depends on the heat transfer within the layers.

138. ZONENEINTEILUNG UND ZWEITLUFTZUFUHRUNG BEI WANDERROSTEN (Division into Zones and Secondary Air Supply for Chair Grate Firings)
Wilhelm Gumz
Feuerungstech. 30 (11), 256-7 (1942)

The advantages of the division into zones has been illustrated by showing the composition of the combustion gas in the body of the furnace. Complete burnout for lignite could not be obtained and the additional help of secondary air was employed to reach a complete burnout.

143. DE LA COMBUSTION DES MATIERES VOLATILES (Combustion of Volatile Materials)
Victor Kammerer
Bull. Soc. indus. Mulhouse 92 (2), 111-33 (1926)

A discussion of the mechanism of combustion on mechanical stokers, in the light of the results of a certain number of boiler tests, with practical indications on conditions required to obtain optimum results with coals having medium and high volatile contents. [Chem. Abstracts 20, 2241 (1926)]

144. ZUR AERODYNAMIK DER BRENNSTOFFSCHUTTUNG IN ROSTFEUERUNGEN (On the Aerodynamics of Fuel Charging in Grate Firing)
H.G. Kayser
Forsch. Gebiete Ingenieruw. B. 6 (2), 89-100 (March-April 1935)

Aerodynamics of fuel layer in grate furnaces; theoretical considerations; mulitgrain mixtures; fuel layer as discontinuous system; it is claimed results of investigation can be applied to other fields of engineering where flow through granular materials play a role. [Eng. Index, 105 (1935)]

146. VERSUCHE AN WANDERROSTFEUERUNGEN (Experiments on Chain-Grate Furnaces)
W. Koeniger
Arch. Warmewirtsch. 10, 243-8 (1929)

In earlier experiments with traveling grate plants, variation in load had almost no influence on the degree of efficiency of the grate firings if the pressures were not varied substantially. The efficiency varied only slightly under conditions ranging from half-load to 25% overload. The influence of storage heat on the load variations, grate and combustion chamber loads, law of draft resistance and change of draft by buoyancy forces are discussed. [Chem. Abstracts 23, 4106 (1929)]

151. DIE SCHUTTHOHE EINER ROSTFEUERUNG (The Height of the Bed in Grate Firing)
A.R. Leye
Brennstoff- u. Warmewirtsch. 17 (2), 15-21 (1935)

Regulation of load on boiler grates by depth of fuel bed and passage of air; laws governing combustion process in coke layer; furnace should be designed to dispense as far as possible with secondary air admission. [Eng. Index, 105 (1935)]

153. VERBRENNUNGSVERLAUF VON STEINKOHLE AN EINER WANDERROSTFEUERUNG (The Combustion of Bituminous Coal on a Traveling Grate)
Rud Loewenstein
Wärme 57, 97-101 (17 February); 121-5 (24 February 1934)

An experimental apparatus and an experimental procedure have been developed for traveling grate firings. Experiments were carried out with different fuel layer heights at equal grate loads on the traveling grate of a 30-atmosphere-vertical-tube boiler with natural draft. Measurements were made of the air temperature under the grate, the grate rod temperatures, the air velocity and the gas composition over the layer. The findings of the combustion process are, compared with the results of former investigations of experimental firings. Conclusions for zone division and underwind blast pressure for gas to coal ratio and advantages and disadvantages of large fuel layer heights are given.

154. BEITRAGE ZUR FEUERUNGSTEDHNIK VON STEINKOHLEN AUF DEM WANDERROST (Contribution to Firing Practice of Bituminous Coal on a Traveling Grate)
Marcard
Warme 55, 397-401 (11 June 1932)

See Reference No. 154

155. BEITRAGE ZUR FEUERUNGSTEDHNIK VON STEINKOHLEN AUF DEM WANDERROST
(Contribution to Firing Practice of Bituminous Coal on a Traveling Grate)
Marcard
Wärme 55, 417-22 (18 June 1932)

Contributions to technique of firing bituminous coals on traveling-grate stoker development of traveling grate for high-capacity combustion equipment; influence of fuels, air distribution and furnace design on ignition and combustion. [Engineering Index 22, 1282 (1932)]

156. DIE VERBRENNUNG ALS STROMUNGSVORGANG (Combustion as a Flow Process)
W. Marcard
Wârme 60, 257-66 (24 April 1937)

Topics discussed are: the importance of fluid dynamics for the combustion technique; general form of the combustion process; individual combustion processes; change of state for liquid and gaseous fuels; air flow under and in the grates; elimination of combustion gases and mixing; off-gas explosion, explosion weight; percentage of parts participating in the heat transfer; calculation of temperatures at furnace end; new ways of boiler design.

160. DIE WIRKUNG DER ZWEITLUFT IN DER WANDERROSTFEUERUNG (D84)
(The Effect of Secondary Air on Traveling Grate Firing)
Albert R. Mayer
Z. bayer. Revisions-Ver. 42 (4), 31-3 (February 1938)

This reference covers the first part of a thesis discussion from Mayer. It deals with combustion processes on the traveling grate firing. It points out that the combustion process can only take place after gas and air have come together by processes like diffusion, turbulence and "free whirl formation." These processes are discussed.

161. DIE VORGANGE IM FEUERRAUM EINES KESSELS MIT WANDERROSTFEUERUNG UND IHRE ANDERUNG DURCH ZWEITLUFTZUFUHR (The Reactions in a Boiler Furnace and Their Alteration with Twin Air Supply)
Albert R. Mayer
Feuerungstech. 26 (5), 148-50 (1938)

It is shown by analyses of gas samples taken from numerous positions in the combustion chamber that conditions therein are unsatisfactory with a great load of 80-90 kg/sq m/hr., corresponding to 75% of the normal boiler capacity. The gas composition can be made uniform throughout the chamber by introducing secondary air, which also accelerates the combustion can be obtained by suitable adjustment of the quantity and velocity of the secondary air, thus affording a means of controlling combustion and increasing the efficiency of the combustion chamber. [Chem. Abstracts 33, 4405 (1939)]

162. UNTERSUCHUNGEN UBER ZWEITLUFTZUFUHR IN WANDERROSTFEUERUNGEN (Investigation of Secondary Air Supply in Traveling Grate Firings)
Albert R. Mayer
Feuerungstech. 26 (7), 201-10 (1938)

Gas composition is determined in the body of a furnace burning coal with high and low grate load. Also noted are the caloric values of the combustion gases, the influence of secondary air on the burnout of the combustion gases, relationship of secondary air to the required height of the combustion chamber and side effects of secondary air.

166. VERFEUERUNG GASREICHER KOHLE IN EINER NEUZEITLICHEN WANDERROSTFEUERUNG (Combustion of Gas-Rich Coal on a Modern Traveling Grate)
Albert Muller
Z. Reichshauptst. Techn. Uberwach. 3 (11/12), 61-72 (13 June); (13/14), 76-81 (11 July 1942)

On account of the numerous experiments and investigations, it has been realized that the combustion process on the grate as well as in the combustion chamber of modern traveling grate firings has still to be improved, particularly if gas-rich brands with long flames are used. Experiments were conducted with this in mind. These experiments were carried out to determine the advantage of the zone regulation of the traveling grate or the combustion process on the grate and the burnout of the gas in the combustion chamber versus the zone length operated grate. In addition, the influence of the secondary air on the combustion of the gas in the combustion was investigated. Experiments show that it is particularly difficult to achieve complete combustion while using gas-rich coals in high combustion chambers of modern radiator boilers with short ignition bodies at an economical air excess. The role of the combustion chamber height was to be investigated with respect to the burnout of the combustion gases for primary air as well as secondary air operation of the firing. In order to answer these different problems, a special Bavarian coal with 36% volatile compounds was burned in a modern traveling grate firing and the combustion process on the grate as well as in the combustion chamber was observed. To achieve this, the composition of the coal was analyzed at different loads from the layer regulator to the slag stopper. In addition, the pyrolyzing gases from the coal layer were analyzed in several heights above the grate. Thereby, it was shown to be efficient to compare certain tests with zone regulation with one of the same loads but without the zone regulation [from W. Meier]. Knowledge of coal and gas analysis allows one to theoretically obtain, most importantly, the amount of air which enters through the grate and to obtain, especially, the adaptation of the air throughput to the locally required amount of air for the whole grate length theoretically. In this way, the great advantage the regulated air supply to the fuel has over the zoneless grate operation is shown. In addition, the total heat history of the coal burned on the grate and in the combustion chamber can be shown. Secondary air on the gas burnout was investigated for larger load variations as a function of amount and entrance velocity by gas samples from different heights above the grate. The findings of the new investigations allow one to determine which amounts of secondary air and which air entrance velocities achieve a favorable operation for gas-rich coals in a traveling grate firing. Finally, proposals are made for the choice of the average combustion chamber height.

167. VERIEUERUNG GASREICHER KOHLE IN EINER NEUZEITLICHEN WANDERROSTFEUERUNG
(Combustion of Gas-Rich Coal in Modern Traveling Grate Firing)
Albert Muller
Feuerungstech. 31 (3), 68-9 (1943)

The plant of the Technical University of Munich, which has been investigated by W. Meier - dissertation Munich, 1935, as a zoneless traveling grate firing, has now been investigated with zone division and with secondary air supply. This was done in order to distinguish the impacts of the zone division and the secondary air supply. With zone division the burnout of the fuel layer is substantially different and travels more to the entrance than could have been expected. The processes in the combustion chamber are improved but the mixing effect is not yet sufficient to prohibit that in a combustion chamber, which is 5.75 meters high. Unburned gas can be found in the last measuring plant which is 4.65 meters over the grate. By addition of secondary air which amounted to 20% of the total air supplied, the CO could not be abolished completely but it decreased substantially. The result was improved with 30% secondary air. The effect of the secondary air was decreased with an increased combustion chamber load. However, the entrance velocity of the secondary air was from 10.9 to 32.6 meters per second which is obviously too low. In spite of this, we could increase, under these conditions and with a moderate air excess, the combustion chamber load to 295,000 kcals per cubic meter and hour, and we could decrease the combustion chamber height to 4 meters. The zone division loses a little of its importance if secondary air is supplied into the combustion chamber.

170. DIE GRENZEN DER FEUERRAUMBELASTUNG UND IHRE RUCKWIRKUNG AUF DIE AUSLEGUNG DES KESSELS (Limits of the Combustion Chamber Load and the Resulting Effect on the Design of the Boiler)
W. Pauer
Arch. Warmewirtsch. u. Dampfkesselw. 20 (8), 197-202 (1939)

The problem of the combustion chamber load which is determined by the burning time of the dust particles and the allowed combustion chamber temperature is solved substantially for coal dust firings. The applicability of the relations developed for coal dust firings to gas and oil firings has not been pointed out clearly. The generalization that the permissible load is only a time function, which is valid for gas- and oil-firing, is not confirmed. It is the purpose of this paper to examine the validity of the famous permissible load formulas and clarify their applicability for the different kinds of firings.

172. DIE VERBRENNUNG VON BRAUNKOHLEN AUF DEM ARBATSKY-WANDERROST (Combustion of Brown Coal on the Arbatsky Traveling Grate)
E. Praetorius
Braunkohle 30, 241-7, 266-9 (1931)

Tests at the Power Station Gruenberg on a provisionary Arbatsky-grate have shown that the combustion of brown coal on traveling grates with a high grate space and width load is not only possible, but that width loads can be obtained which up to now could only be obtained from coal firing and coal high performance grates. It should not be difficult to improve the degree of efficiency over one obtained under unfavorable test conditions, from 63% to about 80% and higher. It will be possible to obtain, with the use of the Arbatsky grate for brown coals and other low-caloric fuels, 200 square meter heat transfer surfaces for one meter width of boiler and it will be possible to achieve heat transfer surface outputs of 50-60 kilograms per square meter.

177. ZUR PHYSIK DER VERBRENNUNG FESTER BRENNSTOFFE (The Physics of Combustion of Solid Fuel)
P. Rosin and H.-G. Kayser
Z. Ver. deut. Ing. 75 (26), 849-57 (27 June 1931)

The following topics are discussed: physical relationship between the process of coal combustion and the solution of solid bodies; model experiments for dissolving salt in water currents; weight and time of the reactions for single bodies; the aerodynamics of the grate; the behavior of multiple body heats at high air velocities; instability and resistance to fly coke; aerodynamics of the dust firing; limiting load and particle size; load regulation as a function of the amount of air; fuel layer and grate steering, regulation of fire zoning; air regulation according to conditions of the combustion process and behavior of the ash.

178. DIE RAUMLICHE UND ZEITLICHE ENTWICKLUNG DER VERBRENNUNG IN TECHNISCHEN FEUERUNGEN (The Change of Combustion with Respect to Time and Space in Industrial Furnaces)
Kurt Rummel and Hellmuth Schwiedessen
Arch. Eisenhuttenw. 6 (12), 543-9 (June 1933)

The combustion process is not sufficiently described by measuring temperature, velocity and composition; particularly the state of the combustion in one point or in one plane of the furnace cannot be predicted. Qualitative description is only possible by introducing a new definition for the degree of combustion in one point and by plotting the curves of equal degree of combustion. Mathematical derivations show, in addition, that qualitative descriptions can be made too if the degree of combustion is related to the volume which is enclosed by areas of equal degree of combustion [volume of reaction]. Additional derivations allow predictions of the velocity with which the combustion proceeds to the practical degree of combustion [average related load] and the combustion time and weight. The derivations are explained and proved by practical examples.

179. STAND UND ENTWICKLUNG DER FEUERUNGSTECHNIK. EIN QUERSCHNITT UND UMRISS UBER DIE FORSCHUNG AUF DEM GEBIETE DER FEUERUNGSTECHNIK IN DEN LETZTEN JAHREN (State and Development of Firing Techniques. A Summary of Research in the Field of Firing Techniques in Recent Years)
Fr. Schulte and E. Tanner
Z. Ver. deut. Ing. 77 (21), 565-72 (27 May 1933)

The heater tube imperical development of the firing technique requires an extension towards the physical side of the up-to-now almost purely chemical combustion research. New knowledge of the combustion process is discussed by limiting ourselves mainly to the discussion of coal. The chemical reaction of the layer and the air required in the individual steps of combustion are discussed in a way suited for the practical preparation of firings. The processes of ignition and ignition throughout the fuel layers are discussed, and important results for cold dust firing are summarized. Results supported by weight and temperature measurements are mentioned for the whirling of the ash from the grate, the formation of slag, and the wear of the grate. It was found that a certain minimum content of ash is necessary for the burning of the coal on the grate to achieve low wear-and-tear of the grate. The flow pattern in the body of the furnace and in the fuel layer, the gas velocity, the gas composition and the temperature distribution are briefly discussed.

180. VERBRENNUNGSPROBLEME IN BRENNKAMMERN VON HOCHLEISTUNGSDAMPFKESSELN
(Combustion Problems in Combustion Chambers of High Performance Boiler Furnaces)
Karl Schwarz
Brennstoff-Warme-Kraft 1 (2), 45-52 (May 1949)

Problems created in a high performance boiler furnaces, like increased danger of clogging and corrosion, are viewed according to firing technique. These problems result from the increased use of fuels with a high content of ash and water. In particular, those cases are discussed which are due to incomplete combustion. The flow and mixing problems in the off-gas combustion air flow are explained for two practical cases, that of the modern grate boiler and that of the dust boiler.

188. VERBRENNUNGSVERLAUF BEI STEINKOHLEN MITTLERER KORNGROBEN
(Combustion of Coal of Medium Particle Size)
Helmut Werkmeister
Arch. Warmewirtsch. u. Dampfkesselw. 12 (8), 225-32 (1931)

The methods used hitherto for calculating the firing technique and examining the fuel chemistry produced imprecise knowledge of the process of combustion of solid fuels. For the design and operation of firing, it is, however, very important to control the combustion process. In an extensive test series, the combustion of nut coal with different gas contents was investigated with a special process. The results are given in the form of characteristic burning lines as a function of the combustion time. Approximate formulas for calculating the technical combustion process have been used.

189. WINDVERTEILUNG UND FEUERGASBESCHAFFENHEIT BEI WANDERROSTFEUERUNGEN
(Blast Distribution and Combustion Gas Composition in Traveling Grate Firing)
H. Werkmeister
Z. Ver. deut. Ing. 78 (26), 788 (30 June 1934)

Blast pressure is discussed as a function of height of the coke layer, particle size and combustion of the layer.

191. BEKAMPFUNG DER VERSCHLACKUNG VON DAMPFKESSELN (Fighting of Slagging in Boiler Furnaces)
Arthur Zinzen
Brennstoff-Warme-Kraft 2 (3), 63-8 (1950)

The general improved melting diagram for fuel ashes allows one to judge about the behavior of ashes. The chemical reactions of combustion have been investigated for the luminous flame and the zones in which the unburned gases, the sulfur compounds, which originate in the flame, and the fly coke have to be burned. The conclusions for the planning and the carrying out of boiler furnace firings are meant to show how to prevent boiler slagging.

205. DAMPFKESSELFEUERUNGEN (Steam-Boiler Furnaces)
H. Berner
Z. Ver. deut. Ing. 65 (15), 371-5 (9 April 1921)

Among the topics discussed are: the changed requirements of furnaces; fuels of low quality as seen from the furnace technical viewpoint; the combustion process and loss of output with fuels of low quality; the possibility of the general furnace; mechanical draft; variation in furnace losses; the different types of grates.

209. BESSERER FLAMMENAUSBRAND IM FEUERRAUM DURCH FLAMMENWIRBELUNG VERFAHREN, MOGLICHKEITEN UND BETRIEBSERGEBNISSE (Superior Flame Combustion in the Furnace by Means of Flame Turbulence)
Karl Cleve
Arch. Warmewirtsch. u. Dampfkesselw. 20 (6), 149-53 (1939)

Greater furnace capacity can be obtained by the use of a turbine-type burner, by the constriction of the furnace cross section above the grate, and by the use of high-pressure secondary air, properly directed. These all depend on good mixing of the flame parts. A diagram is given showing air velocities in front of a three-part turbine-type burner, when the parts are used separately and together. [Chem. Abstracts 34, 241 (1940)]

222. DIE WIRTSCHAFTLICHKEIT DES TORF-DAMPFKESSELBETRIEBES (The Economy of Peat-Boiler Operation)
A.H.W. Hellemans
Feuerungstech. IV (11), 126-31 (1916)

Proper design and operation of boilers for burning fuel is discussed. [Eng. Index, 259 (1916)]

230. STAND U. ENTWICKLUNGSZIELE DER MODERNEN STEINKOHLENFEUERUNGSTECHNIK (State and Development of Modern Coal Firing Practice)
W. Kretschmer
Intern. Bergwirtsch. u. Bergtech. 24, 169-72 (15 August); 192-6 (15 September); 211-15 (15 October 1931)

The ignition and combustion operation of the coal is first explained in order to ascertain the proceedings in the firing. Finally the different constructions of coal firings are dealt with and it will be seen to what extent these already comply with the previously laid down requirements.

232. DER NEUE MODERNE FEUERUNGSROST (New Modern Grate)
J. Lauf
Der Bergbau 38 (19), 329-33 (6 May 1925)

Details of turbine furnace, a forced draft furnace regulated by means of steam-jet nozzle; manual stoking; claimed to possess all advantages of economic operation. [Eng. Index, 86 (1925)]

253. LES FOURS A FLAMMES (The Circulation of Hot Gases in Furnaces)
H. Drouot
Tech. Mod. 14, 151-7 (1922)

W.E. Groume-Grjimailo works out the theory of the circulation of the hot gases in furnaces by analogy from the laws of hydraulics. Small-sized sections of furnaces have been built and enclosed between parallel glass plates, and the laws verified experimentally by means of water and colored petroleum (representing the light hot gases), taking into account the facts that the hot gases and air are miscible, and that their density varies with temperature. The findings have been applied to furnaces already in operation or under construction, and on the whole they have given satisfactory results. Modern practice in furnace construction is discussed in the light of this theory, showing what features are defective and how to improve them. [Chem. Abstracts 16, 1997 (1922)]

255. ISSLEDOVANIE PROTSESSA TURBULENTNOGO GORENIYA C UCHETOM VTORICHNYKH REAKTSIYI (Investigation of Turbulent Combustion with Calculation of Secondary Reactions)
S.A. Gol'denberg
Izvest. Akad. Nauk SSSR Otdel. Tekhn. Nauk (5), 657-66 (1951)

Turbulent heterogeneous combustion, as carried out experimentally by flow of O at 500-1000° through cylindrical C tubes, is affected significantly by the secondary reactions of CO_2 reduction and CO combustion. Under these conditions the coefficient of gas formation = 4.3×10^7 exp-(-28,500/RT). [Chem. Abstracts 46, 2775 (1952)]

257. GRUNDLAGEN DER STROMUNGSTECHNIK DES INDUSTRIEOFENS
(Principles of Flow Techniques in Industrial Furnaces)
Michael Hansen
Arch. Eisenhuttenw. (11/12), 337-44 (November/December 1949)

The factors determining the flow of heat and gases, namely, continuity, turbulence, viscosity, Reynolds number, and design factors, are treated theoretically and methods for their determination are discussed. [Chem. Abstracts 44, 4294 (1950)]

261. STROMUNGSTECHNISCHE FRAGEN IM DAMPFKESSEL- UND FEUERUNGSBAU
(Aerodynamic Problems in Boiler and Furnace Design)
W. Marcard
Warme 56 (19), 291-4 (13 May 1933)

The importance of aerodynamics as an auxiliary science is stressed. The fundamental laws of aerodynamics are used in a model experiment in which the investigated media are water vapor, air and combustion gases. The experiments have been carried out in important parts of the boiler.

263. STROMUNGSTECHNISCHE BETRACHTUNGEN IM FEUERUNGS- UND DAMPFKESSELBAU
(The Hydromechanical Viewpoint in the Construction of Furnaces and Boilers)
F. Michel
Feuerungstech. 17 (23-4), 233-8 (15 December 1930)

Designers of boilers and furnaces have not learned to apply modern discoveries in the field of fluid flow. Proper attention to these principles should make possible better utilization of the available combustion volume reduction in friction losses and an improvement in heat transfer. [Chem. Abstracts 25, 1708-9 (1931)]

265. DER STROMUNGSVORGANG IN DER BRENNKEMMER VON ROSTFEUERUNGEN. EIN BEITRAG ZUR BERECHNUNG DER STROMUNGSVORGANGE AUF GRUND VON MODELLVERSUCHEN (The Flow Process in the Combustion Chamber on Grate Firing. A Contribution to the Calculation of Flow Processes from Model Experiments)
L. Schiegler
Z. Ver. deut. Ing. 82 (29), 849-55 (16 July 1938)

Examination of the combustion gas flow in all the furnace firings is a difficult task for which the general solution has not yet been found. In this paper, which will be continued in a second article, an approximate solution has been given for certain flow conditions resulting from model investigations. This solution has been confirmed by practical experience and allows, under certain preconditions, the calculation of given flow problems in the firing technique.

APPENDIX A

EQUATIONS FOR INVISCID NON-CONDUCTING FLOW IN TWO ZONES

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EQUATIONS FOR INVISCID NON-CONDUCTING FLOW IN TWO ZONES

Referring to Figure III-1 in Chapter III, we will assume that:

1. Pressure is constant across Section a.
2. u_{1a} and u_{2a} are different but constant across Section a in each zone.
3. h is constant across Section a.
4. ρ_1 is less than ρ_2 as the temperature is higher in Zone 1.

Because of Assumptions 1, 2 and 3, the total head, H, though different in the two zones is constant everywhere within each zone. At Section b, the streamlines are assumed parallel, so that the quantity $(p + \rho gh)$ is constant across the section; and as a result, the velocity is also constant. Considering the streamlines just on each side of the interface between the two zones, conditions at Section b can be related to those at Section a by the following equations:

$$\frac{u_{1b}^2}{2g} = \frac{u_{1a}^2}{2g} + \frac{(P_a - P_b)}{\rho_1 g} g_c - (y_b + A_{2b}) \qquad \text{(A-1)}$$

$$\frac{u_{1b}^2}{2g} = \frac{u_{1a}^2}{2g} + \frac{(P_a - P_b)}{\rho_2 g} g_c - (y_b + A_{2b}) \qquad \text{(A-2)}$$

Conservation of mass flow rate yields the following equations:

$$u_{1a} A_{1a} \rho_1 = u_{1b} (A - A_{2b}) \rho_1 \qquad \text{(A-3)}$$

$$u_{2a} A_{2a} \rho_2 = u_{2b} A_{2b} \rho_2 \qquad \text{(A-4)}$$

Equations (A-1) through (A-4) relate the variables u_{1b}, u_{2b}, A_{2b} and P_b and can be used to determine these quantities, given the conditions at Section a. Combination of the equations to eliminate all variables but A_{2b} results in:

$$A_{2b}^5 [-(\rho_2 - \rho_1)g] + A_{2b}^4 [(q_{2a} - q_{1a}) - (\rho_2 - \rho_1)g(y_b - 2A)]$$

$$+ A_{2b}^3 [-2A(q_{2a} - q_{1a}) - (\rho_2 - \rho_1)gA(A - 2y_b)]$$

$$+ A_{2b}^2 [A^2(q_{2a} - q_{1a}) + \frac{w_1^2}{2\rho_1} - \frac{w_2^2}{2\rho_2} - (\rho_2 - \rho_1)gy_bA^2] \qquad \text{(A-5)}$$

$$+ A_{2b} [\frac{w_2^2 A}{\rho_2}] - \frac{w_2^2 A^2}{2\rho_2} = 0$$

where:

$$(q_{2a} - q_{1a}) = \frac{1}{2} [\frac{w_2^2}{\rho_2 A_{2a}^2} - \frac{w_1^2}{\rho_1 A_{1a}^2}]$$

w_1 = flow rate per unit width in Zone 1.

w_2 = flow rate per unit width in Zone 2.

Solution of the above equation can be accomplished with a standard computer program for determining the roots of polynomials. Such a program has been used to obtain the results shown in Table III-2.

APPENDIX B

INSTABILITY IN STRATIFIED FLOW

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INSTABILITY IN STRATIFIED FLOW

As noted in Section C-1 of Chapter III, in a linear feed incinerator, the hottest gases are produced very near the input end of the burning grate, while the gases at the discharge end are cool because of the large percent excess air. If the flow of overfire gases is in the same direction as the fuel bed, the gases will naturally stratify with the hot, fuel-rich components forming a blanket along the roof of the combustion chamber. This hot blanket would also have a rather high velocity because of the accelerations produced by the rise to the roof (see Section D of Chapter III). An important question is whether the stable density distribution will significantly inhibit mixing of the hot gas with the cooler, oxygen-rich gases below.

The question can be answered by considering the stability of a stratified fluid of infinite extent with an initially parallel velocity field, as in Figure B-1. We are primarily interested in examining the competing effects of density in promoting stability and shear in generating instability, so it will be sufficient, at first, to take a simple two-strata model with the shear confined to a single interface. It is a convenient approximation to take the dense fluid to be infinitely deep. The mathematical solution to this problem is a modification of that given in Reference 2 pp. 159-163. The modification consists of inserting an upper boundary and dropping the effect of interface surface tension.

In each of the two regions, the disturbance velocity field is irrotational and may be subsumed in a potential function. These functions are denoted by ϕ_1 and ϕ_2, respectively, and each is a function of x, y (see Figure B-1) and time (t). It is convenient also to introduce the variable $\zeta(x,t)$ which represents the vertical displacement of the disturbed interface. There are five boundary conditions on these three variables. At the interface there are two kinematic conditions:

$$\frac{\partial \zeta}{\partial t} + u_1 \frac{\partial \zeta}{\partial x} = \frac{\partial \phi_1}{\partial y} \qquad \text{(B-1)}$$

$$\frac{\partial \zeta}{\partial t} + u_2 \frac{\partial \zeta}{\partial x} = \frac{\partial \phi_2}{\partial y} \qquad \text{(B-2)}$$

which exposes the fact that the vertical velocity of the fluid on each side matches that of the interface. There is also a dynamic interface condition which assumes continuity of pressure from one field to the other:

$$\rho_1\left(\frac{\partial \phi_1}{\partial t} + u_1 \frac{\partial \phi_1}{\partial x} + g\zeta\right) = {}_2\left(\frac{\partial \phi_2}{\partial t} + u_2 \frac{\partial \phi_2}{\partial x} + g\zeta\right) \qquad \text{(B-3)}$$

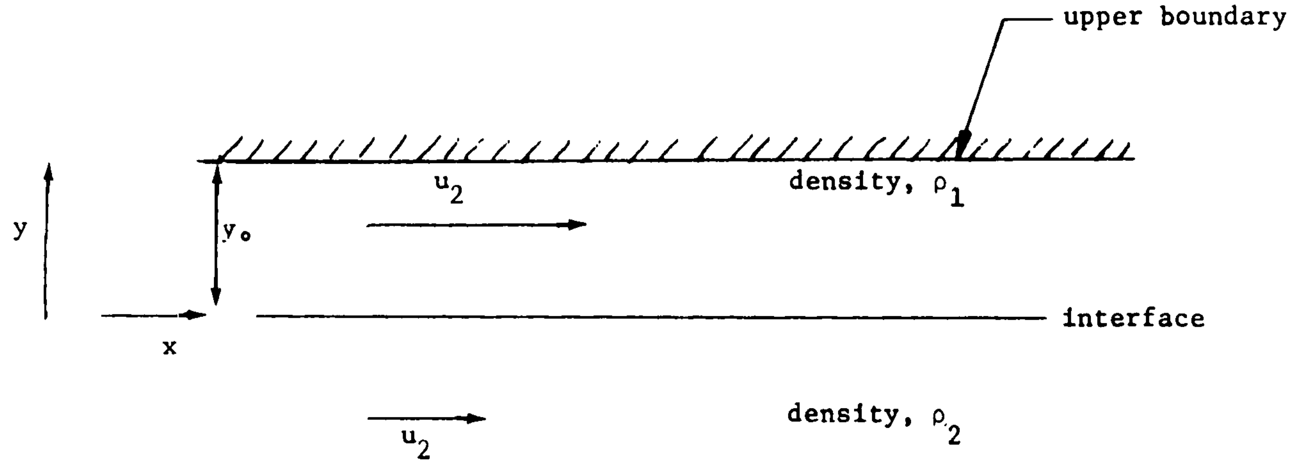


FIGURE B-1

SCHEMATIC OF STRATIFIED FLOW CONDITION

At the upper boundary, the vertical velocity component must vanish

$$\frac{\partial \phi_1}{\partial y} = 0 \qquad \text{at } y = y_\circ \tag{B-4}$$

while in the lower field the velocities must vanish at the limit of negative infinity

$$\frac{\partial \phi_2}{\partial y} = 0 \qquad \text{at } y \to -\infty \tag{B-5}$$

Following the usual procedure for problems such as this, a wave-type solution for each variable is assumed

$$\phi_2 = m_3 \exp[Ky + i\ (\sigma t - Ky)]$$

$$\phi_1 = m_1 \exp[-Ky + i\ (\sigma t - Kx)] + m_2 \exp[Ky + i\ (\sigma t - Kx] \tag{B-6}$$

$$\zeta = m_4 \exp[i\ (\sigma t - Kx)]$$

where the wavenumber, K, by definition, is positive and real. Each of the potential functions as chosen satisfies Laplace's equation (as it must) and the form for ϕ_2, automatically satisfies the boundary condition (B-5). Substition of (B-6) into (B-1) through (B-4) gives four algebraic equations:

$$i(\sigma - Ku_2)m_4 = Km_3$$

$$i(\sigma - Ku_1)m_4 = -Km_1 + Km_2$$

$$2[i(\sigma - Ku_2)C + gm_4] = \rho_1[i(\sigma - Ku_1)(m_1 + m_2 + gm_4] - m_1 e^{-Ky_\circ} \tag{B-7}$$

$$+ m_2 e^{Ky_\circ} = 0$$

These equations are linear and homogeneous in the coefficients m_1, m_2, m_3 and m_4. In order that non-zero solutions may exist, the growth exponent, σ, must satisfy the characteristic equation

$$(u_2 - \frac{\sigma}{K})^2 + [\frac{\rho_1}{\rho_2}\frac{e^{2ky_\circ} + 1}{e^{2ky_\circ} - 1}]\ (u_1 - \frac{\sigma}{K})^2 = (\frac{\rho_2 - \rho_1}{\rho_2})(\frac{g}{k}) \tag{B-8}$$

This quadratic equation in σ contains two parameters, one a geometric parameter related to the thickness of the low density layer:

$$G_1 \triangleq \frac{\rho_1}{\rho_2} \frac{e^{2Ky_o} + 1}{e^{2Ky_o} - 1} = \frac{\rho_1}{\rho_2} \coth Ky_o \qquad \text{(B-9)}$$

and another describing the effect of gravity acting on the density differential:

$$G_2 \triangleq \frac{g}{K} \frac{(\rho_2 - \rho_1)}{\rho_2} \qquad \text{(B-10)}$$

The solution of the quadratic is given by:

$$\frac{\sigma}{K} = \sqrt{\frac{u_2 + G_1 u_1 \pm - G_1 (u_1 - u_2)^2 + (1 + G_1) G_2}{1 + G_1}} \qquad \text{(B-11)}$$

Stability of the stratified flow is characterized by disturbances which propagate but do not grow in time, i.e., by real σ. If σ has an imaginary component, then the flow is unstable. From Equation (B-11), it is clear that the shear, $u_1 - u_2$, contributes to instability while a positive G_2 contributes to stability. The threshold of stability is found when the sum of terms inside the square root is zero, or when

$$\begin{aligned}(u_1 - u_2)^2 &= (1 + \frac{1}{G_1})G_2 \\ &= (1 + \frac{\rho_2}{\rho_1} \text{Tanh } Ky_o)\frac{g}{K} \frac{(\rho_2 - \rho_1)}{\rho_2}\end{aligned} \qquad \text{(B-12)}$$

If we regard ρ_1, ρ_2 and $(u_1 - u_2)$ as specified parameters, then (B-12) represents a condition on wavenumer, K. The wavenumber which satisfies (B-12) is a critical wavenumber which we denote by K_c. For $K > K_c$, the waves are unstable. A sketch of the growth rate (imaginary part of σ) versus Ky_o is shown in Figure B-2 for $\frac{\rho_1}{\rho_2} = .25$ and $(u_1 - u_2)^2/gy_o = 6$.

This result is unrealistic for large K because it predicts growth rate increasing without limit. The unreality is due to the assumption of an infinitely thin shear layer. What is needed is a problem specification which includes a continuous velocity profile and density profile as well. The solution of such a problem is not generally possible in closed form and requires a computer calculation. Since any velocity or density profiles would be highly conjectural, it does not seem worthwhile to pursue this in detail. There are,

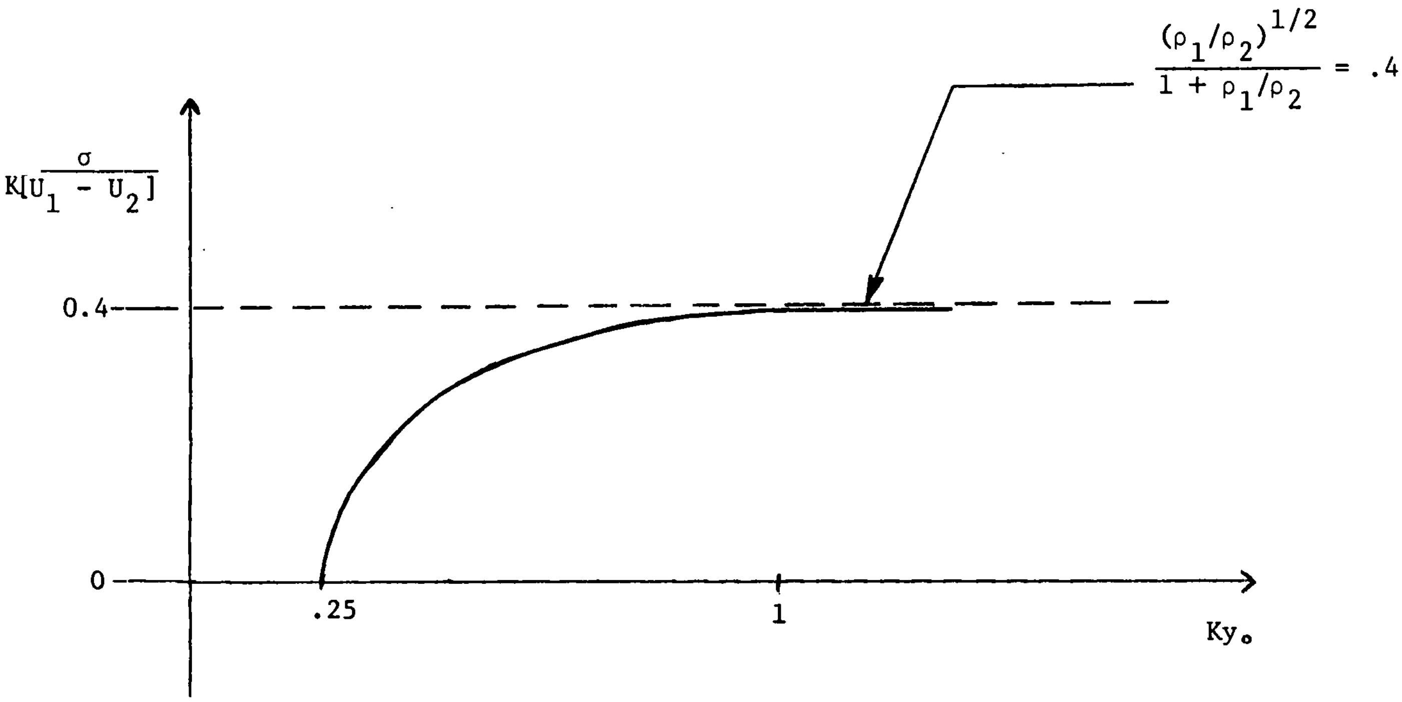


FIGURE B-2

GROWTH RATE OF DISTURBANCES IN NON-ISOTHERMAL PARALLEL FLOW however, certain well-known results which can be invoked to obtain an understanding of the relevant physics. For a fluid of uniform density, it is known[1] that if a shear layer is spread out over a thickness ψ, then all very small disturbances (very high K) are stable. There is a high wavenumber cut-off, K_{co}, above which stability is found, and $K_{co} \approx 1/\psi$. Even relatively large density differences have very small effect on this result.

The question at hand, then, is whether K_{co} is approximately equal to (or less than) K_c found previously, in which case the combined effects of density stratification and finite thickness of shear layer stabilize disturbances over the entire range of sizes. A rough answer can be given rather simply. Assuming that the dimension of the shear layer is roughly equal to the dimension h, we have $K_{co} \approx 1/y_\circ$. If we introduce a characteristic length

$$\nu = \frac{(u_1 - u_2)^2}{2g} \tag{B-13}$$

then (B-12) can be written

$$1 + \frac{\rho_2}{\rho_1} \tanh(K_c y_\circ) = \frac{2\nu}{y_\circ} \left(\frac{\rho_2}{\rho_2 - \rho_1}\right) (K_c y_\circ) \tag{B-14}$$

and since tanh $(K_c y_\circ)$ is never greater than unity, an upper bound for K_c is given by

$$K_c y_\circ \leq \frac{\rho_2^2 - \rho_1^2}{2\,\rho_1\rho_2} \cdot \frac{y_\circ}{\nu} \tag{B-15}$$

In order for the stratified flow to be completely stable, it is necessary that

$$\frac{y_\circ}{\nu} \frac{\rho_2^2 - \rho_1^2}{2\,\rho_1\rho_2} > 1 \tag{B-16}$$

or

$$\frac{\rho_2^2 - \rho_1^2}{\rho_1\rho_2} > \frac{(u_1 - u_2)^2}{g y_\circ} \tag{B-17}$$

This requirement is a stringent one which will not ordinarily be satisfied. This can readily be seen if we imagine the dense fluid to be at rest and the heated fluid to have achieved its velocity, u_2, by rising through a height L. Then

$$\rho_1 \frac{U_1^2}{2} = (\rho_2 - \rho_1)gL \quad , \quad U_2 = 0 \qquad \text{(B-18)}$$

and (B-17) becomes

$$y_o \frac{\rho_1 + \rho_2}{2\rho_2} > L \qquad \text{(B-19)}$$

In other words, the total rise must be less than the thickness of the heated layer iteslf. For most continuous-feed incinerator configurations, the rise of heated gases is larger than this, so the effect of density gradient will not eliminate turbulent mixing.

REFERENCES

1. "Dynamics of Non-Homogeneous Fluids", C.S. Yih, McMillan Company (1965).

APPENDIX C

DECAY OF SHEAR VELOCITY IN A CHANNEL

APPENDIX C

DECAY OF SHEAR VELOCITY IN A CHANNEL

To obtain an estimate of the rate of shear velocity decay as a result of turbulent mixing, we assume that the momentum exchange across the mid-planes of unconfined and confined jets are the same. For simplicity, we consider the approximations to Gaussian profiles shown in Figure C-1: M-M is the midplane. We will treat a "linearized" problem, where density and velocity gradients are relatively small compared to average values.

The momentum decrease in the high velocity region of the unconfined shear layer in the distance dx and, hence, the momentum transfer across M-M, is proportional to the shaded area in Figure C-1. Therefore, the rate of momentum transfer is proportional to

$$-\frac{1}{2}\left(\frac{u_1 - u_2}{2}\right)\frac{db}{dx} \tag{C-1}$$

Similarly, for the confined shear layer, the rate of momentum transfer across M-M with distance downstream is proportional to

$$\left[\left(\frac{A}{2} - b\right) + \frac{b}{2}\right]\frac{du}{dx} = \frac{1}{2}(A - b)\cdot\frac{du}{dx} \tag{C-2}$$

For the same fluids and velocities in the two cases, the proportionality constants will be the same, so that our assumption that momentum exchange is the same yields

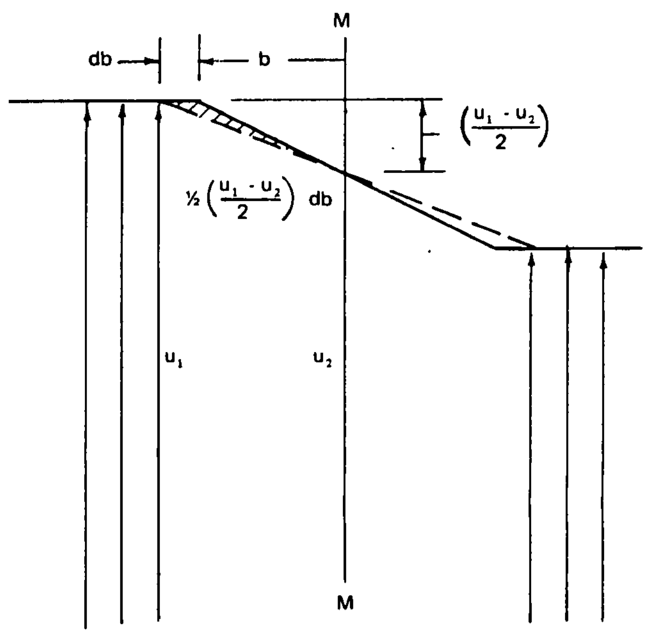
$$\frac{1}{2}(A - b)\frac{du}{dx} = -\frac{1}{2}\left(\frac{u_1 - u_2}{2}\right)\frac{db}{dx} \tag{C-3}$$

For an unconfined shear layer, the rate of spreading is constant, i.e.,

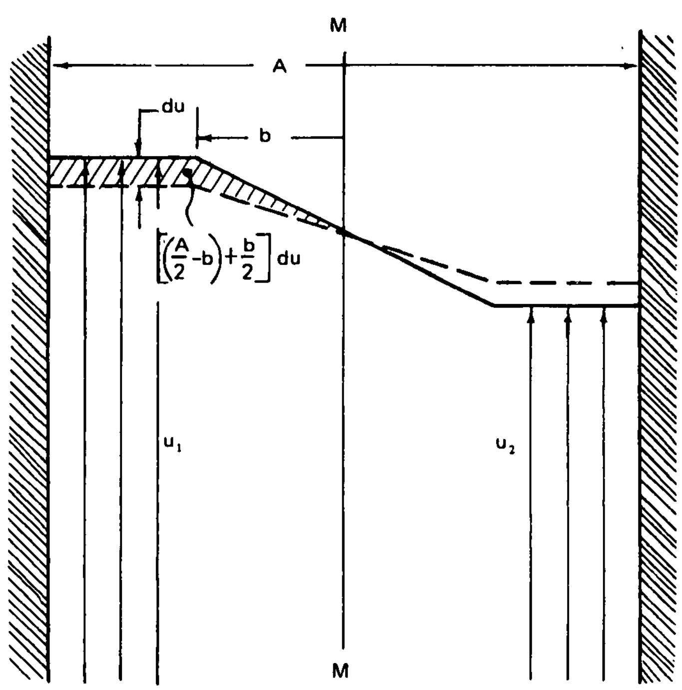
$$\frac{db}{dx} = \xi_o = \text{constant} \tag{C-4}$$

where ξ_o is the spreading angle for an unconfined shear layer. For a linearized case, we can use the approximation

$$\frac{d(u_1 - u_2)}{dx} \cong 2\frac{du}{dx} \tag{C-5}$$



(a) Unconfined



(b) Confined

FIGURE C–1 DIAGRAM ILLUSTRATING MOMENTUM EXCHANGE IN A SHEAR LAYER

Combining (C-3, 4, and 5), we get

$$(A - b) \frac{d(u_1 - u_2)}{dx} = \xi_o (u_1 - u_2)$$

or

$$\frac{1}{(u_1 - u_2)} \cdot \frac{d(u_1 - u_2)}{dx} = - \frac{\xi_o}{A(1 - \frac{b}{A})} \qquad \text{(C-6)}$$

If we assume that a typical Gaussian velocity profile can be represented by the simple profile of Figure C-1(b), with

$$\frac{b}{A} = \frac{1}{3}$$

then

$$\frac{1}{(u_1 - u_2)} \cdot \frac{d(u_1 - u_2)}{dx} = - \frac{3\xi_o}{2A} \qquad \text{(C-7)}$$

APPENDIX D

EXPERIMENTAL APPARATUS FOR JET-IN-CROSSFLOW STUDIES

APPENDIX D

EXPERIMENTAL APPARATUS FOR JET-IN-CROSSFLOW STUDIES

Experiments to provide experimental data on the behavior of a gas jet in a crossflow of lower density gas (as for a cold overfire air jet discharging into a heated furnace atmosphere) were conducted with the apparatus shown schematically in Figures D-1 and D-2. Figure D-1 shows the low velocity wind tunnel, in which uniform upflow of air is produced in a vertical test section. Velocities in the test section ranging from about 0.5 to 2 FPS could be attained. A jet of nitrogen gas at -320°F with a density about 3.7 times that of the air was injected horizontally into the test section, as shown in Figure D-1. The cold nitrogen mixing with the moist room air causes condensation of water vapor and produces visible flow patterns.

The nitrogen jet was produced with the arrangement shown in Figure D-2. By dissipating a measured electrical heat input in the liquid nitrogen, a known volume flow rate of saturated nitrogen vapor can be generated. Hence, the jet velocity can be readily determined from the relation

$$\bar{u}_o = \frac{4q_H}{\rho_g h_{fg} \pi d_o^2}$$

where:

$\bar{u}_o$ = jet velocity (feet/sec)

q_H = heater input power (Btu/sec)

ρ_g = density of saturated nitrogen vapor (lb_m/ft^3)

h_{fg} = latent heat of vaporization (Btu/lb_m)

d_o = diameter of nozzle (feet)

The heat input to the liquid nitrogen, and hence the velocity of the jet, is controlled by the voltage input to the heater and is easily varied by means of the Variac. The arrangement shown in Figure D-2 provides excellent thermal isolation of the liquid so that boil-off due to ambient heat leak is very small compared to that produced by the heater, for jet velocities of interest. The insulated, short flow-length passage from the vapor space in the dewar to the jet nozzle exit insures minimal heat transfer to the flowing cold vapor. Temperature measurements show that it issues from the nozzle at very close to the boiling point temperature.

Air upflow velocities in the test section are measured with a Hastings-Raydirt hot-wire anemometer whose calibration has been checked against flow rates determined by pilot tube traverses across the diffuser duct at a plane near its inlet.

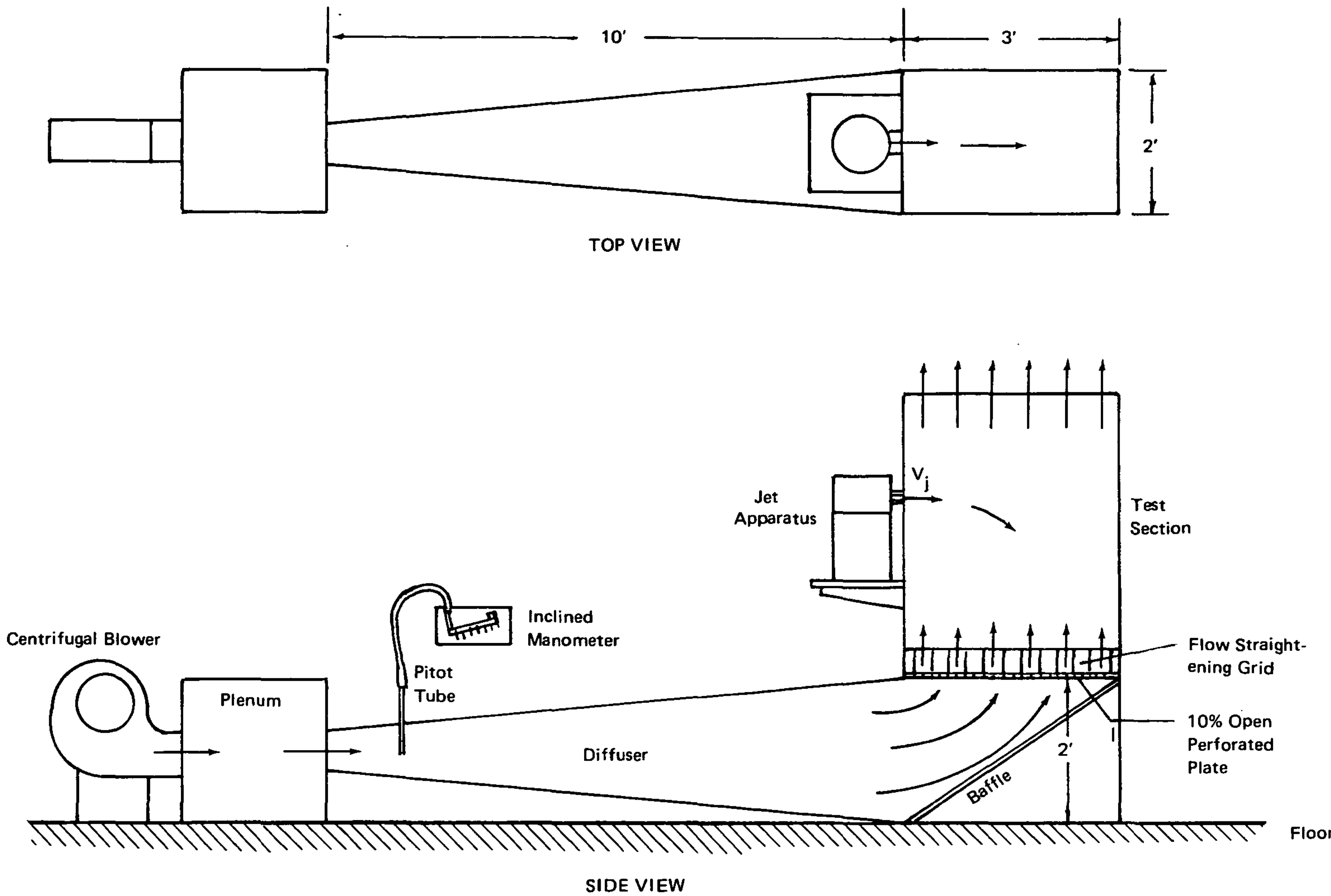


FIGURE D–1 LOW VELOCITY WIND TUNNEL

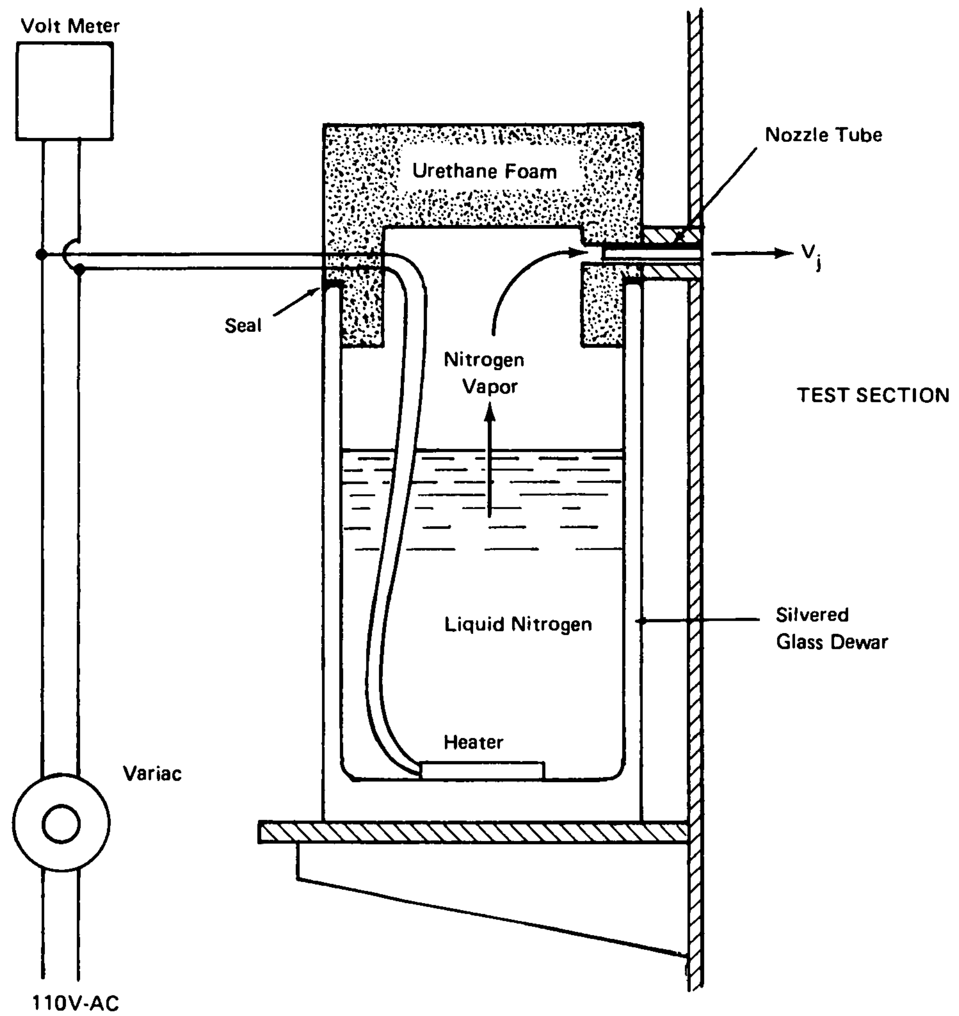


FIGURE D–2 APPARATUS TO PRODUCE COLD NITROGEN-VAPOR JET

The front and side panels of the test section are Plexiglass to permit photographing flow patterns of the jet in crossflow (such as shown in Section D of Chapter IV) indicated by the condensed water vapor fog.

A photograph of the assembled test apparatus is shown in Figure D-3.

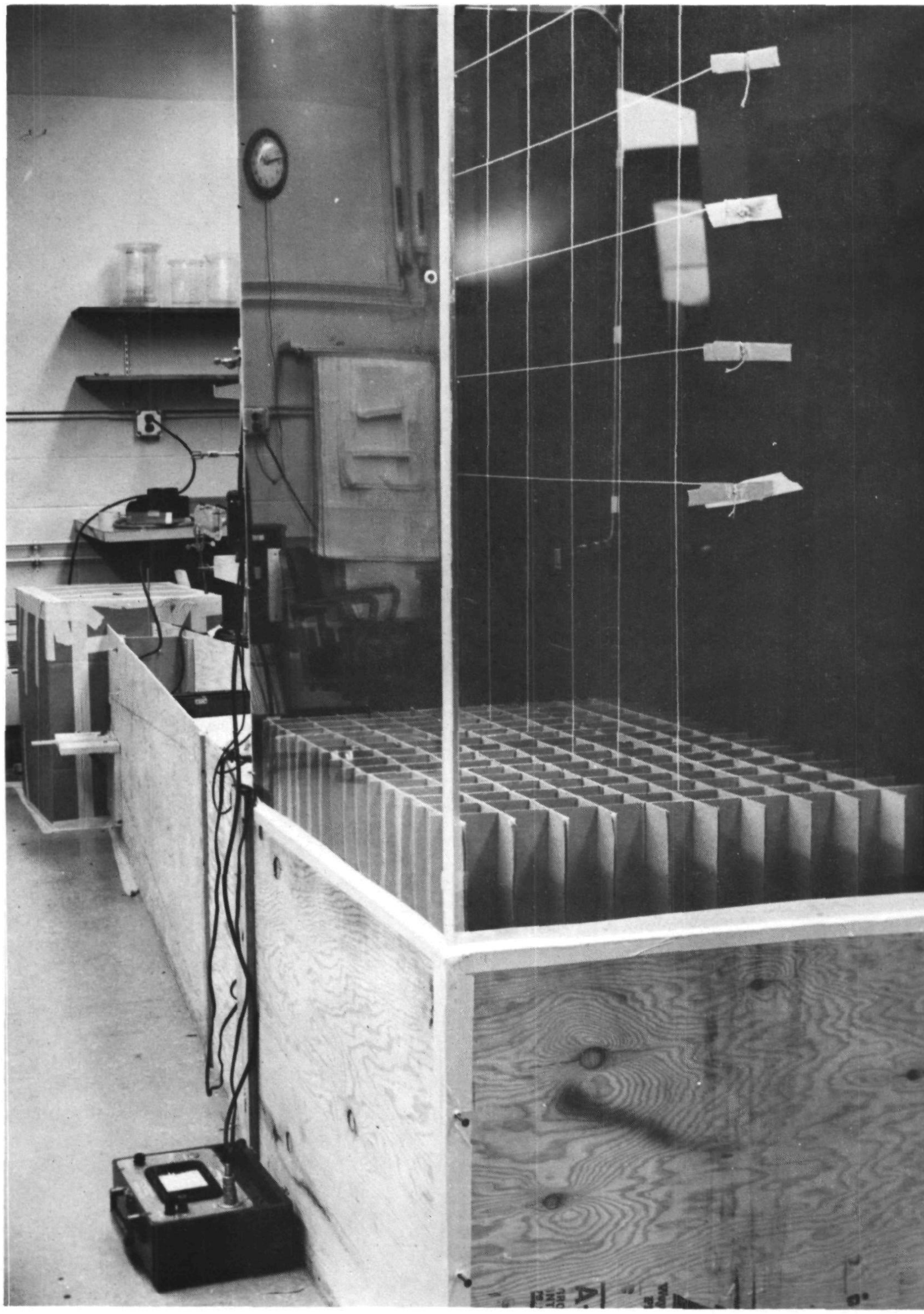


FIGURE D-3

PHOTOGRAPH OF TEST APPARATUS

APPENDIX E

DERIVATION OF COMBINED EFFECT MODEL

APPENDIX E

DERIVATION OF COMBINED EFFECT MODEL

The model is based on conservation of momentum in the coordinate system shown on Figure E-1.

In the axial (y) direction, conservation of momentum requires that:

$$\rho_j \; u_j^2 \; S_n \; \sin\beta \; \bar{u}_\circ^2 \; S_{no} \; \sin\beta_\circ \qquad \text{(E-1)}$$

Rearrangement of Equation (E-1) gives:

$$\frac{dy}{dt} = u_j \; \sin\beta = \frac{\rho_\circ \; \bar{u}_\circ^2 \; S_{no} \; \sin\beta_\circ}{\rho_j \; u_j \; S_n} \qquad \text{(E-2)}$$

In the cross flow (x) direction, conservation ot momentum requires that change in momentum flux be equal to the drag and buoyant forecasting on the volume element dy in thickness.

$$d \; \dot{M}_x = d \; F_{xD} + d \; F_{xB} \qquad \text{(E-3)}$$

where:

$$\dot{M}_x = \rho_j \; u_j^2 \; S_n \; \cos\beta \qquad \text{(E-4)}$$

$$d \; F_{xD} = \frac{1}{2} \; C_x \; \rho_a \; u_1^2 \; h \; dy \qquad \text{(E-5)}$$

(C_x is the normal drag coefficient)

$$d \; F_{xB} = g(\rho_a - \rho_j)\left(\frac{S_n}{\sin\beta}\right) dy \qquad \text{(E-6)}$$

Substitution of Equations (E-4), (E-5) and (E-6) into Equation (E-3) and integration from y = 0 to y = y yields:

$$\rho_j \; u_j^2 \; S_n \; \cos\beta - \rho_\circ \; u_\circ^2 \; S_{no} \; \cos\beta_\circ = \frac{1}{2} \; C_x \; \rho_a \; u_1^2 \int_0^y hdy + g \int_0^y (\rho_a - \rho_j) \; \frac{S_n}{\sin\beta} \; dy \qquad \text{(E-7)}$$

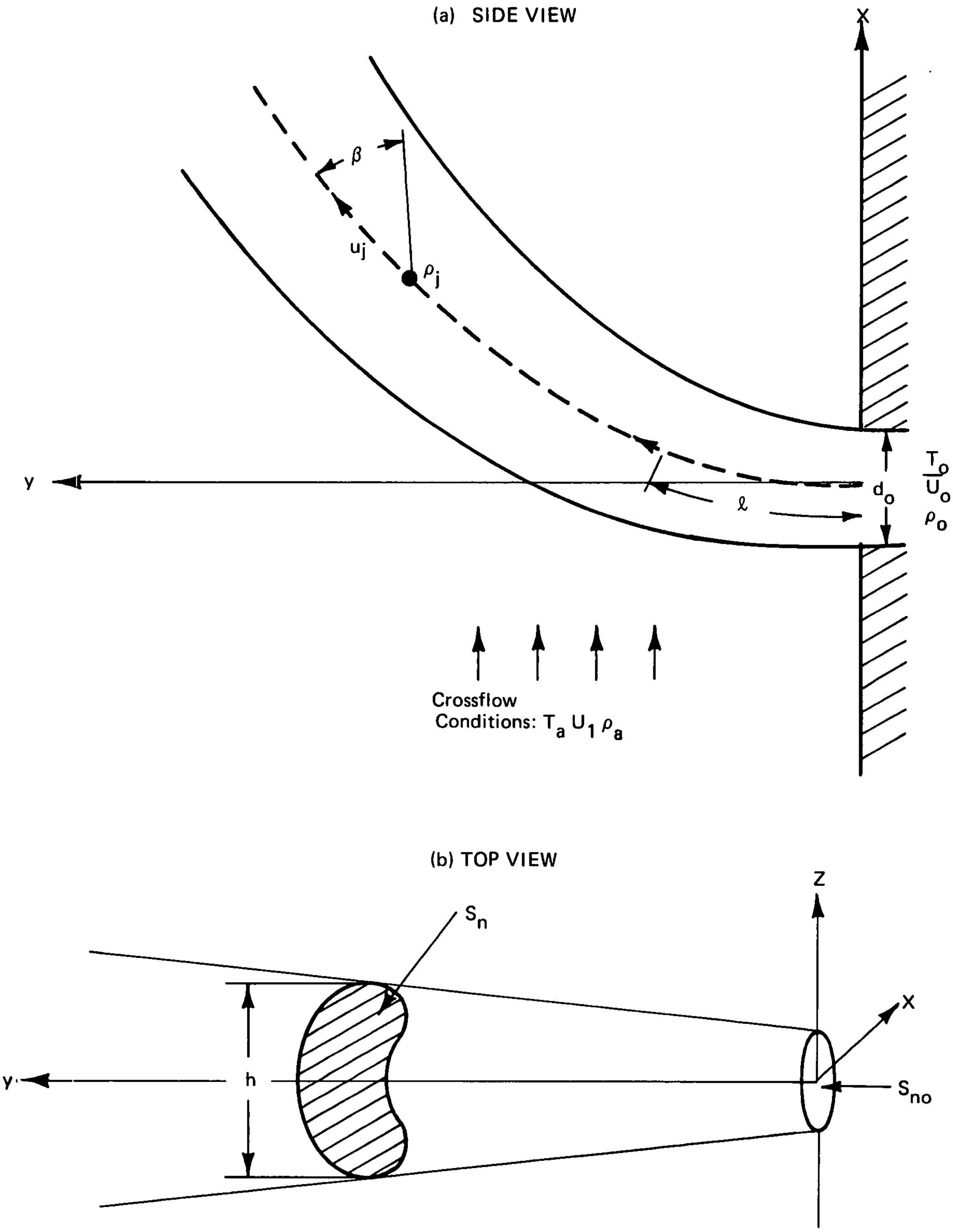


FIGURE E–1 TRAJECTORY OF DEFLECTED JET

Rearrangement yields:

$$\frac{dx}{dt} = u_j \cos\beta = \left(\frac{1}{\rho_j\ u_j\ S_n}\right)\left[\frac{1}{2}\, C_x\, \rho_a\, u_1^2 \int_o^y h dy + g \int_o^y \rho_a - \rho_j)\ \frac{S_n}{\sin\beta}\, dy \right.$$

$$\left. + \rho_\circ\ \bar{u}_\circ\ S_{no} \cos\beta_\circ \right] \qquad \text{(E-8)}$$

Division of Equation (E-8) by Equation (E-2) yields the slope of the jet centerline.

$$\frac{dx}{dy} = \frac{\frac{1}{2}\, C_x\, \rho_a\, u_1^2 \int_o^y h dy + g \int_o^y (\rho_a - \rho_j)\ \frac{S_n}{\sin\beta}\, dy}{\rho_\circ\ \bar{u}_\circ^2\ S_{no} \sin\beta_\circ} + c_o + \beta_o \qquad \text{(E-9)}$$

In order to evaluate Equation (E-9) and allow integration to describe the jet trajectory, we must establish the functional relationships $h(y)$, $\rho_j(y)$ and $S_n(y)$.

For an undeflected jet, Abramovich **assumed** that:

$$h = 2.25\ d_o + a_2\ \lambda \qquad \text{(E-10)}$$

where λ is the arc length along the jet path and the constant a_2 has a value of about 0.22. Deflected jets spread more rapidly and there is now indication that for these jets a_2 may be as high as 0.32.

The relationship between λ and y is unknown at this point. For simplicity, we assume that:

$$\lambda \cong a_1 y \qquad \text{(E-11)}$$

where a_1 is taken to be a constant of the order of $1/\sin\beta$. Substituting Equations (E-11) into Equation (E-10) yields (taking $a_2 = 0.22$):

$$h = 2.25\ d_o + 0.22\ a_1 y \qquad \text{(E-12)}$$

so that the first integral in Equation (E-9) becomes:

$$\int_o^y h dy = \int_o^y (2.25\ d_o + 0.22\ a_1 y)\ dy$$

$$= 2.25\ d_o\ y + 0.11\ a_1 y^2 \qquad \text{(E-13)}$$

The second integral in Equation (E-9) can be treated as follows, using Equation (E-1):

$$\int_0^y (\rho_a - \rho_j) \frac{S_n}{\sin\beta} dy = \rho_\sigma \bar{u}_\circ^2 S_{no} \sin\beta \int_0^y \frac{(\rho_a - \rho_j) dy}{\rho_j u_j^2 \sin^2\beta} \quad \text{(E-14)}$$

We make use of the following relationships which hold for undeflected jets:

$$\frac{T_j - T_1}{T_\circ - T_1} \sim \frac{u_j}{\bar{u}_\circ} \quad \text{(E-15)}$$

and:

$$\frac{u_j}{\bar{u}_\circ} = 3.2 \frac{d_o}{\lambda} = 3.2 \frac{d_\circ}{a_1 y} \quad \text{(E-16)}$$

together with the physical relationship:

$$\frac{\rho_a}{\rho_j} = \frac{T_j}{T_a} \quad \text{(E-17)}$$

Substitution of these relationships into Equation (E-14) yields

$$\int_0^y (\rho_a - \rho_j) \frac{S_n}{\sin\beta} dy = \frac{\rho_\circ S_{no} \sin\beta_\circ (T_a - T_1)}{3.2\, d_\circ T_1} \int_0^y \frac{a_1 y}{\sin^2\beta} dy \quad \text{(E-18)}$$

Making use of the fact that $a_1 \sim 1/\sin\beta$, the integral in Equation (E-18) can be approximated by:

$$\int_0^y \frac{a_1 y}{\sin^2\beta} dy \cong \int_0^y a_1^3 \, y dy = \frac{a_1^2 y^2}{2} \quad \text{(E-19)}$$

Substitution of Equations (E-18), (E-19) and (E-13) into Equation (E-9) yields:

$$\frac{dx}{dy} = \frac{C_x \, \rho_a \, u_1^2 \, (2.25 \, d_o \, y + 0.11 \, a y^2)}{2 \, \rho_\circ \, \bar{u}_\circ^2 \, S_{no} \, \sin\beta_\circ} + \frac{g \, (T_a - T_1) \, a_1^3 \, y^2}{6.4 \, \bar{u}_\circ^2 \, d_\circ \, T_1} \qquad \text{(E-20)}$$

Integration of Equation (E-20), recognizing that y = 0 at x = 0 yields:

$$\left(\frac{x}{d_o}\right) = \left(\frac{2.25 \, C_x \, \rho_a \, u_1^2 \, d_\circ^2}{4 \, \rho_a \, \bar{u}_\circ^2 \, S_{no} \, \sin\beta_\circ}\right) \left(\frac{y}{d_\circ}\right)^2 + \left[\frac{0.11 \, C_x \, \rho_a \, u_1^2 \, a_1 d_o^2}{6 \, \rho_a \, u_\circ^2 \, S_{no} \, \sin\beta_\circ} + \frac{g \, (T_\circ - T_1) \, a^3 \, d_\circ^2}{19.2 \, u_\circ^2 \, d_\circ \, T_1}\right] \left(\frac{y}{d_\circ}\right)^3 + \left(\frac{y}{d_o}\right) \cot\beta_\circ \qquad \text{(E-21)}$$

In order to evaluate Equation (E-21), numerical values for C_x and a must be selected. The effective drag coefficient C_x is believed to be in the range of 4 to 5, based on analysis of experimental crossflow data.

Since $a_1 \approx 1/\sin\beta$, an average value of $\sin\beta$ should be used. To describe the trajectory near the jet mouth, the value of $\sin\beta_\circ$ should be used.

APPENDIX F

RESULTS OF FLUE SAMPLING TESTS AT MUNICIPAL INCINERATOR
NEWTON, MASSACHUSETTS

APPENDIX F

RESULTS OF FLUE SAMPLING TESTS AT MUNICIPAL INCINERATOR NEWTON, MASSACHUSETTS

It has been suggested (Chapter II) that although a large fraction of the combustibles are emitted in the pyrolysis zone of the grate, some slow-burning refuse components continue to burn (or begin to burn) on the discharge grate. The high excess air levels often found over the discharge grate could lead to quenching of combustion reactions and consequent pollutant carry-over. To obtain an indication of the importance of this source of combustibles, a test was carried out at the municipal incinerator in Newton, Massachusetts.

Samples of flue gas and gas temperature measurements were taken using uncooled probes in the flue immediately downsteaam of the furnace (see Figure VI-1, Chapter VI). The results are reported below:

1. SUMMARY

a. CO level is greater near the wall, and at the bottom of the flue in the low-temperature region.

b. CO_2 level is greatest in the center of the duct, and at higher temperatures.

c. Temperature fluctuations of greater than 100°F occur in less than a 5-minute period.

2. DUCT AND PORT DIMENSIONS

The duct is 11.1 feet across by 12 feet high (outside) - 1 foot thick wall. Five metal sleeves 12 inches long x 5.5 inches in diameter had been previously inserted into the sidewall in a vertical row on 2.2 foot centers with the uppermost port 23 feet from floor level and 11 feet from duct bottom level.

3. SAMPLING

Samples were taken from the upper three ports only. The bottom two ports are below the fly ash level in the duct. Two sets of samples were taken: a) at 2 feet in from the duct sidewall and b) at 5.5 feet in from the duct sidewall. The samples were taken simultaneously from the upper three ports due to rapid temperature fluctuations of about 100°F over a 5-minute period. Temperatures were monitored by shielded chromel-alumel thermocouple located at the tip of the gas sampling probe.

4. ANALYSIS

Analysis was by gas chromatography. The CO content of several samples was also determined by color indicator tubes made by the Bacharach Industrial Instrument Company. Oxygen, nitrogen and carbon monoxide levels were determined using a Linde 5A molecular sieve column and thermal conductivity detection.

Carbon dioxide was determined using a Porapak Q (cross-linked polystyrene) column and thermal conductivity detector. Instrument response to each gas was determined by external calibration.

5. DATA

		ppm CO[b]		%				
		GC	Bacharach	O_2	N_2	CO_2	Sum[c]	Temp
(d)	*A1[a]	540		23.5	83.0	0.14	106.6	1160°F
	A2	250	400	-	-	-		1385°F
(e)	A3	55	80	16.8	80.0	3.9	100.7	1470°F
	A4	<50	<50	-	-	-		1355°F
(d)	B1	320		12.9	83.8	7.6	104.3	1360°F
	B2	55	100	-	-	-		1675°F
(e)	B3	<50		17.5	83.8	7.5	108.8	1600°F
	B4	-		-	-	-		1610°F
(d)	C1	<50	70	-	-	-		1470°F
	C2	<50		12.2	77.0	7.0	96.2	1680°F
(e)	C3	<50		10.3	79.9	9.1	99.3	1635°F
	C4	-		-	-	-		1655°F

(a) Two sets were taken at each point. All samples were not analyzed. Samples identified as 3&4 were taken in center of duct.

(b) Detection limit = 50 ppm. Note consistent difference between Bacharach and GC values.

(c) Total of $O_2 + N_2 + CO_2$, Theoretical = 100%.

(d) Duplicates of samples ∿ 1 foot inside wall.

(e) Duplicates of samples about in middle of duct.

*A = Lowest port; C = Highest port.

6. DISCUSSION

The data show the presence of carbon monoxide (CO) in the low-temperature gases. Based on reasoning presented in Chapter III, this suggests their generation in the discharge grate region, perhaps by quenching of the combustion reactions.

The absence of CO in the hot gases is not conclusive proof that they did not exist there (in fuel-rich eddies). It would be expected that at the temperatures noted in the upper regions of the duct (about 1600°F) the mixing obtained in the uncooled sampling probe would result in rapid oxidation of the CO **to** CO_2. For the test program, it should be noted, water-cooled sampling probes will be used.