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AUTHOR: C. A. Kodres

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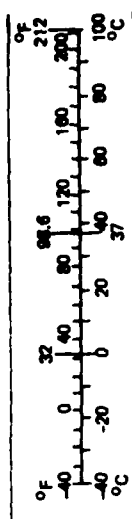Approved for public release: distribution unlimited.

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METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in²	square inches	6.5	square centimeters	cm²
ft²	square feet	0.09	square meters	m²
yd²	square yards	0.8	square meters	m²
mi²	square miles	2.6	square kilometers	km²
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2,000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft³	cubic feet	0.03	cubic meters	m³
yd³	cubic yards	0.76	cubic meters	m³
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm²	square centimeters	0.16	square inches	in²
m²	square meters	1.2	square yards	yd²
km²	square kilometers	0.4	square miles	mi²
ha	hectares (10,000 m²)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1,000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m³	cubic meters	35	cubic feet	ft³
m³	cubic meters	1.3	cubic yards	yd³
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F

°F −40   0   32   80   120   160   212
   −40 −20   0   20   37   40   60   80   100
°C

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A mathematical model is developed to simulate the operation of the heat recovery incinerators at NAS Jacksonville. The model is used to conduct a parametric examination of the facility. Airflows, including leakages, are the dominant parameters affecting operation of the HRI. Because of poor airflow control, and partly because of air leakage, the Jacksonville HRIs rarely operate in the starved air mode they were designed for.

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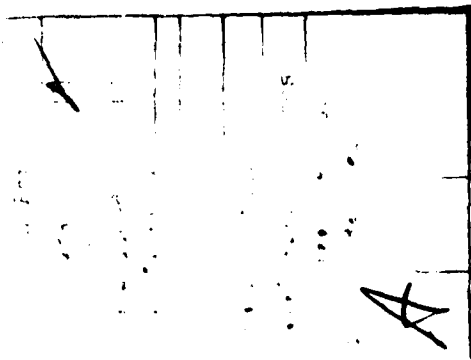1. Solid waste   2. Combustion   1. YO817-006-01-002

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CONTENTS

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INTRODUCTION

The solid waste facility at the Naval Air Station in Jacksonville, Florida, (NAS JAX) is one of two prototypes installed by the Navy in the late 1970s. These facilities have dual objectives: to incinerate the solid waste generated at the base, recovering the heat of combustion in the form of steam (thus conserving fossil fuels) and, simultaneously, to reduce landfill disposal loads.

Over the last 2 years, the performance of the Jacksonville incinerators has been deteriorating. Many of the difficulties experienced at the facility are purely mechanical and obvious. Several problems, however, are associated with the actual physics of incineration and are more subtle. The erratic production of steam is an example.

An experimental parametric examination of the heat recovery incinerators (HRI) is planned. These tests are for troubleshooting purposes but, of perhaps more importance, they will also contribute to an overall understanding of the operation of this type of incinerator.

To prepare for these tests, an analytical parametric study was conducted. The purpose of this preliminary analysis was to determine the significant parameters affecting the operation and performance of the HRI; these parameters will be included in the experimental program. Steam generation was the dependent variable. Other measures of performance (e.g., environmental) were not considered. Results of the study are presented in this report.

DESCRIPTION OF INCINERATORS

The solid waste facility consists of three outdoor incinerators for burning the waste; each is equipped with a boiler for recovering the heat. Each incinerator has a design capacity to burn 2,000 pounds of waste per hour and, concurrently, 7.5 gallons of oil per hour as an auxiliary fuel.

Configuration

Figure 1 is a schematic of the NAS JAX heat recovery incinerators. The primary and secondary combustion chambers are refractory lined. During operation, the ram loader forces the waste into the interior of the primary combustion chamber (PCC). Two internal rams push the burning waste toward an opening in the chamber floor where the ash is discharged into a water-sealed quench tank.

Combustion air is supplied to the incinerators by a forced draft blower at the rate of about 460 lb/min. Total combustion air remains constant (i.e., the total of the underfire air plus air supplied to the secondary combustion chamber (SCC) is always equal to 460 lb/min). There is a small quantity of air injected with the oil.

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Ambient air leaks into the primary combustion chamber overfire, through the fire door and through holes in the refractory. Probable leakage occurs down the dump stack, through the damper.

The boilers are the water tube type. Combustion gases leaving the SCC make a single pass around the 264 staggered water tubes.

The significant dimensions of the HRI are listed in Appendix A.

Operation

The incinerators are designed to operate with insufficient underfire air supplied to the PCC for the complete combustion of the waste. Flame temperatures thus remain low, preventing slagging and, possibly, the jamming of the internal rams. Lowering airflow rates through the burning waste also decreases the entrainment of solid particles, a potential pollution problem. The major portion of the combustion air is diverted directly to the SCC.

Figures 2 and 3 summarize the operation of the Jacksonville incinerators (the origin of these figures will be explained later). Figure 2 shows temperature variations throughout the HRI as a function of combustion air distribution. To the left side of the peak of these curves is the designed operating point; the curves peak with approximately theoretical (stoichiometric) air.

Total combustion air supplied to the incinerators is set. If the waste feed rate is constant, it follows that the temperature of the combustion products as they leave the secondary combustion chamber to enter the boiler is nearly a constant, and overall thermal efficiency of the HRI varies little with departure from the design point. Figure 3 illustrates the effect of combustion air distribution on the efficiency of the facility. The only factor to vary noticeably is the heat transfer loss through the walls of the PCC, and this amounts to just a few percent.

PYROLYSIS OF WASTE

Most types* of waste (e.g., paper, wood, and plastics) are distilled when subjected to heat (Ref 1). This distillation process, called pyrolysis, is an irreversible degradation of the solid to form various volatile gases and a carbonaceous solid residue (Ref 2).

The volatile matter is emitted volumetrically from the interior of the solid and represents the primary combustibles. With most types of waste, the volatiles comprise about 80% of the total mass. Once the volatiles are released, the carbon residue is itself combustible.

The heat of pyrolysis may be endothermic or exothermic depending upon type of solid and local temperature (Ref 3 and 4). For most types of waste it is endothermic.

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*Most of the total mass of a mixed waste as well; examine the composite waste of Appendix B.

MATHEMATICAL SIMULATION OF HRI

A mathematical model of the NAS JAX HRI has been developed based on the hypothesized combustion reaction,
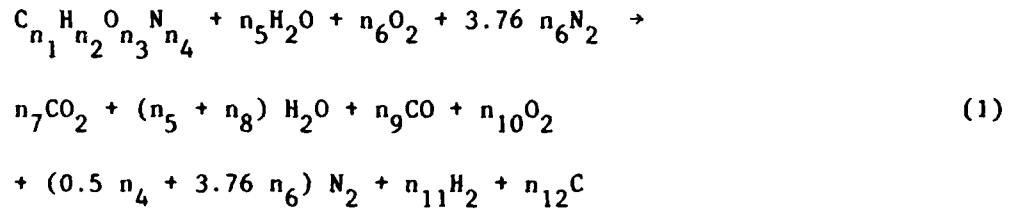
$$C_{n_1} H_{n_2} O_{n_3} N_{n_4} + n_5 H_2O + n_6 O_2 + 3.76 \, n_6 N_2 \rightarrow$$

$$n_7 CO_2 + (n_5 + n_8) H_2O + n_9 CO + n_{10} O_2 \tag{1}$$

$$+ (0.5 \, n_4 + 3.76 \, n_6) N_2 + n_{11} H_2 + n_{12} C$$

In addition it is assumed that:

1. Steady state exists.

2. Kinetic and potential energy changes are negligible.

3. The reactions go to completion regardless of the temperature.

4. Combustion is diffusion controlled, limited only by the mass flow rates of fuel and oxygen.

5. The products of combustion are perfectly mixed.

6. Therefore, all temperature gradients are normal to the incinerator walls; the individual components of the incinerator may be represented one dimensionally.

The molar coefficients n_1 through n_5 are obtained from ultimate and proximate analysis of the waste (fuel); n_6 is from the air supplied for combustion. Equation 1 is balanced by applying conservation of species and allocating available oxygen in order of increasing activation energy for combustion in air: first to hydrogen to form water vapor, then to carbon to form carbon monoxide. Any oxygen remaining is assumed to oxidize the carbon monoxide to form carbon dioxide.

Heat absorbed in breaking down the waste, primarily a heat of pyrolysis, is determined by balancing Equation 1 for stoichiometric air, then subtracting the heats of formation of the combustion products from the heating value of the waste. Once the heat of pyrolysis is known, heat released during combustion with less than stoichiometric air is back-calculated in an analogous manner.

By applying conservation of energy to the flame, primary combustion chamber, secondary combustion chamber, and boiler, in sequence, temperatures throughout the HRI and, finally, steam generation are determined. Figure 2 was derived in this way.

The details of the mathematical simulation, including an estimate of accuracy, are given in Appendix C.

PERFORMANCE CRITERIA

The efficiencies of both the boiler alone and the overall heat recovery incinerator are defined using the heat loss method (Ref 5),

$$\eta \equiv 1 - \frac{\Sigma \text{ LOSSES}}{\Sigma \text{ INPUT}} \tag{2}$$

For the boiler:

Σ LOSSES = sensible heat in stack gases + steam lost to blowdown

Σ INPUT = sensible heat in products of combustion entering boiler + sensible heat of feed water

For the overall HRI:

Σ LOSSES = heat lost vaporizing moisture with waste + heat lost vaporizing moisture generated by burning hydrogen in waste + carbon carried out as ash + sensible heat of ash + heat transfer through walls of PCC and SCC + carbon monoxide in stack gases + sensible heat in stack gases + steam lost to blowdown

Σ INPUT = chemical energy in waste and oil + sensible energy in waste, oil, air, and feed water + external power requirements

With a parametric study, this definition of efficiency is preferable because it isolates individual components of the HRI, simplifying identification of significant parameters. For example, Figure 3 would suggest placing an emphasis on the operation of the boiler at, perhaps, the expense of the ash and ash recovery system.

Individual terms in the summations are mathematically described in Appendix D.

PARAMETRIC EXAMINATION

In the evaluations of heat recovery incinerator parameters that follow, PCC underfire air will usually be used as the independent variable.

The parameter values used as a baseline and fuel (waste) compositions are summarized in Appendices A and B, respectively. With baseline values, when burning 1,500 lb/hr of "composite" waste, theoretical air is about 136 lb/min.

The boilers have never been flow tested to determine actual heat transfer characteristics. Boiler performance anticipated by the manufacturer, as summarized on the nameplates, is being used in these analyses.

Combustion Air

Three different NAS JAX heat recovery incinerator operating modes are possible with, at most, only minor modifications to the existing configuration:

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1. Set the combustion airflow to the PCC and SCC combined at 460 lb/min. This is the design mode discussed earlier and shown on Figures 2 and 3. The primary combustion chamber is kept cool by operating with insufficient underfire (U/F) air (U/F air plus leakage < 136 lb/min on Figure 2) or, alternatively, by operating with much excess underfire air in order to dilute the combustion products.

2. Operate with insufficient underfire air, but limit the air to the secondary combustion chamber to maximize the temperature of the combustion products entering the boiler. This operating mode is summarized by Figure 4. It will be referred to as the "optimized starved-air" mode.

3. Operate with excess underfire air, sufficient for PCC cooling, but cut off all airflow to the secondary combustion chamber to eliminate further cooling of the combustion products. This mode is illustrated by Figure 5.

Thermal efficiency of the HRI operating in an optimized starved-air mode peaks when the total combustion airflow reaches its stoichiometric value, then decreases slowly as the SCC air is increased. Although secondary combustion chamber temperature decreases rapidly as SCC air is increased, boiler performance falls off slowly, the decrease attenuated by a corresponding increase in the boiler overall heat transfer coefficient as the flow over the tubes increases.

A limitation of HRI operation in the optimized starved-air mode is the tolerable boiler inlet temperature. Several boilers capable of handling combustion gases as hot as 2,800°F are commercially available, but for the majority of heat recovery boilers, including NAS JAX, an inlet temperature of about 2,000°F is considered maximum. Some dilution of SCC combusion products with outside air is necessary.

Thermal efficiency of the HRI in a mode where all air to the SCC is cut off peaks when PCC airflow reaches its stoichiometric value, then falls off slowly as this airflow is increased. Compare Figures 4 and 5. To keep the PCC temperature at an acceptable level, however, the incinerator would have to operate with much excess underfire air, far to the right in Figure 5.

Table 1 is an attempt to compare the different combustion air modes on a one-to-one basis. Column 1 summarizes the NAS JAX HRI as it is probably operating: a PCC temperature of 1,800°F with excess air to the PCC.* Column 2 shows the effects of an easy "fix"; simply cutting off all air to the SCC results in a 6% increase in steam generation.

The potentials of the different operating modes are perhaps better illustrated if design limitations on the Jacksonville HRIs are momentarily ignored. Columns 3 and 5 of Table 1 assume that the PCC temperature is permitted to reach 2,400°F, slightly below the melting temperature of glass. These two columns summarize operation on the two sides of the stoichiometric peak of Figure 2. This is about the minimum temperature predicted by the model with insufficient air and permits a direct comparison of all three modes (i.e., with columns 4 and 6). Cutting off

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*PCC temperatures are currently averaging about 1,800°F, while SCC temperatures are averaging about 1,500°F.

all air to the SCC increases steam generation by about 14%. Operation in the optimized starved-air mode increases steam generation by about 17%; enough SCC air is being supplied to limit the boiler inlet temperature to 2,800°F.

Initial Temperature of Combustion Air

Preheating the combustion air is a common technique for improving the performance of coal-fired boilers. For excess air operation, the net result is an increase in the temperature of the combustion products. When operating in a starved-air mode with limitations on boiler inlet temperature, the net result is an increase in the mass flow rate of the combustion gases passing through the boiler.

Figures 6 and 7 show the effect of preheating the combustion ai . the performance of the NAS JAX heat recovery incinerators operating designed. If stack gases are used for this preheating, the efficien of the HRI might be increased by 4% or 5%, depending upon the select of the heat exchanger. Stack gas temperatures at Jacksonville range from 500°F to 600°F. The concept of using this "wasted" heat to inc the initial temperature of the combustion air to 200°F, even 250°F, . not unreasonable.

Air Leakage

Leakage is not a convenient experimental parameter, but it exists, nevertheless, and must be considered. Two air leakages will be examined: overfire leakage into the PCC, all sources lumped together, and leakage down the dump stack.

Figure 8 illustrates the effect of overfire leakage on temperatures in the primary combustion chamber – the stoichiometric peak is moved to the left. The characteristics of this figure support the supposition that the NAS JAX HRI normally operates with excess underfire air while burning most types of waste.

Because of a high oxygen content, stoichiometric air required to burn waste is quite low (compared with, say, coal; Figure 9). Thus, when operating with insufficient air, any leakage into the PCC will tend to increase the temperature of the combustion products very rapidly. The Jacksonville incinerators achieve desired PCC temperatures by increasing underfire air. Once PCC temperature settings are exceeded because of leakage (or any other reason), underfire air is increased, moving the operating point to the right (of Figure 2 or 8) and over the stoichiometric peak until the desired PCC temperature is reached. The incinerator is now operating with considerable excess air.

Heat recovery incinerators are not as vulnerable to leakage down the dump stack. The effect of this leakage is summarized by Figure 10. Dump stack leakage dilutes the combustion products with cooler ambient air prior to entering the boiler, which decreases the temperature difference between these gases and the steam being generated. Concurrently, the boiler overall heat transfer coefficient increases as the gas flow over the tubes increases, attenuating the decrease in the performance of the boiler. These same countering trends have been introduced previously in reference to the effects of changing SCC airflows.

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Type of Waste

Effects of waste type on primary combustion chamber temperatures and on steam generation are illustrated on Figures 11 and 12, respectively. The curves can only be considered as typical. Each waste type represents just a single sample. In addition, waste characteristics can be expected to vary with the time of year (e.g., the moisture content and the composition of the "composite").

Most types of waste require even less combustion air than the composite used as a baseline. Plastics, however, require more than twice the air needed by the composite; airflows optimum for paper would probably be inadequate to even sustain the combustion of most plastics. Thus, the operating mode and the type of waste cannot be considered independently.

Waste Feed Rate

Effects of waste feed rate on PCC temperatures and on steam generation are illustrated on Figures 13 and 14. These figures show the obvious: more waste produces higher temperatures and more steam.

It is noteworthy to observe that the NAS JAX heat recovery incinerator apparently cannot burn 2,000 lb/hr of waste and still maintain a PCC temperature of 1,800°F, as rated, when operating in an excess air mode. Examine Figure 13 and recall that the total capacity of the combustion forced-air blower is 460 lb/min.

Moisture With the Waste

The major loss attributable to moisture with the waste is not the heat lost vaporizing the water but the decrease in the amount of fuel burned. The heat of vaporization of water, about 970 Btu/lb, although significant, is small compared with the heating value of the dry fuel, typically 8,500 Btu/lb. In Figure 15, a 30% moisture content would be expected to decrease HRI efficiency by about 5%; yet steam generation falls off by nearly 35%, the heat lost vaporizing the moisture plus the decrease in the dry weight of the waste burned.

Another consideration is the effect of moisture on flame temperature. A 30% moisture content in fuel burned at the Jacksonville facility would decrease the flame temperature to less than 1,400°F; this may not be hot enough to sustain combustion.

Boiler Performance

As shown in Figure 3, boiler losses, illustrated here in terms of the sensible heat remaining in the stack gases, are, by far, the major inefficiencies affecting the overall performance of the heat recovery incinerator*. In Figure 16, the boiler is "switched" by changing the

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*The importance of boiler efficiency emphasizes a need to examine and compare various heat exchanger configurations as part of the preliminary design of these solid waste facilities (e.g., the potential of waterwalls and of preheaters or economizers).

7

heat transfer characteristics. The range of coefficients examined is representative of the range available among commercial boilers. Note the order of magnitude difference in the steam generated.

Incinerator Heat Transfer Characteristics

The resistance to conduction through the walls is the dominant resistance to heat transfer out of the incinerator. This is illustrated by Figure 17, a plot of HRI wall temperatures. Compare temperature gradients to the walls and through the walls.

Convection film coefficients at the inner and outer wall surfaces were estimated at 50 Btu/hr-ft^2°F (turbulent forced convection) and 5 Btu/hr-ft^2°F (free convection), respectively. These values are representative. Regardless, the conductance through the walls is only 0.75 Btu/hr-ft^2°F, and any error made in modelling convection heat transfer will be trivial.

For the same reason, incinerator heat transfer is not a significant parameter. Any heat transfer variables that could be experimentally examined would have little effect on the overall efficiency of the HRI.

Figure 18 isolates HRI heat transfer losses in terms of fundamental sources. Near the stoichiometric peak, the major heat transfer loss is attributable to radiation from the flame. At the probable operating points, convection and radiation losses are of approximately equal magnitude.

Ash Composition

Most types of waste have a very high volatiles content, typically around 80% (compared with about 30% for coal). Thus, the combustion of the char is a minor part of the waste combustion process. There is little solid left to burn after the products of pyrolysis are released.

Pyrolysis is a volumetric phenomenon. The surface-to-volume ratio of most types of waste is quite large (e.g., paper); therefore, the distillation rate of waste is limited only by the rate of the required heat transfer to the waste. It follows that the volatiles in the waste are normally released very rapidly. For these two reasons, the losses due to energy remaining in the ash, both chemical and sensible, are small. The losses shown in Figure 3, less than 2% total, are typical.

Augmenting Combustion by Burning Oil

Figure 19 is included to complete the parametric study. It would take much more oil than is currently being burned at Jacksonville to appreciably affect performance of the heat recovery incinerators.

SUMMARY

The dominant variable affecting the operation of the NAS JAX heat recovery incinerators is airflow. Underfire and secondary combustion air, as well as the air control mode, are, therefore, parameters to be examined.

The facility is particularly sensitive to airflow when operating with air insufficient for the complete combustion of the waste in the PCC. This sensitivity can be attributed to the high oxygen content of most types of waste. As a result, stoichiometric air requirements are very low. The difference between 3,500°F flames and extinguished flames is typically less than 150 lb/min of airflow.

This same sensitivity is pertinent if the source of the air is leakage. A leaky incinerator will have difficulty operating in a starved-air mode. The Jacksonville incinerators fall in this category. These HRI are currently operating with PCC temperatures about 300°F greater than corresponding SCC temperatures, implying excess air operation.

Type of waste, feed rate, and moisture content are parameters that may vary over a wide range during hour-to-hour operation of the heat recovery incinerators. It follows that combustion air requirements vary continuously and, possibly, radically. The incinerator air control system must be capable of correctly responding to these variations.*

Deficiencies of the NAS JAX HRI control system are another reason the incinerators operate with excess air a large portion of the time. The system responds to high PCC temperatures by increasing underfire air. Acceptable PCC temperatures are not again achieved until the HRI is operating with considerable excess air.

The dominant variable affecting the overall thermal efficiency of the Jacksonville HRIs is boiler efficiency; boiler losses are the major irreversibilities limiting the conversion of solid waste into steam. In an established facility such as Jacksonville, there are few boiler parameters that could be experimentally examined. The overall heat transfer coefficient is the exception. This parameter is automatically changed when feed rates, airflows, or secondary combustion chamber temperatures are varied.

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*Control of an incinerator designed to operate on both sides of the stoichiometric peak is doubly complex; several independent blowers are normally utilized. Another option would be to base the air controls on more than one variable (i.e., T_{PCC} and T_{SCC} or perhaps T_{PCC} and hydrocarbons in the combustion products).

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NOMENCLATURE

A	Surface area
BD	Boiler blowdown
C	Carbon content as determined by ultimate analysis
$C_p(T)$	Specific heat at temperature T
HHV	Higher heating value
h_{CONV}	Convection heat transfer film coefficient
$\Delta h(T)$	Enthalpy at temperature T relative to enthalpy at T_{DATUM}
Δh_{fg}	Heat of vaporization
K	Thermal conductance
k	Coefficient of thermal conductivity
$LMTD$	Logarithmic mean overall temperature difference
$\dot{M}$	Mass flow rate
n_1 thru n_{12}	Molar coefficients
Q	Heat of formation
$\hat{Q}_{LOST}$	Energy lost vaporizing moisture in waste
$\dot{q}$	Heat flux
T	Temperature
ΔT	Temperature relative to the datum temperature; $\Delta T \equiv T - T_{DATUM}$
U	Overall heat transfer coefficient of boiler
$\varepsilon(T)$	Emissivity at temperature T
η	Efficiency
μ	Viscosity
σ	Stefan-Boltzmann constant

SUBSCRIPTS

AIR	Refers to airflows, combustion or leakage air as applicable
ASH	Refers to ash leaving the incinerator
AVG	Average value
COND	By conduction heat transfer
CONV	By convection heat transfer
DATUM	Datum for defined properties such as enthalpy
DRY	Refers to fuel (waste) conditions with all moisture removed
FEED	Refers to feed water entering the boiler
FLAME	Refers to the flame, adiabatic or homogeneous as applicable
FUEL	Refers to waste fed into the incinerator
F→G	From the flame to the combustion products (gases)
F→W	From the flame to the incinerator walls
G→W	From the combustion gases to the incinerator walls
LEAK	Air leakage, into the PCC or down the dump stack as applicable
LOST	Lost vaporizing the moisture in the fuel
MIX	Refers to products of combustion in the PCC or SCC as applicable
MOIST	Refers to moisture with the fuel
OIL	Refers to oil concurrently burned with the waste
PCC	Primary combustion chamber
RAD	By radiation heat transfer
SCC	Secondary combustion chamber
SHELL	Refers to outer skin of incinerator walls
STACK	Refers to combustion products exiting the boiler
STEAM	Refers to steam generated by heat recovery boiler
STOICH	Stoichiometric condition
WALLS	Refers to inner skin of incinerator walls
WET	Refers to fuel (waste) conditions as burned, with all moisture present
W→∞	From the outer skin to the surrounding atmosphere
∞	Ambient condition

ABBREVIATIONS

HRI Heat recovery incinerator

PCC Primary combustion chamber

SCC Secondary combustion chamber

U/F Underfire combustion air

Table 1. The Effect of the Operating Mode on the Performance
of the NAS JAX Heat Recovery Incinerator

Variable	HRI Operating Mode					
	Current (1)	No Air To SCC (2)	Current (3)	No Air To SCC (4)	Current (5)	Optimized Starved Air (6)
Feed Rate, lb/hr	1,500	1,500	1,500	1,500	1,500	1,500
Waste Type	Composite	Composite	Composite	Composite	Composite	Composite
Air to PCC, lb/min	360	360	240	240	60	60
Air to SCC, lb/min	100	0	220	0	400	120
PCC Gas Temperature, $^{\circ}F$	1,815	1,815	2,424	2,424	2,448	2,448
SCC Gas Temperature, $^{\circ}F$	1,448	1,736	1,424	2,269	1,424	2,730
Total Gas Flow, lb/min	514	414	514	294	514	234
Boiler Heat Transfer Coefficient, $Btu/hr\text{-}ft^2\text{-}^{\circ}F$	12.94	11.57	12.92	9.69	12.92	8.63
Boiler Efficiency	0.70	0.73	0.70	0.80	0.70	0.82
HRI Overall Efficiency	0.61	0.65	0.60	0.67	0.60	0.69
Steam Generation, lb/hr	7,444	7,884	7,290	8,275	7,284	8,543
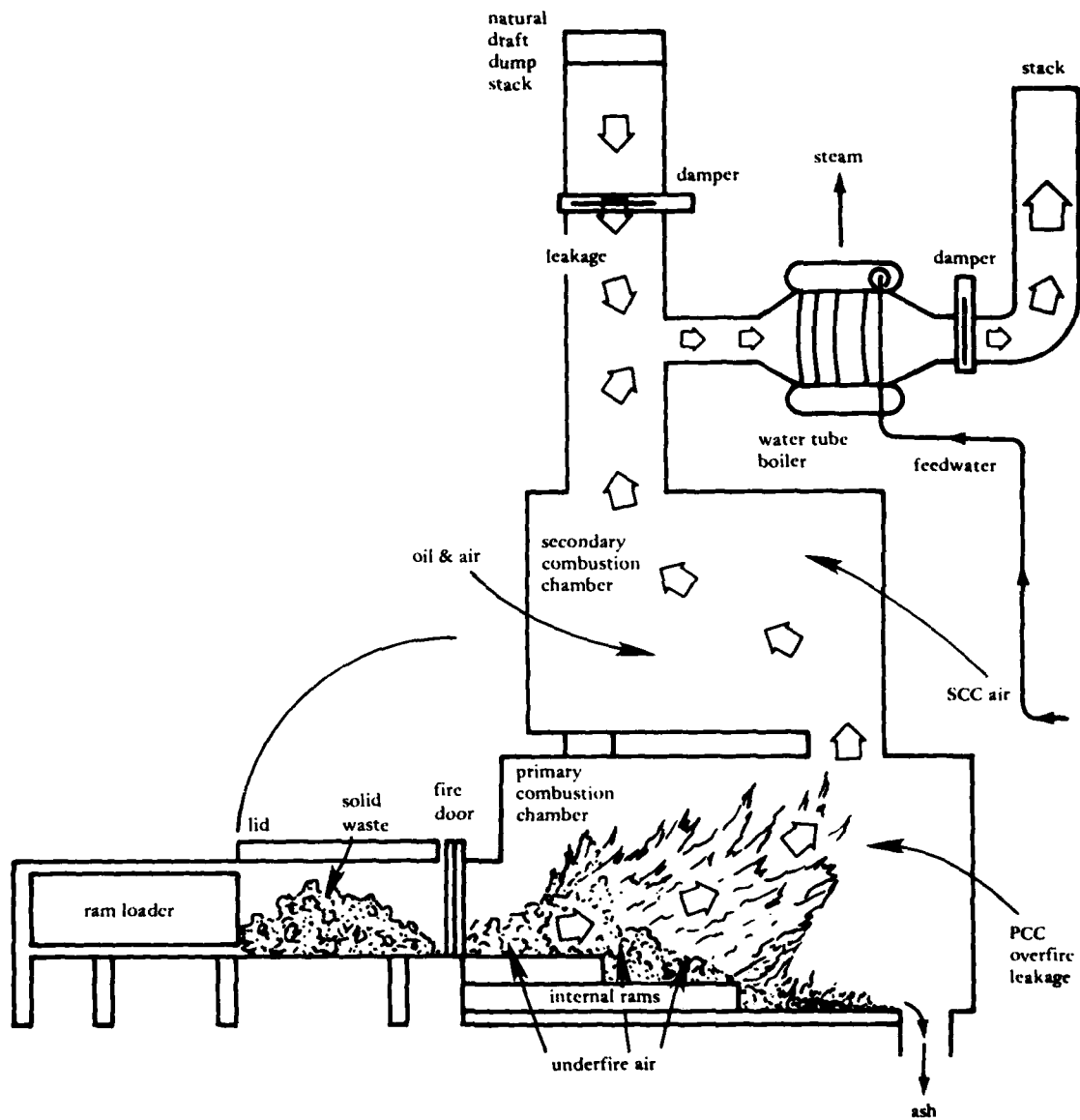
Figure 1. Schematic of Jacksonville Naval Air Station heat recovery incinerators.

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Notes:
(1) Feed rate = 1,500 lb/hr wet
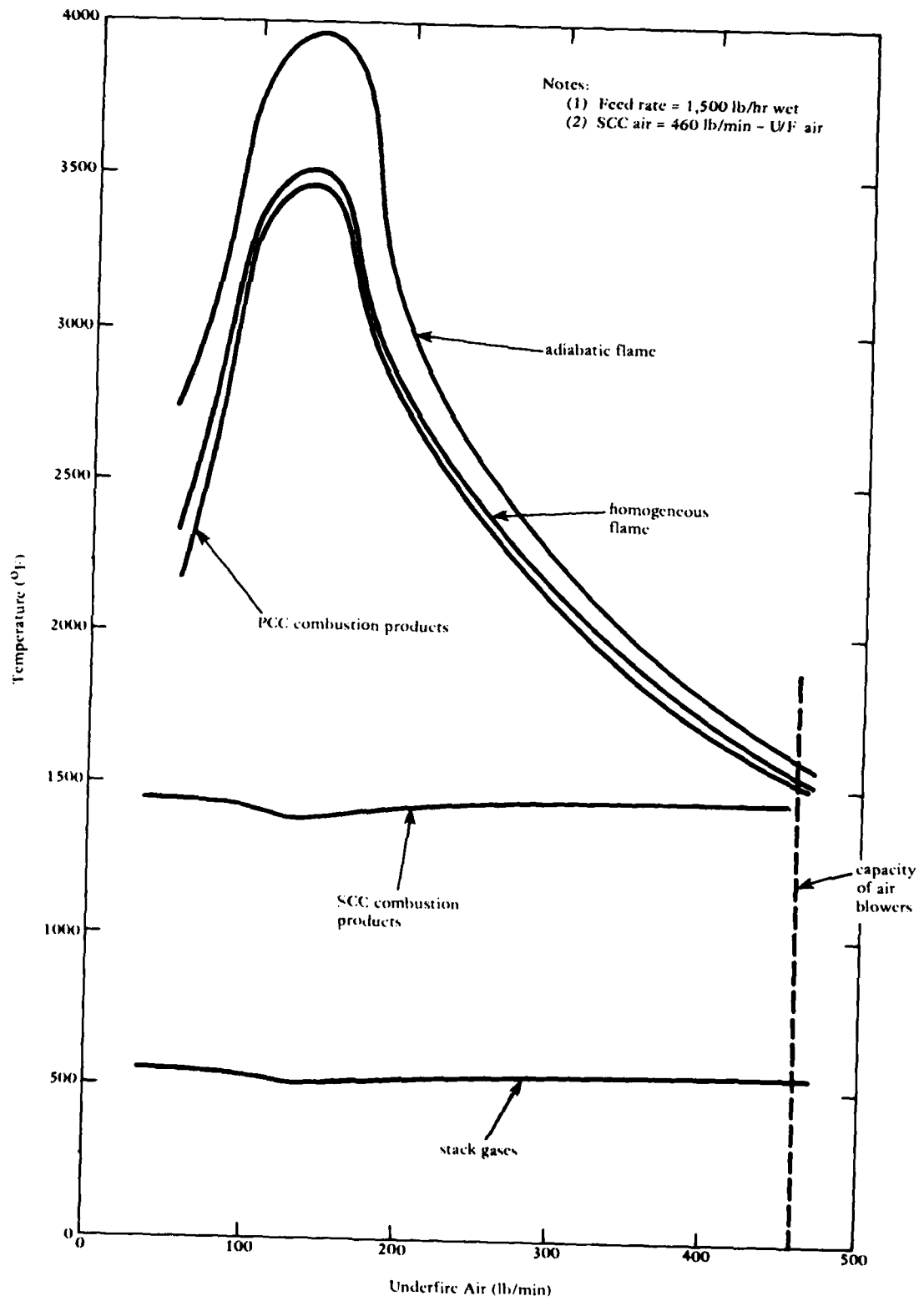(2) SCC air = 460 lb/min – U/F air

Figure 2. Temperatures throughout NAS Jacksonville heat recovery incinerator when burning "composite" waste.
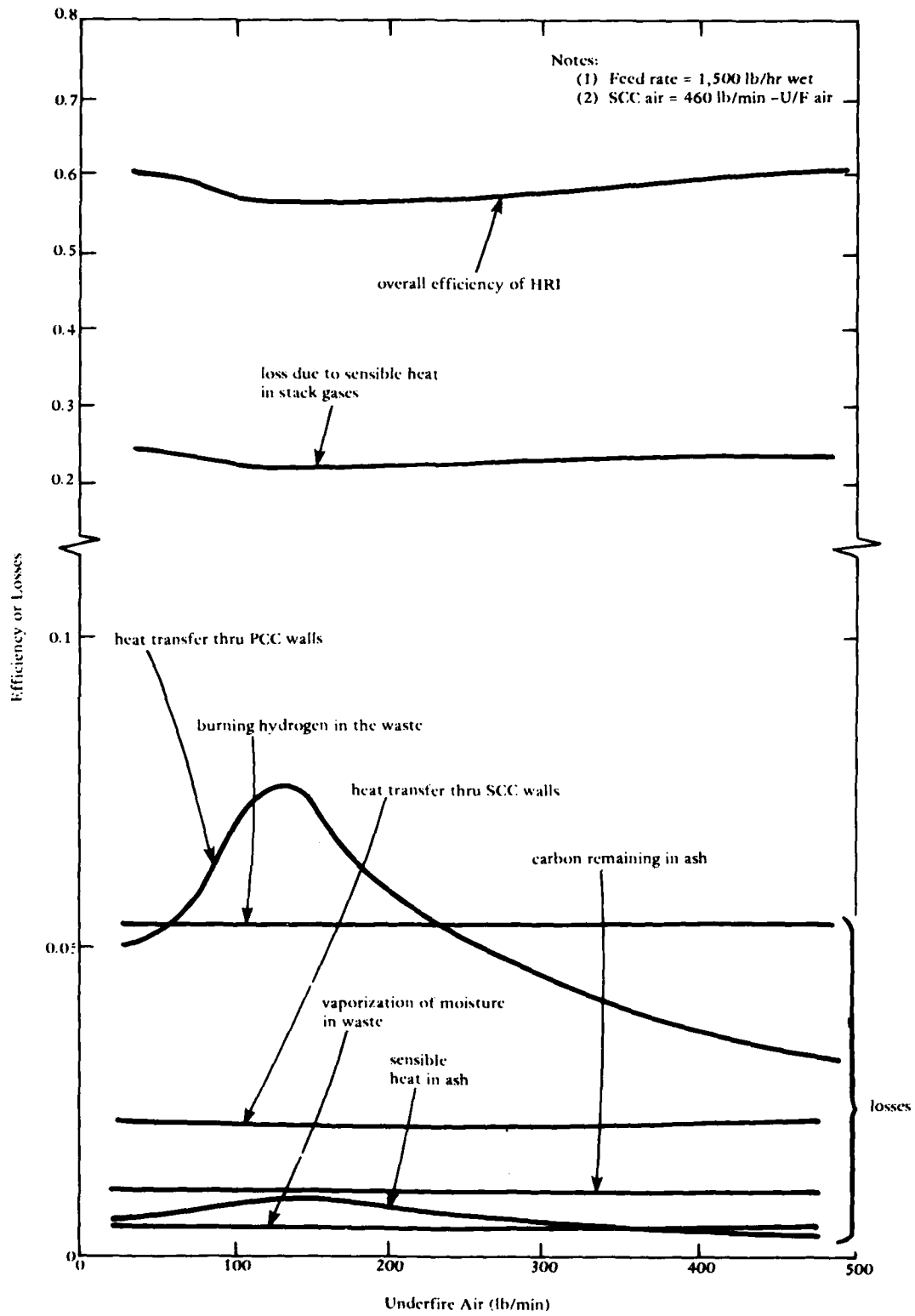
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Figure 3. Efficiency of NAS Jacksonville heat recovery incinerator burning "composite" waste.
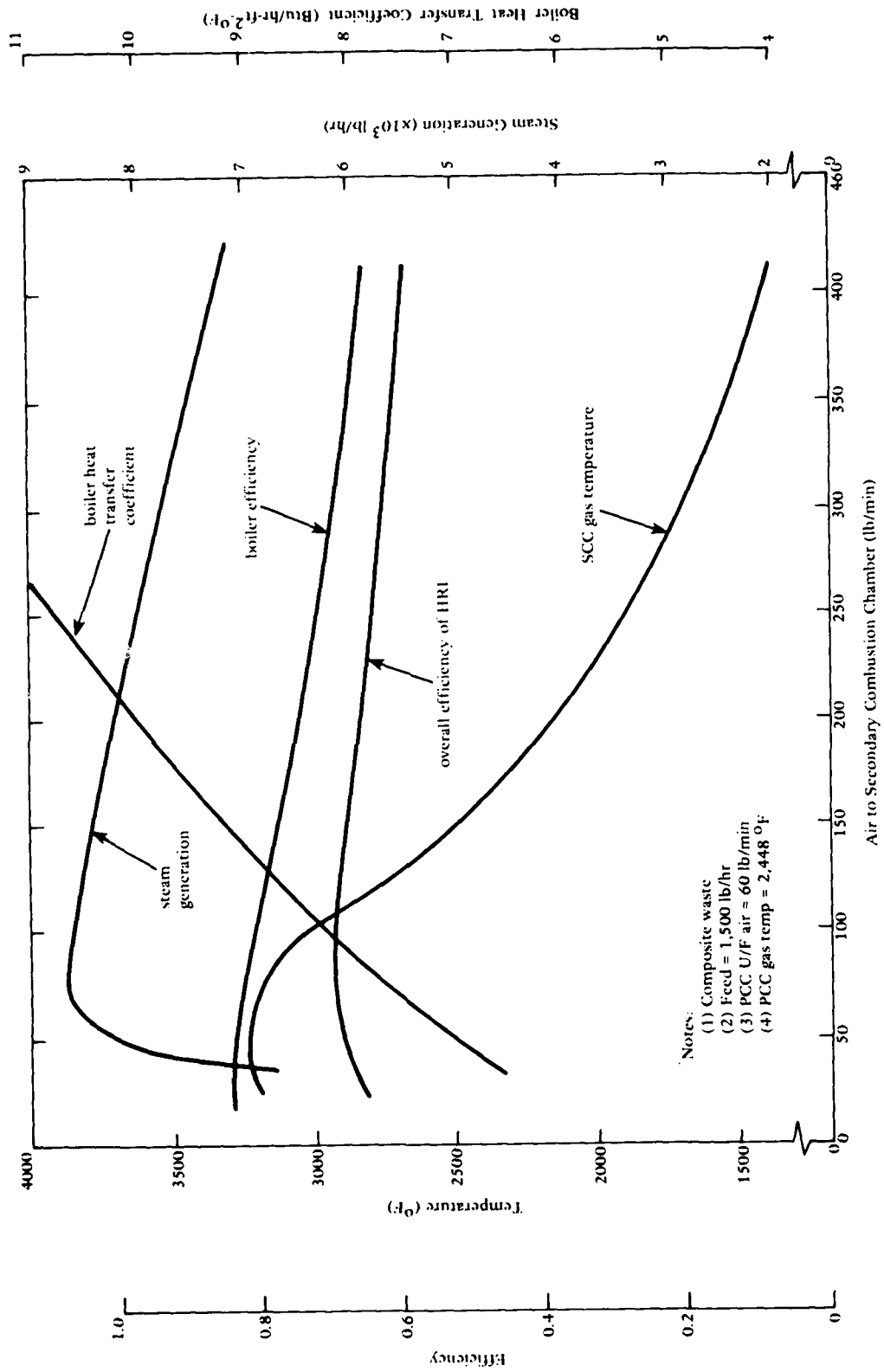
Figure 4. Performance of the NAS Jacksonville heat recovery incinerator when operating in an optimized starved-air mode.
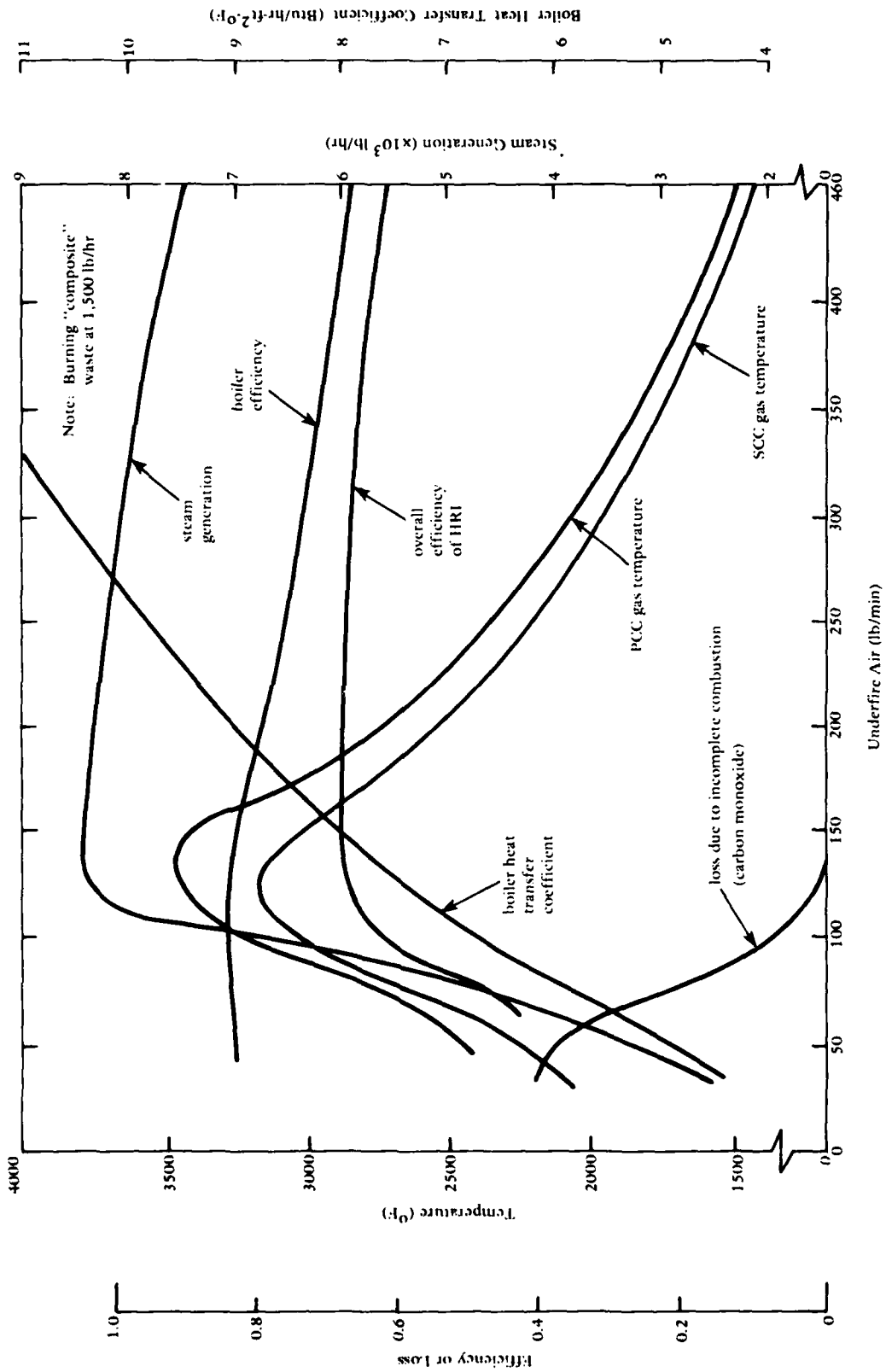
Notes:
(1) Composite waste
(2) Feed = 1,500 lb/hr
(3) PCC U/F air = 60 lb/min
(4) PCC gas temp = 2,448 °F

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Figure 5. Performance of the NAS Jacksonville heat recovery incinerator when no air is supplied to the secondary combustion chamber.
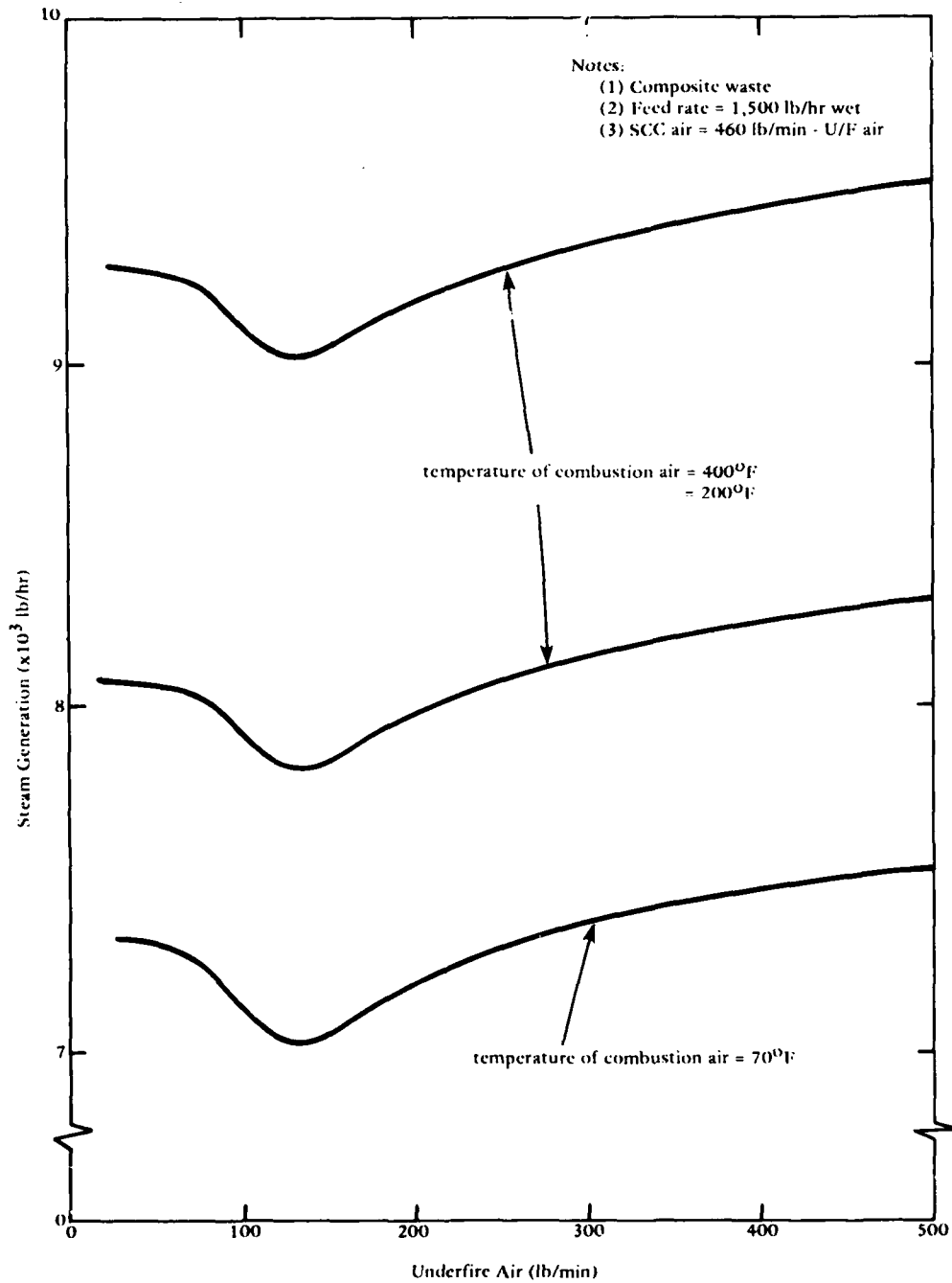
19

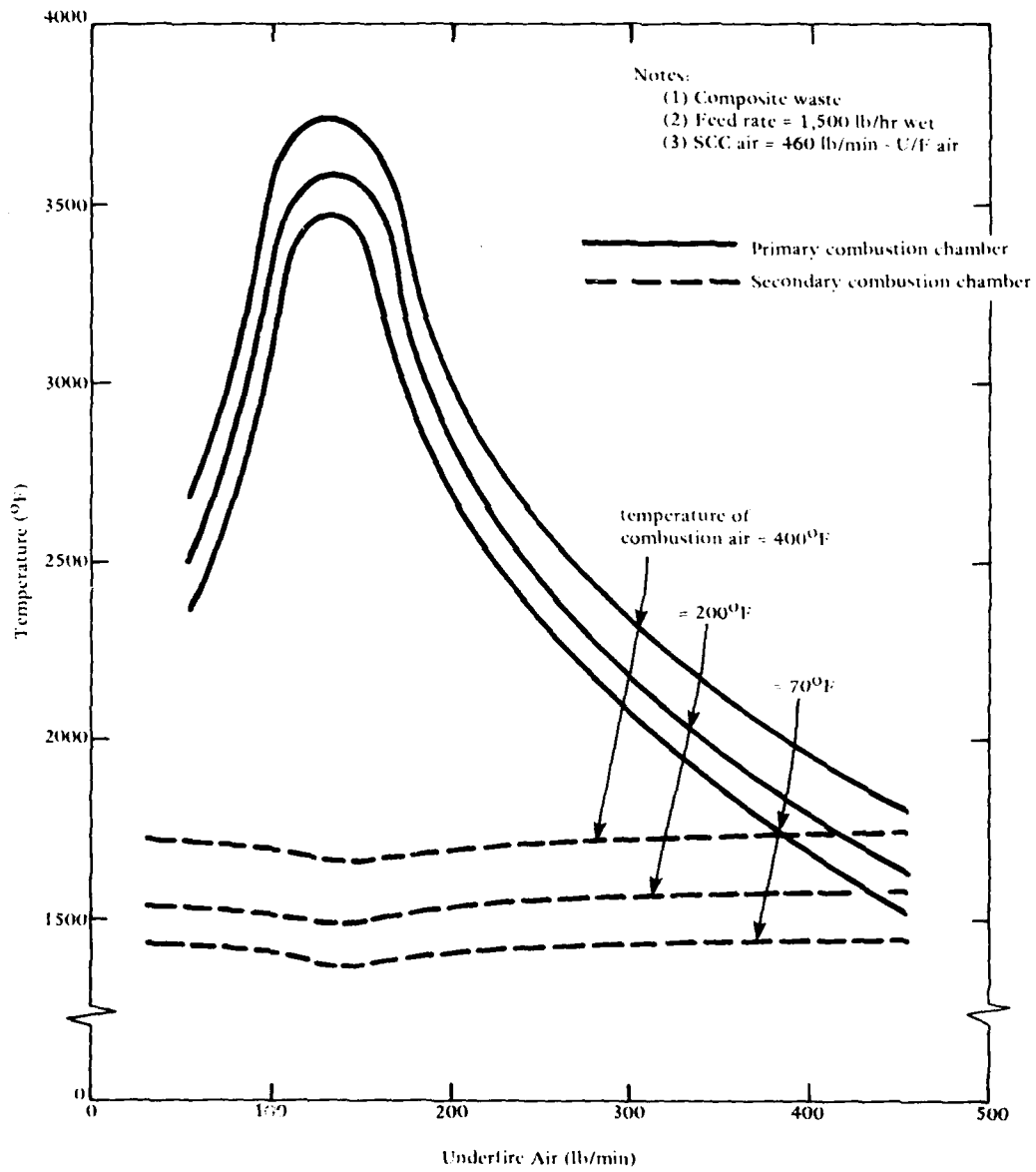Figure 6. Effect of the initial temperature of combustion air on the performance of NAS Jacksonville HRI.

Figure 7. Effect of the initial temperature of the combustion air on NAS Jacksonville HRI combustion chamber temperatures.
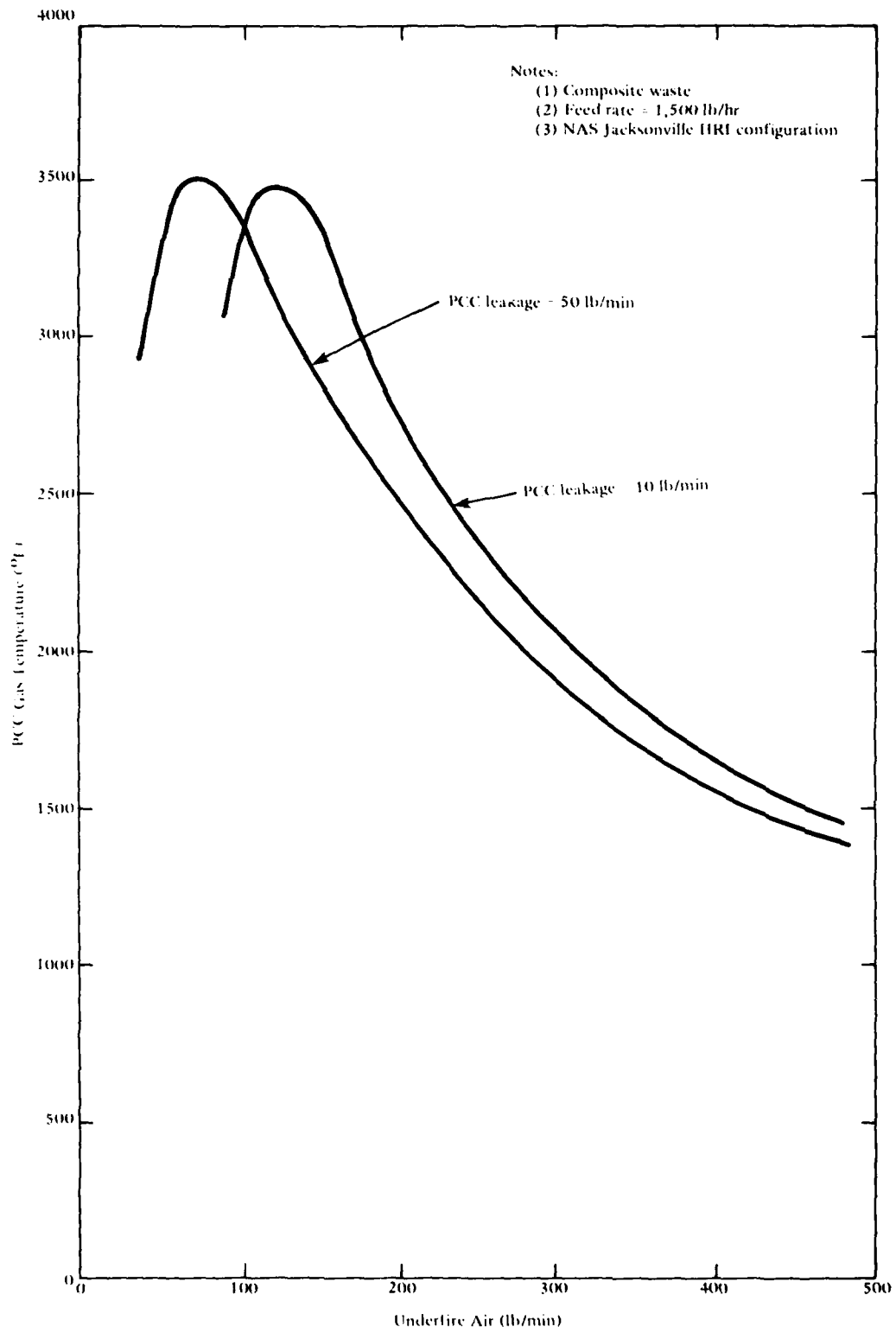
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Notes:
(1) Composite waste
(2) Feed rate = 1,500 lb/hr
(3) NAS Jacksonville HRI configuration

PCC leakage = 50 lb/min

PCC leakage = 10 lb/min

Underfire Air (lb/min)

PCC Gas Temperature (°F)

Figure 8. Effect of overfire air leakage on the gas temperature in the primary combustion chamber.
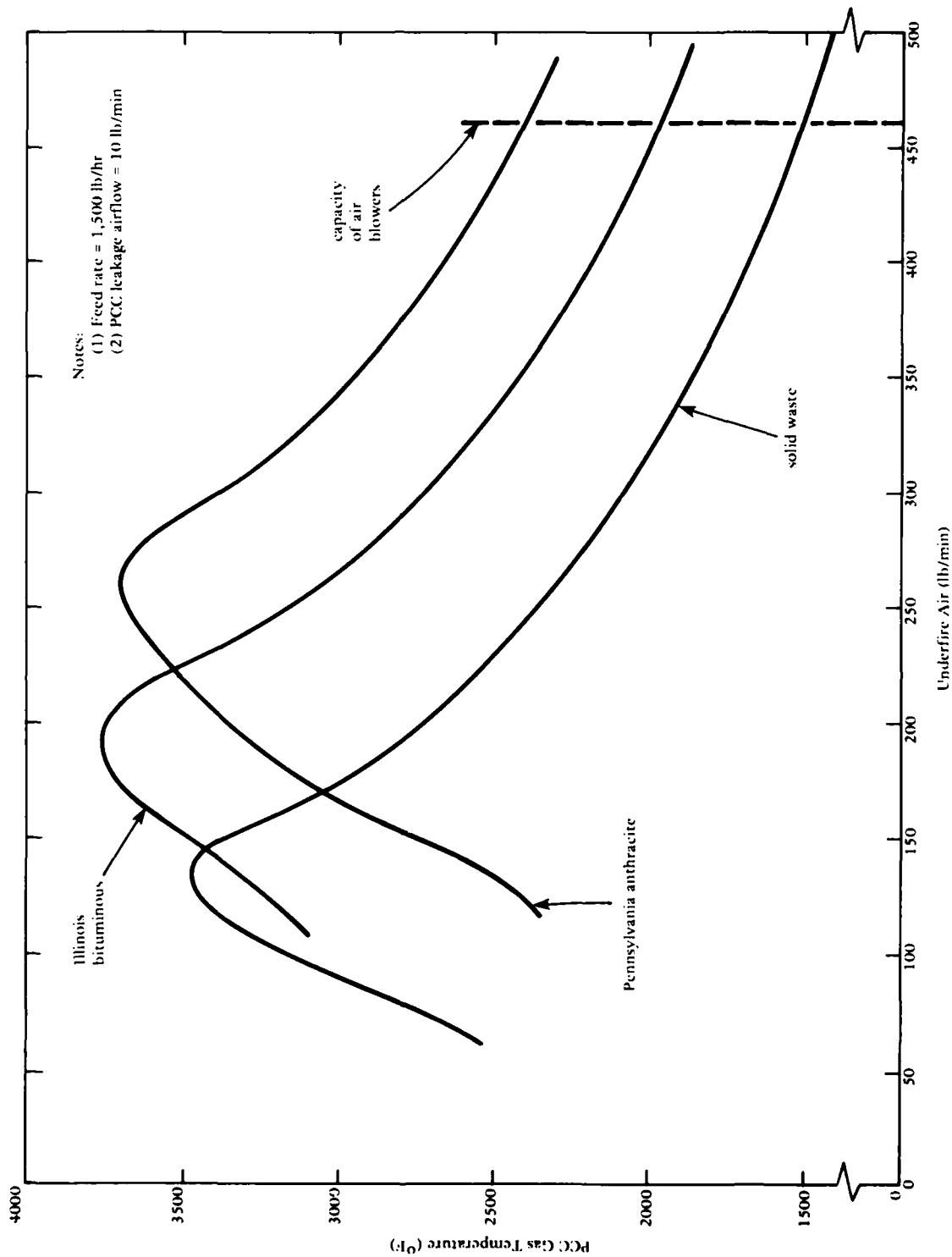
22

Figure 9. Comparison of the theoretical air required for the combustion of solid waste and coal.
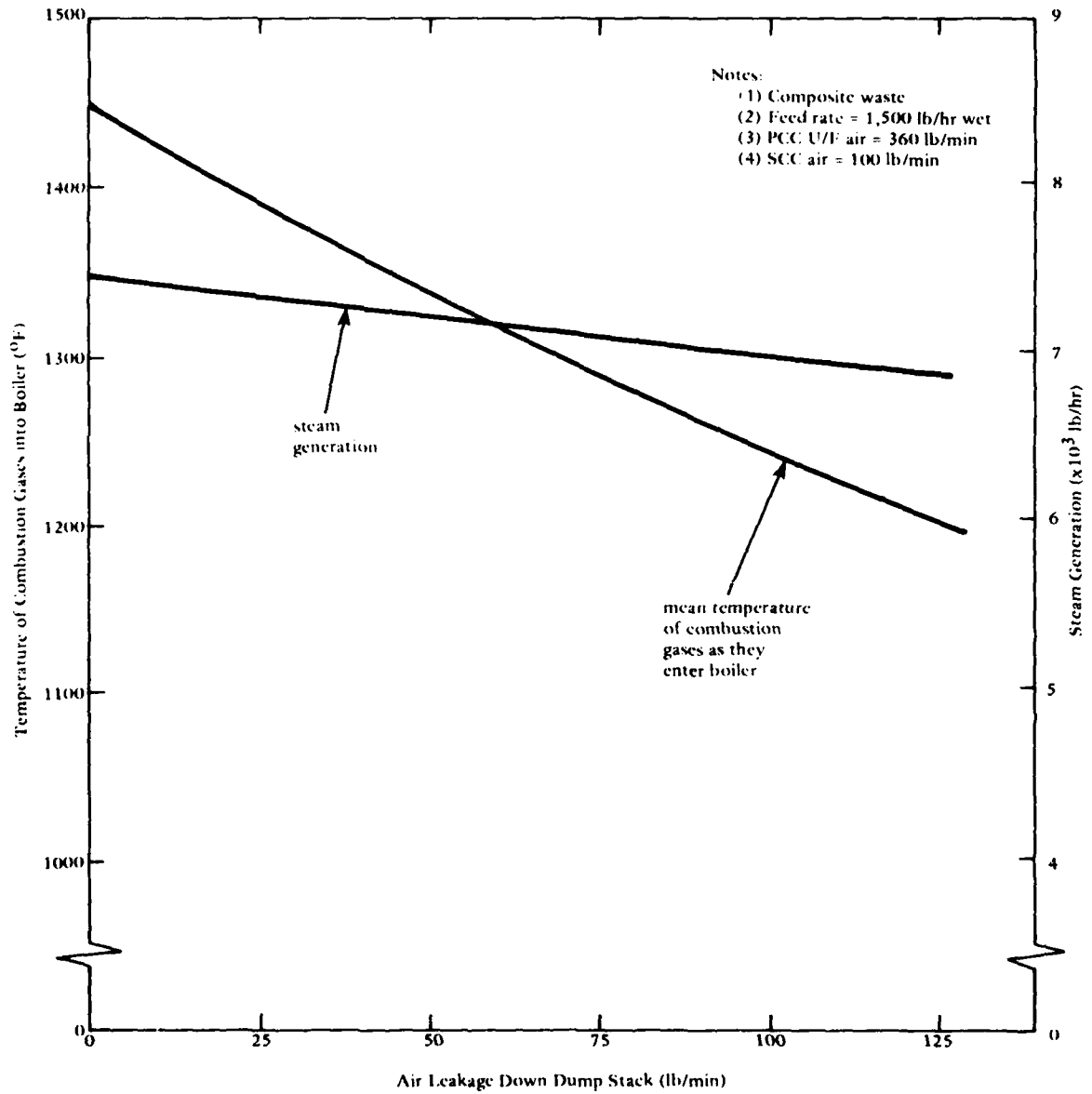
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Figure 10. Effect of leakage down the dump stack on steam generated by the NAS Jacksonville heat recovery incinerator.
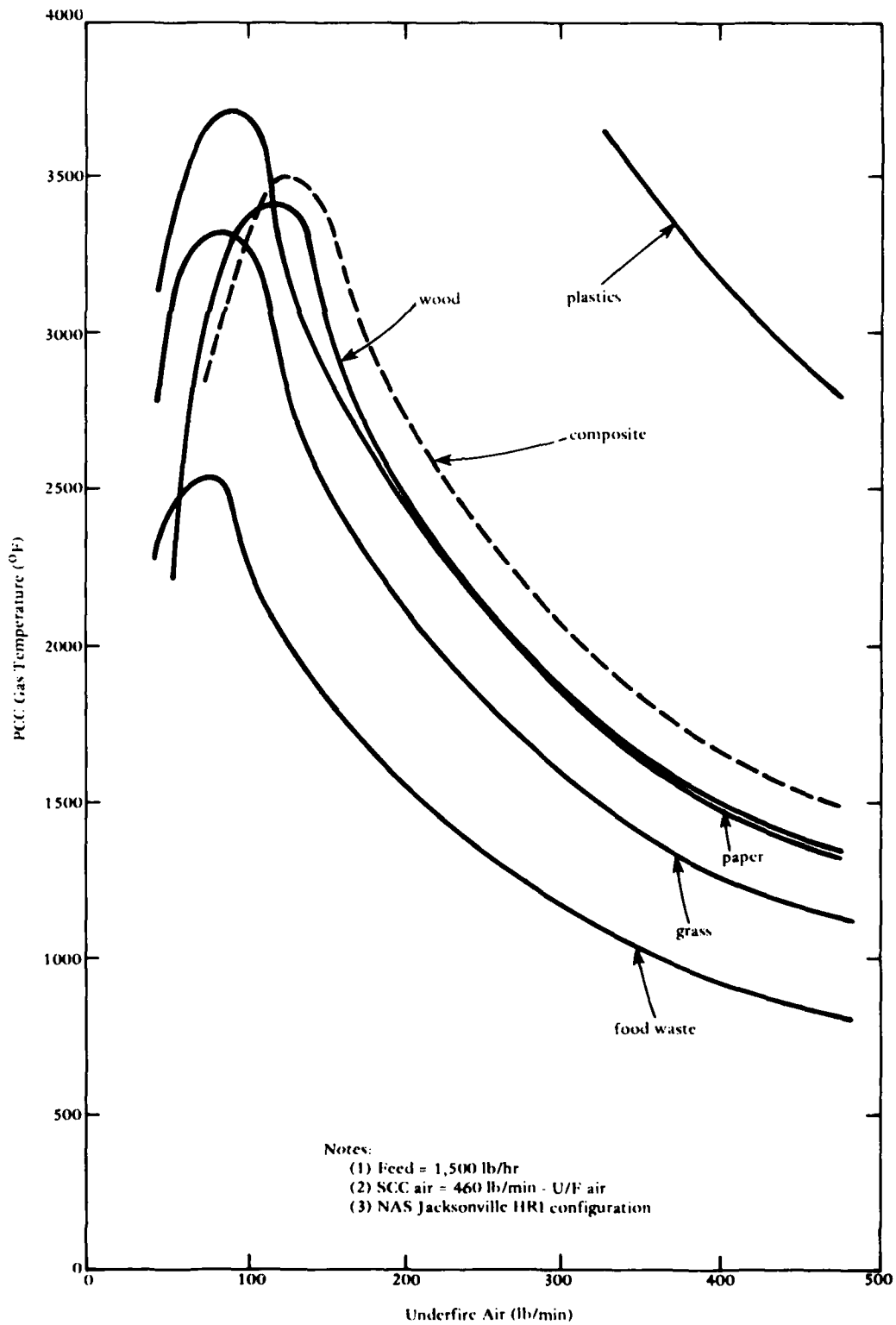
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Figure 11. Effect of the type of waste on the primary combustion chamber gas temperature.

Notes:
(1) Feed = 1,500 lb/hr
(2) SCC air = 460 lb/min - U/F air
(3) NAS Jacksonville HRI configuration

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Notes:
(1) Waste feed = 1,500 lb/hr wet
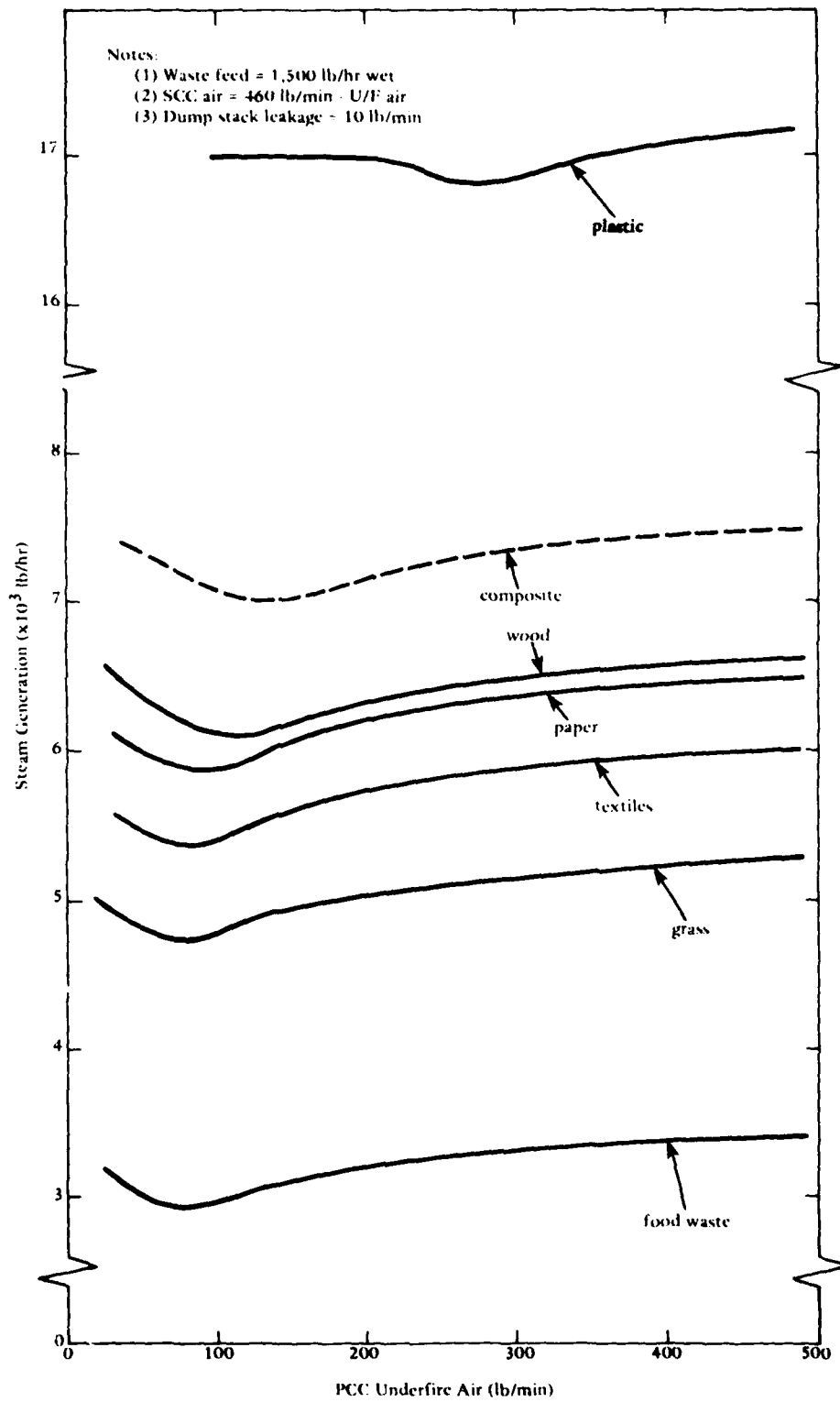(2) SCC air = 460 lb/min - U/F air
(3) Dump stack leakage = 10 lb/min

Figure 12. Effect of type of waste on NAS Jacksonville heat recovery incinerator steam generation.

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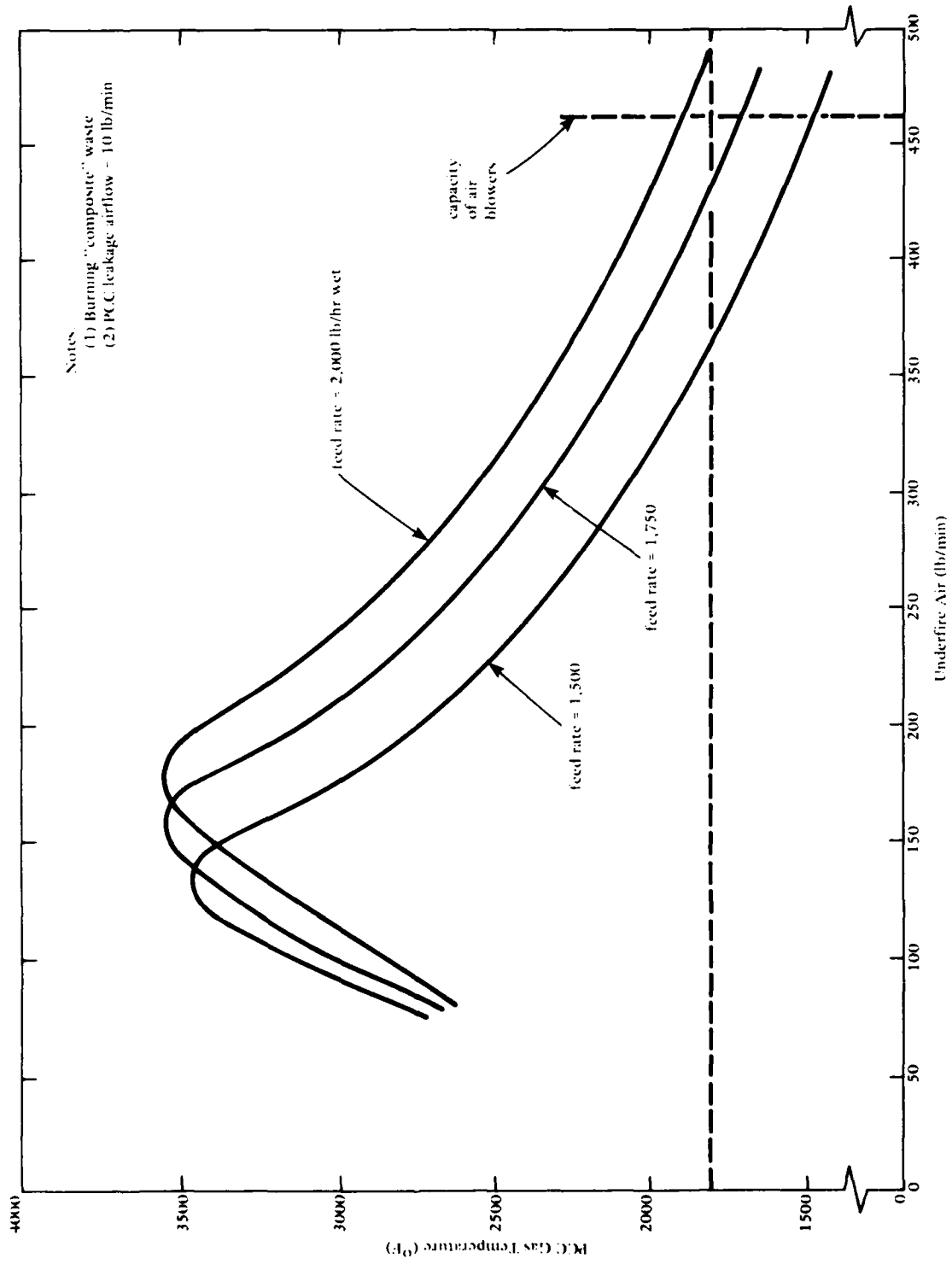
Figure 13. Effect of waste feed rate on the NAS Jacksonville HRI primary combustion chamber temperature.

Notes
(1) Burning "composite" waste
(2) PCC leakage airflow – 10 lb/min

feed rate = 2,000 lb/hr wet

feed rate = 1,750

feed rate = 1,500

capacity of air blowers

Underfire Air (lb/min)

PCC Gas Temperature (°F)

Figure 14. Effect of waste feed rate on NAS Jacksonville heat recovery incinerator steam generation.

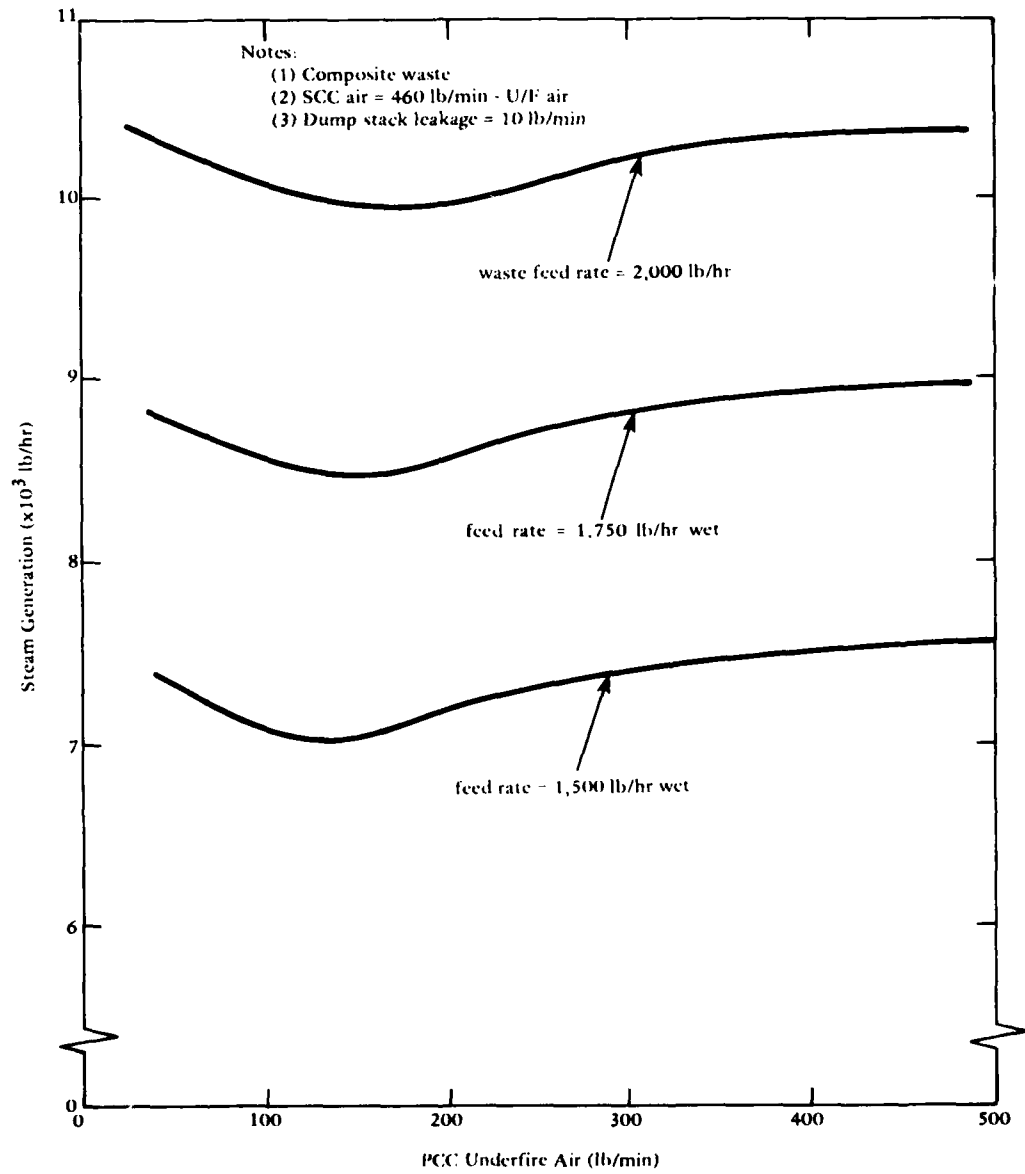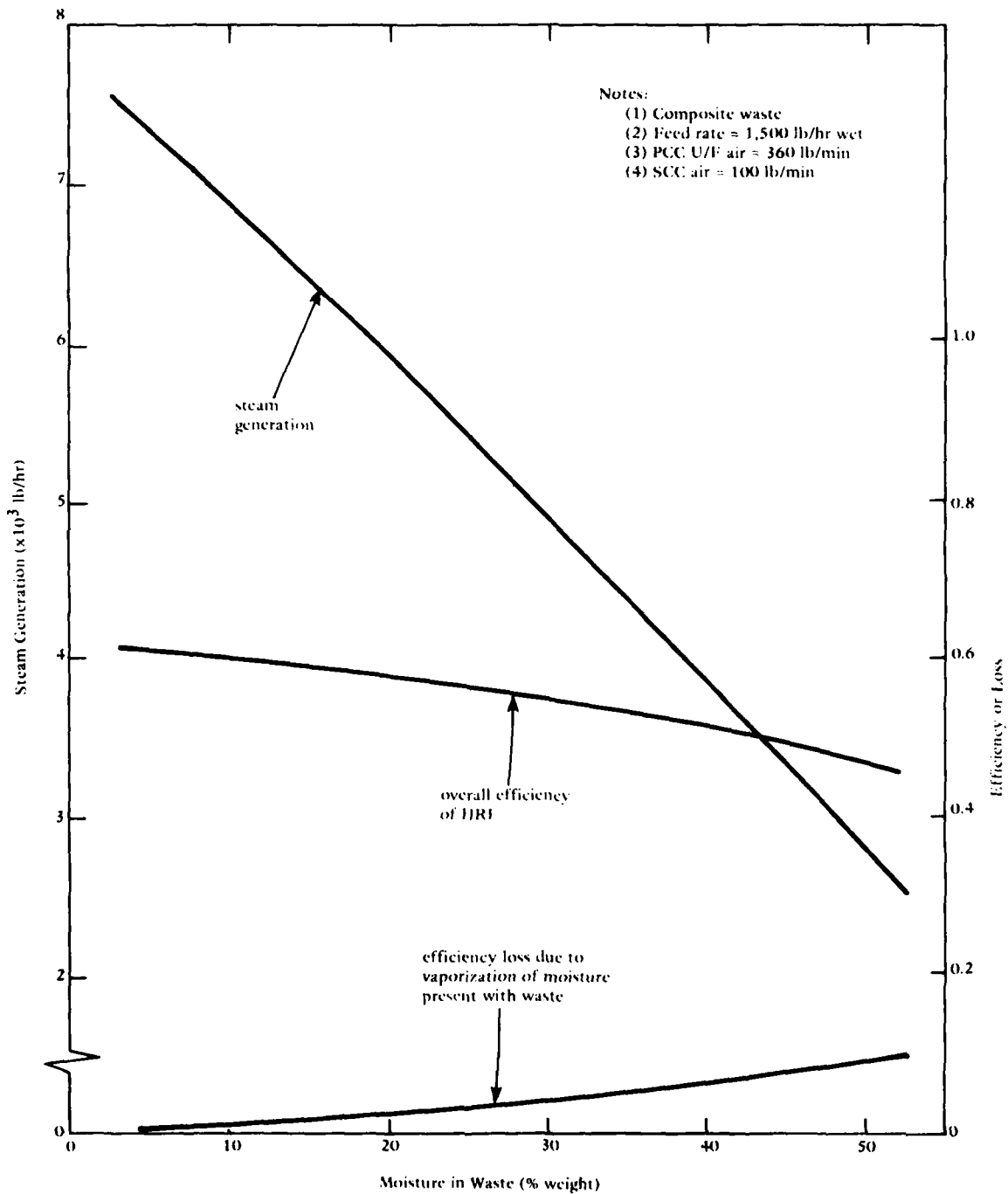Figure 15. Effect of moisture with the waste on the performance of the NAS Jacksonville heat recovery incinerator.

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Notes:
(1) Composite waste
(2) Feed rate = 1,500 lb/hr
(3) PCC U/F air = 360 lb/min
(4) SCC air = 100 lb/min

steam
generation

boiler efficiency

overall efficiency
of HRI

loss due to
sensible heat
in stack gas

design
performance

Boiler Overall Heat Transfer Coefficient (Btu/hr-ft^2-°F)

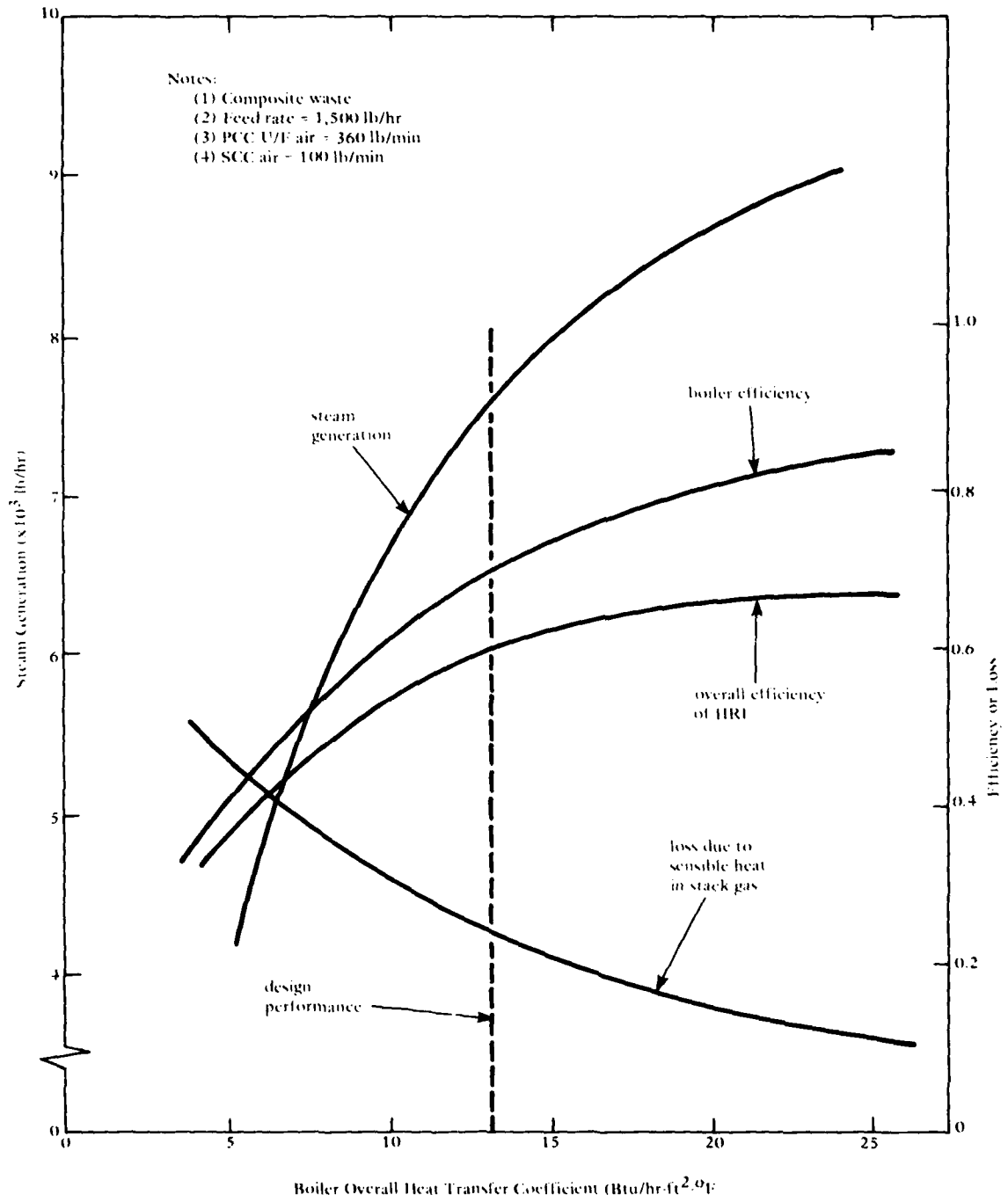Steam Generation (×10^3 lb/hr)

Efficiency or Loss

Figure 16. Effect of boiler performance on the overall performance of the NAS Jacksonville
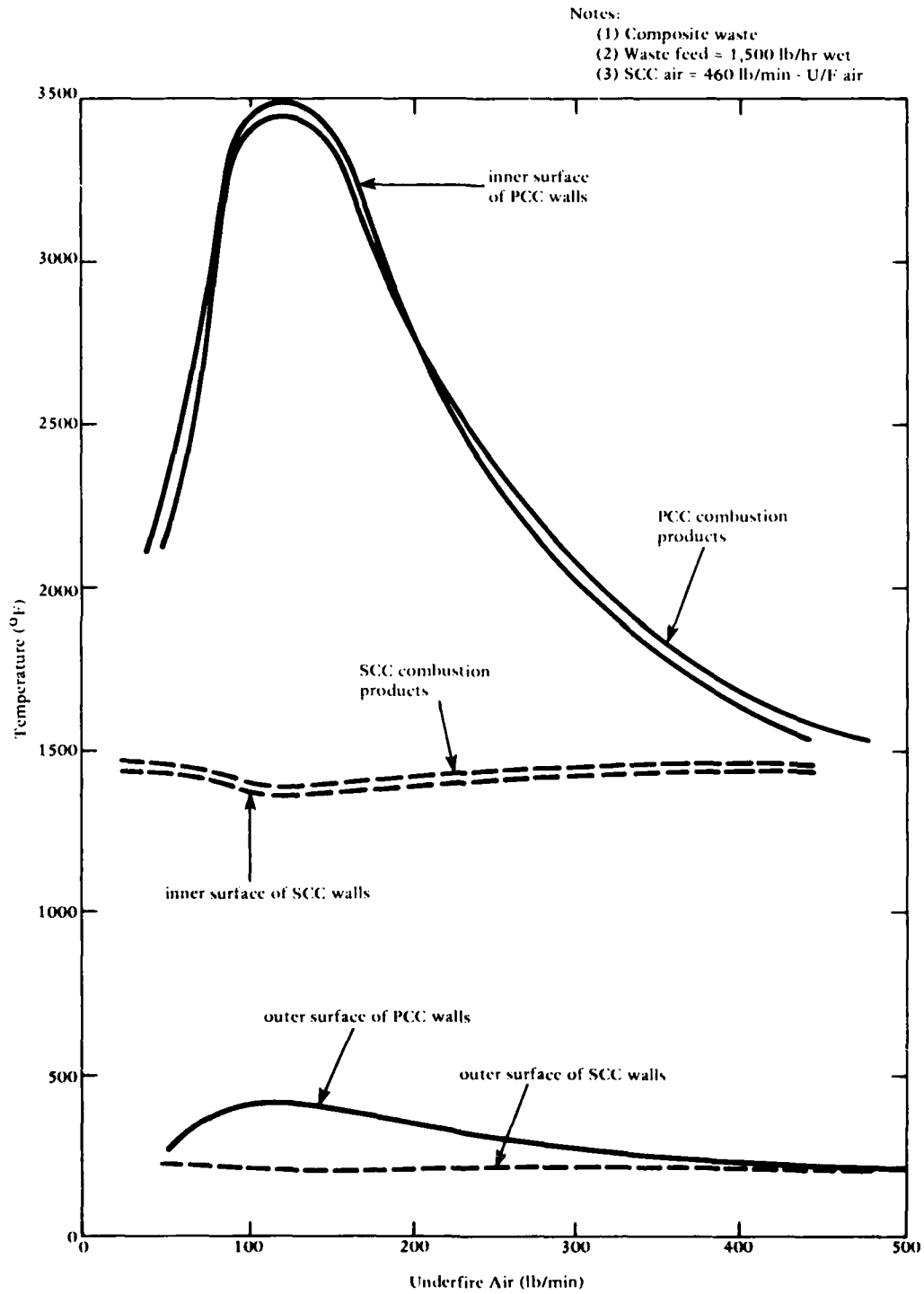heat recovery incinerator.

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Figure 17. Temperatures of the walls of the NAS Jacksonville heat recovery incinerator.
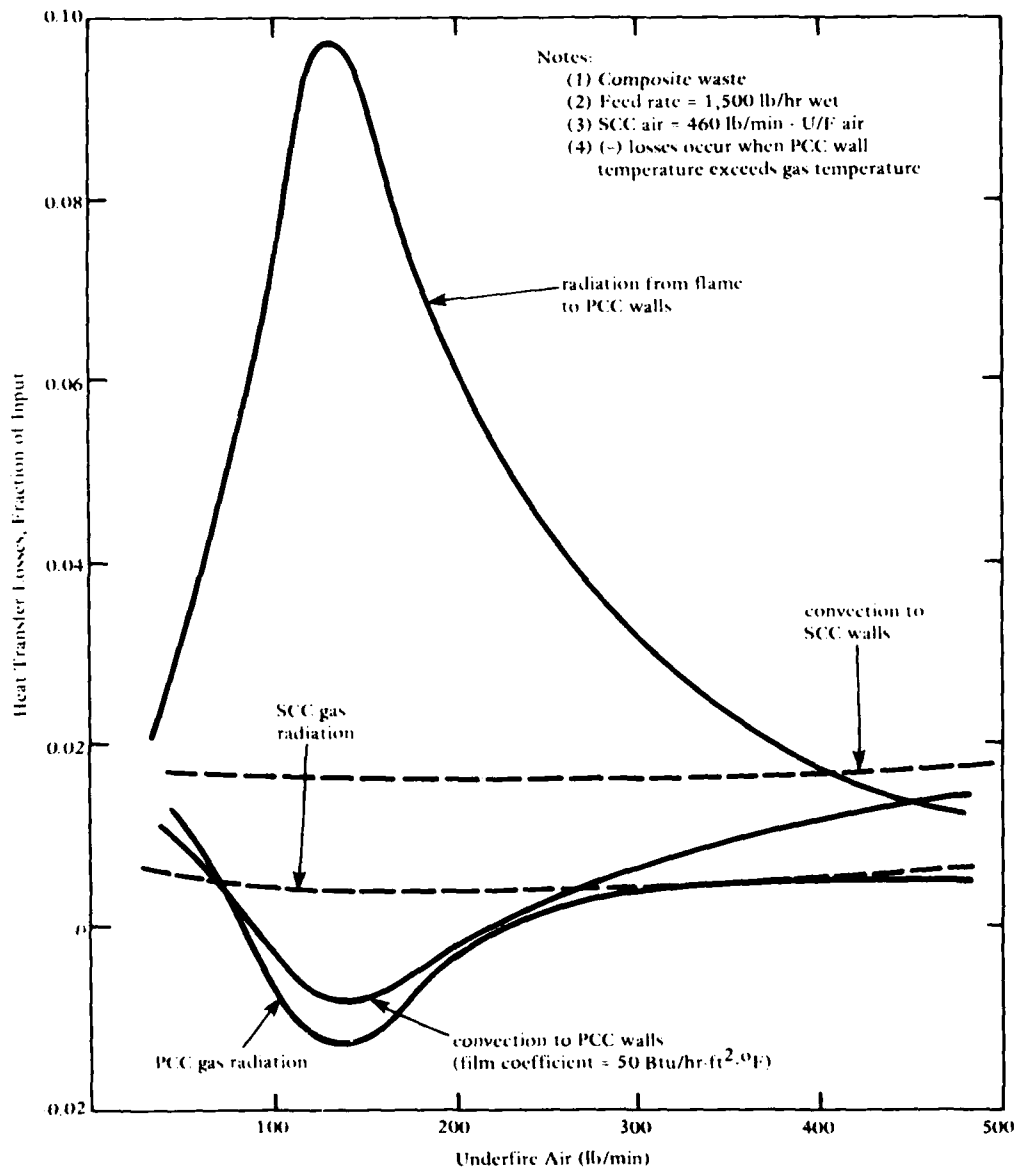
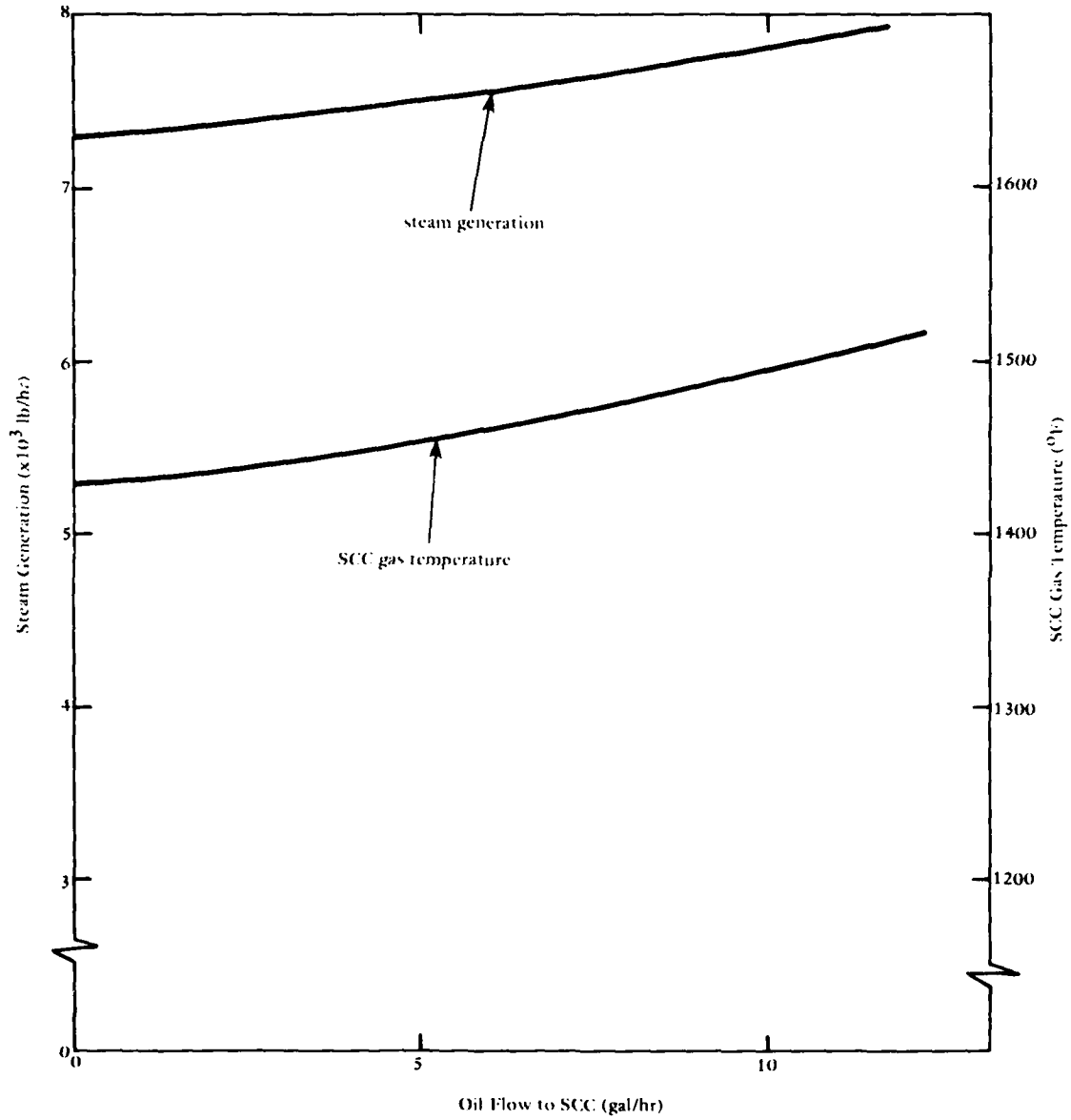Figure 18. Summary of NAS Jacksonville heat recovery incinerator heat transfer losses.

Figure 19. Effect of injecting oil into the secondary combustion chamber on the steam generation of the NAS Jacksonville heat recovery incinerator.

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Appendix A

BASELINE ESTABLISHED FOR PARAMETRIC EVALUATION
OF NAS JAX HEAT RECOVERY INCINERATORS (HRI)

The following magnitudes of HRI variables establish the baseline around which the parametric evaluation is conducted. Several of these parameters were never studied (e.g., incinerator convection coefficients), and several were given only a cursory examination (e.g., oil flows). Regardless, all affect the performance of the HRI and, unless otherwise specified, can be considered as input to the analyses.

Ash

    Removal rate = 200 lb/hr
    Higher heating value = 1,417 Btu/lb

    Ultimate analysis (percent of dry weight)

        Carbon 5.00
        Other 95.00

Oil as Auxiliary Fuel

    To primary combustion chamber = 0 lb/hr
    To secondary combustion chamber = 16 lb/hr
    Higher heating value = 19,700 Btu/lb

    Ultimate analysis (percent of dry weight)

        Carbon 86.00
        Hydrogen 12.00
        Oxygen 0.50
        Nitrogen 0.00
        Other 1.50

Combustion Air

    Total output of blowers   = 460 lb/min
    With primary oil burner   =   0
    With secondary oil burners = 12 lb/min

Leakage Air

To primary combustion chamber   = 10 lb/min
To secondary combustion chamber = 0
Down the dump stack   = 10 lb/min

Heat Transfer Parameters

Ambient air temperature   = 70°F
Surface area of flame front = 112 ft^2
Surface area of PCC   = 488 ft^2
Surface area of SCC   = 360 ft^2
Emissivity of outer skin of incinerator ≈ 0.75

Convection film coefficients

Inner surface of PCC = 50 Btu/hr-ft^2-°F
Inner surface of SCC = 50 Btu/hr-ft^2-°F
Outer surface of incinerator = 5 Btu/hr-ft^2-°F

Thermal conductance through walls

Of primary combustion chamber = 0.75 Btu/hr-ft^2-°F
Of secondary combustion chamber = 0.75 Btu/hr-ft^2-°F

Mean beam length

Of primary combustion chamber   = 4.7 ft
Of secondary combustion chamber = 3.9 ft

Boiler Characteristics

Surface area of tubes = 968 ft^2

Feed water properties

Temperature = 227°F
Enthalpy   = 195 Btu/lb

Steam properties

Temperature = 353°F
Pressure   = 140 psia
Enthalpy   = 1,193 Btu/lb

Boiler blowdown = 2.0% of steam generated

Overall heat transfer coefficient = 12.94 Btu/hr-ft^2-°F

Power Requirements

Blowers, pumps, waste processing equipment, etc. ≈ 100 kW

Appendix B

COMPOSITION OF WASTE UTILIZED FOR PARAMETRIC EXAMINATION
OF NAS JAX HEAT RECOVERY INCINERATORS (HRI)

Table B-1 summarizes the compositions of the different types of
waste considered in these HRI studies. The samples analyzed were acquired
at the Naval Air Station during September 1980. Except for a low moisture
content, indigenous to the Jacksonville area, the sample components
compare closely with other data of this type (Ref 6).
For purposes of establishing a baseline, a "composite" sample was
formed. The composition of this sample is as follows (percent by weight):

Paper 34.7

Corrugated Boxes 27.9

Plastics 14.8

Food waste 17.1

Textiles 1.2

Grass 2.1

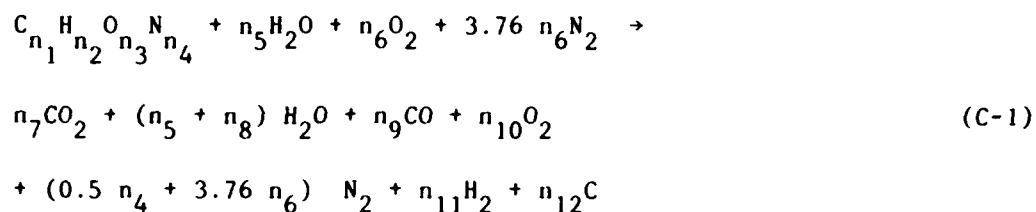Wood 2.2

Table B-1. Composition of Waste Samples Considered in Parametric Evaluation of NAS JAX Heat Recovery Incinerator (percent by weight)

Ultimate Analysis (dry)	Type of Waste							
	Paper	Corrugated Boxes	Plastics	Food Waste	Textiles	Grass	Wood	Composite
Carbon	39.54	41.30	81.36	46.25	34.25	40.43	45.65	47.56
Hydrogen	6.14	6.24	7.91	6.78	6.43	5.81	6.07	6.22
Oxygen	52.47	49.51	10.41	33.25	53.31	42.97	45.61	43.84
Nitrogen	0.13	0.08	0.05	2.20	0.23	1.35	0.20	0.09
Other	1.72	2.87	0.27	11.52	5.78	9.44	2.47	2.29
Moisture	7.93	7.68	0.05	48.95	4.33	24.56	8.07	4.32
Heating Value, Btu/lb dry	8,249	7,522	17,390	9,704	7,492	8,692	8,375	8,957

Appendix C

PROCEDURE FOR MODELING THE OPERATION OF NAS JACKSONVILLE
HEAT RECOVERY INCINERATORS (HRI)

The mathematical simulation of the NAS JAX heat recovery incinerators is based on the hypothesized combustion reaction

$$C_{n_1} H_{n_2} O_{n_3} N_{n_4} + n_5 H_2O + n_6 O_2 + 3.76 \, n_6 N_2 \rightarrow$$

$$n_7 CO_2 + (n_5 + n_8) \, H_2O + n_9 CO + n_{10} O_2 \qquad\qquad (C-1)$$

$$+ (0.5 \, n_4 + 3.76 \, n_6) \, N_2 + n_{11} H_2 + n_{12} C$$

In addition, it is assumed that

1. Steady state exists.

2. Kinetic and potential energy changes are negligible.

3. The reaction goes to completion regardless of the temperature.

4. Combustion is diffusion controlled, limited only by the mass flow rates of fuel and oxygen.

5. The products of combustion are perfectly mixed.

6. Therefore, all temperature gradients are normal to tne incinerator walls; the individual components of the incinerator may be represented one dimensionally.

The molar coefficients n_1 through n_5 are obtained from ultimate and proximate analysis of the waste (fuel); n_6 is from the air supplied for combustion. Equation C-1 is balanced by applying conservation of species and allocating available oxygen in order of increasing activation energy for combustion in air: first to hydrogen to form water vapor, then to carbon to form carbon monoxide (Ref 2). Any oxygen remaining is assumed to oxidize the carbon monoxide to form carbon dioxide.

Stoichiometric Air

Stoichiometric air is the air required to oxidize the fuel to water vapor and carbon dioxide,

$$n_{7,STOICH} = n_1 , \quad n_{8,STOICH} = \frac{n_2}{2}$$

$$n_{6,STOICH} = \frac{2\, n_{7,STOICH} + n_{8,STOICH} - n_3}{2}$$

$$\dot{M}_{AIR,STOICH} = 32\, n_{6,STOICH}\, \dot{M}_{FUEL,DRY}$$

where

$$\dot{M}_{FUEL,DRY} = \text{feed rate of waste (dry)}$$

Heat of Pyrolysis

The heat absorbed in breaking down the fuel is primarily a heat of pyrolysis since most types of waste (e.g., paper and wood) are pyrolyzing solids. Regardless of the mode, the energy required to break down the fuel is easily calculated once the stoichiometric products of combustion have been determined,

$$- Q_{FUEL} = - HHV_{FUEL,DRY} - n_{7,STOICH}\, Q_{CO_2} - n_{8,STOICH}\, Q_{H_2O}$$

C-2

where

HHV = higher heating value of fuel (dry)

Q_{CO} = heat of formation of carbon dioxide

Q_{H_2O} = heat of formation of water (liquid)

Adiabatic Flame Temperature

If all the energy released during the combustion reactions is assumed available to heat the products, an upper limit to the flame temperature can be determined. This temperature is usually referred to as the adiabatic flame temperature.

First, subtract the ash to derive the composition of the fuel actually burned,

$$n_1 = n_1 - \left(\frac{\dot{M}_{ASH}}{\dot{M}_{FUEL,DRY}} \right) \times (\text{moles of carbon in ash})$$

etc.

Air supplied to the flame is one of the independent variables affecting HRI performance. The coefficient n_6 of Equation C-1 is determined directly from the underfire airflow to the flame. Once the fuel composition and the air (oxygen) have been established, Equation C-1 is balanced using the method described above.

Subtracting the energy lost vaporizing the moisture in the fuel,

$$\hat{Q}_{LOST} = (\text{Mass fraction of moisture in fuel}) \times (\text{heat of vaporization of water at a pressure of 1 atmosphere})$$

C-3

the net heat released to the flame can be calculated,

$$\dot{q}_{FLAME} = -(n_7 Q_{CO_2} + n_8 Q_{H_2O} + n_9 Q_{CO}$$

$$- Q_{FUEL}^* + \hat{\dot{Q}}_{LOST}) \, \dot{M}_{FUEL,DRY}$$

and, since the mass flow through the flame is known,

$$\dot{M}_{FLAME} = \dot{M}_{FUEL,WET} + \dot{M}_{AIR} - \dot{M}_{ASH}$$

the adiabatic flame temperature can be determined by application of conservation of energy,**

$$T_{FLAME} = T_{DATUM} + [\dot{q}_{FLAME} + \dot{M}_{FUEL,DRY} \, \Delta h_{FUEL}(T_\infty)$$

$$+ \dot{M}_{AIR} \, C_{P,AIR}(T_\infty) \, \Delta T_\infty] \Big/ [\dot{M}_{FLAME} \, C_{P,MIX}(T_{FLAME})$$

$$+ C_{P,ASH} \, \dot{M}_{ASH} \,] \qquad\qquad (C-2)$$

where

T_{DATUM} = reference temperature of defined properties

$C_p(T)$ = specific heat at temperature T

$\Delta T_\infty = T_\infty - T_{DATUM}$

$\Delta h(T)$ = enthalpy at temperature T relative to T_{DATUM}

$C_{P,MIX} = \displaystyle\sum_{MIXTURE}$ (mole fraction x $C_{P,MEAN}$)

$$C_{P,MEAN}(T_2) \equiv \frac{1}{T_2 - T_{DATUM}} \int_{T_{DATUM}}^{T_2} C_p(T) \, dT$$

---

*For most fuels, $Q_{FUEL} < 0$.

**Where applicable, a perfect gas is being assumed.

Note that the temperature dependency of specific heat* makes Equation C-2 nonlinear. The relationships of Sweigert and Beardsley (Ref 7) were used to calculate specific heats as a function of temperature. These relationships and Equation C-2 were solved simultaneously using a Newton-Raphson iteration.

Primary Combustion Chamber Temperatures

Temperatures in the primary combustion chamber are calculated by solving the energy equations governing the flame front, the combustion chamber interior, and the walls of the PCC. Combustion products in both the flame and the PCC interior are assumed to be perfectly mixed. The homogeneous flame temperature derived in this manner can be considered a lower limit to the actual flame temperature.

The flame composition is already known from the adiabatic calculations. Composition of the combustion products in the primary combustion chamber is determined in an analogous manner, taking into account both oil injected into the chamber and possible air leakage.

If underfire air is insufficient for the complete combustion of the fuel (waste) and the oil, PCC air leakage will induce further chemical reaction and, thus, energy release in the primary combustion chamber,

$$\dot{q}_{PCC} = - (n_7 \, Q_{CO_2} + n_8 \, Q_{H_2O} + n_9 \, Q_{CO} - Q_{FUEL}$$

$$- Q_{OIL} + \hat{Q}_{LOST}) \, \dot{M}_{FUEL,DRY} - \dot{q}_{FLAME}$$

Energy terms included in the PCC analyses are illustrated schematically on Figure C-1. Applying conservation of energy to the flame,

---

*The specific heat of ash is assumed to be a constant.

$$\dot{M}_{FLAME}\, C_{P,MIX}(T_{FLAME})\, \Delta T_{FLAME} + \dot{M}_{ASH}\, C_{P,ASH}\, \Delta T_{FLAME} + \dot{q}_{RAD,F\to W}$$

$$+\; \dot{q}_{RAD,F\to G} - \dot{q}_{FLAME} - \dot{M}_{FUEL,DRY}\, \Delta h_{FUEL}(T_{\infty})$$

$$-\; \dot{M}_{AIR}\, C_{P,AIR}(T_{\infty})\, \Delta T_{\infty} = 0 \tag{C-3}$$

where

$\dot{q}_{RAD,F\to W}$ = radiation from flame to walls of PCC

$$= A_{FLAME}\, \sigma\{[1 - \varepsilon_{MIX}(T_{FLAME})]\, T_{FLAME}^4$$

$$- [1 - \varepsilon_{MIX}(T_{WALLS})]\, T_{WALLS}^4\}$$

$\dot{q}_{RAD,F\to G}$ = radiation from flame to products of combustion inside the PCC.

$$= A_{FLAME}\, \sigma\, [\varepsilon_{MIX}(T_{FLAME})\, T_{FLAME}^4 - \varepsilon_{MIX}(T_{PCC})\, T_{PCC}^4]$$

T_{FLAME} = homogeneous flame temperature

T_{PCC} = homogeneous temperature of products of combustion in PCC

T_{WALLS} = PCC inside wall temperature

A_{FLAME} = surface area of flame front

σ = Stefan-Boltzmann constant

Both the flame and inside of the PCC walls are assumed to act as black bodies. The products of combustion are assumed gray; the emissivities of these gases, $\varepsilon_{MIX}(T)$, are derived by curve fitting the data of Hottel et al. (Ref 8). Gas emissivities are thus a function of both composition and temperature.

Applying conservation of energy to the interior of the primary combustion chamber,

$$\dot{M}_{PCC} \, C_{P,MIX}(T_{PCC}) \, \Delta T_{PCC} + \dot{q}_{RAD,G \to W} + \dot{q}_{CONV,G \to W}$$

$$- \dot{q}_{PCC} - \dot{q}_{RAD,F \to G} - \dot{M}_{FLAME} \, C_{P,MIX}(T_{FLAME}) \, \Delta T_{FLAME}$$

$$- \dot{M}_{OIL} \, \Delta h_{OIL}(T_\infty) - \dot{M}_{AIR,LEAK} \, C_{P,AIR}(T_\infty) \, \Delta T_\infty = 0 \qquad \text{(C-4)}$$

where

$\dot{q}_{RAD,G \to W}$ = radiation from combustion gases to PCC walls

         = $A_{PCC} \, \sigma \, [\varepsilon_{MIX}(T_{PCC}) \, T_{PCC}^4 - \varepsilon_{MIX}(T_{WALLS}) \, T_{WALLS}^4]$

$\dot{q}_{CONV,G \to W}$ = convection heat transfer to PCC wall interior

         = $h_{CONV,PCC} \, A_{PCC} \, (T_{PCC} - T_{WALLS})$

A_{PCC} = surface area of PCC walls

$h_{CONV,PCC}$ = convection film coefficient

Finally, applying conservation of energy to the walls,

$$\dot{q}_{COND} - \dot{q}_{RAD,F \to W} - \dot{q}_{RAD,G \to W} - \dot{q}_{CONV,G \to W} = 0 \qquad \text{(C-5)}$$

$$\dot{q}_{RAD,W \to \infty} + \dot{q}_{CONV,W \to \infty} - \dot{q}_{COND} = 0 \qquad \text{(C-6)}$$

where

$\dot{q}_{COND}$ = conduction heat transfer through the walls

         = $KA_{PCC} \, (T_{WALLS} - T_{SHELL})$

$\dot{q}_{CONV,W \to \infty}$ = convection heat transfer off outer surface of PCC walls

         = $h_{CONV,\infty} \, A_{PCC} \, (T_{SHELL} - T_\infty)$

$\dot{q}_{RAD,W \to \infty}$ = radiation off outer surface of PCC walls

         = $A_{PCC} \, \sigma \, \varepsilon_{SHELL} \, (T_{SHELL}^4 - T_\infty^4)$

$$T_{SHELL} = \text{temperature of outer skin of PCC}$$

$$\varepsilon_{SHELL} = \text{emissivity of outer skin of PCC}$$

$$K = \text{conductance of PCC walls}$$

Equations C-3 through C-6, along with the relationships derived for temperature variations of specific heat and emissivity, are solved simultaneously for the temperatures T_{FLAME}, T_{PCC}, T_{WALLS}, and T_{SHELL}. Again, a Newton-Raphson iteration is employed.

Secondary Combustion Chamber

Temperatures of the combustion products and walls in the secondary combustion chamber are calculated in a manner analogous to the PCC problem. The energy equations governing the interior of the SCC, the inner walls, and outer skin are solved simultaneously while allowing both specific heat and emissivity to vary with temperature. If combustion has not been completed in the PCC, secondary air will induce further chemical reactions and require an additional heat source term in the energy equation governing the SCC interior.

Heat Recovery Boiler

The boiler unknowns are the steam generated, the total heat transferred between the combustion products and the feed water/steam, and the temperature of the combustion gases as they enter the stack. Temperature and pressure of the feed water and steam are assumed to be known.

Applying conservation of energy to the combustion gases, the feed water/steam, and to the overall heat recovery boiler (the individual terms are illustrated on Figure C2),

$$(\dot{M}_{SCC} + \dot{M}_{AIR,LEAK}) \, C_{P,MIX} \, (T_{STACK}) \, \Delta T_{STACK}$$

$$+ \, \dot{q}_{STEAM} - \dot{M}_{SCC} \, C_{P,MIX}(T_{SCC}) \, \Delta T_{SCC}$$

$$- \, \dot{M}_{AIR,LEAK} \, C_{P,AIR}(T_\infty) \, \Delta T_\infty \;=\; 0 \qquad\qquad (C\text{-}7)$$

$$\dot{M}_{STEAM} \, \Delta h_{STEAM}(T_{STEAM}) + BD \, \dot{M}_{STEAM} \, \Delta h_{STEAM}(T_{STEAM})$$

$$- \dot{q}_{STEAM} - (1 + BD) \, \dot{M}_{STEAM} \, \Delta h_{FEED}(T_{FEED}) = 0 \qquad (C-8)$$

$$\dot{q}_{STEAM} - U_{MEAN}(\dot{M},T) \, A_{BOILER} \, LMTD = 0 \qquad (C-9)$$

where

$\dot{M}_{SCC}$ = mass flow out of the secondary combustion chamber

$\dot{M}_{AIR,LEAK}$ = air leakage down the dump stack

$\dot{M}_{STEAM}$ = steam generated in the boiler

BD = boiler blowdown as a fraction of steam generated

$\Delta h_{FEED}(T_{FEED})$ = enthalpy of feed water at temperature T_{FEED} relative to T_{DATUM}

$\Delta h_{STEAM}(T_{STEAM})$ = enthalpy of steam at T_{STEAM} relative to T_{DATUM}

$\dot{q}_{STEAM}$ = heat transferred between combustion products and feed water/steam

T_{SCC} = homogeneous temperature of products of combustion in SCC

T_{STACK} = stack gas temperature (i.e., temperature of combustion gases as they exit the boiler)

A_{BOILER} = total surface area of boiler tubes

LMTD = logarithmic mean overall temperature difference, Figure C3

$$\equiv \frac{(T_{SCC} - T_{STEAM}) - (T_{STACK} - T_{FEED})}{\ln \dfrac{T_{SCC} - T_{STEAM}}{T_{STACK} - T_{FEED}}}$$

The boiler overall heat transfer coefficient, U_{MEAN} $(\dot{M},T)$, varies with both temperature and flow rate. The magnitude of this coefficient is determined by noting that the resistance to heat transfer from the combustion gases is the dominant resistance and, thus, only gas properties have an appreciable effect on U_{MEAN}. For example, with a staggered tube configuration (Ref 8),

$$\text{Nusselt No.} \propto (\text{Reynolds No.})^{0.6} (\text{Prandtl No.})^{0.33}$$

Observing that the variation in the one-third power of the Prandtl number is negligible, and lumping the geometry into the constant of proportionality, $\bar{U}$,*

$$U_{MEAN} = \bar{U} \ k_{AVG}(T_{AVG}) \left[\frac{\dot{M}_{SCC} + \dot{M}_{AIR,LEAK}}{\mu_{AVG}(T_{AVG})}\right]^{0.6} \qquad (C\text{-}10)$$

where

$$T_{AVG} \equiv (1/4) \ (T_{SCC} + T_{STACK} + T_{FEED} + T_{STEAM})$$

$k_{AVG}(T_{AVG})$ = thermal conductivity of combustion products at the average temperature T_{AVG}

$\mu_{AVG}(T_{AVG})$ = viscosity of combustion products at temperature T_{AVG}

$$k_{AVG}, \mu_{AVG} \quad \propto \quad \frac{T_{AVG}^{1.5}}{225 + T_{AVG}} \qquad (C\text{-}11)$$

Equations C-11 are usually referred to as the Eucken equations and were derived using the methods of the kinetic theory (Ref 9). For this simulation, the constants of proportionality were determined by assuming that the products of combustion behave in the same manner as air.

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*$\bar{U}$ was back calculated from boiler performance data summarized on the nameplate.

Equations C-7 through C-11 are solved simultaneously, using the techniques described previously.

Accuracy of the Simulation

There is no way of conclusively evaluating the HRI simulation until (and unless) airflows are measured.

The model should be good at simulating HRI operation in an excess air mode. Problems faced in representing incinerator operation with excess combustion air are problems in heat transfer and thermodynamics; the simulation is straightforward, albeit complicated.

Combustion chamber temperatures have been measured. An indication of model accuracy in predicting excess air operation may be acquired by back-calculating from applicable measured SCC temperatures to get the total energy input to the combustion products and then subtracting the enthalpy of the total 460 lb/min of combustion air to get the average Btu content of the waste feed. Using this feed rate, primary combustion chamber temperatures can be predicted and compared with measured values. Figure C4 was developed in this manner. The measured temperatures were recorded during the acceptance tests of the NAS JAX heat recovery incinerators (Ref 10).

Heat transfer and ash losses have not been considered, and thus there is some error associated with this approach. Regardless, Figure C4 shows predicted and measured PCC temperatures to coincide if nearly the entire capacity of the forced draft blower was being used for underfire air when the temperatures were recorded, a condition that is possible, even probable.

With only carbon monoxide introduced, the model is very suspect in its ability to simulate incomplete combustion. Other products of pyrolysis of waste, such as hydrogen, methane, and perhaps some higher hydrocarbons (Ref 1), will certainly have to be included in any sophisticated simulation. Carbon monoxide was selected for this preliminary analysis because it has the highest activation energy for combustion of the more common products of pyrolysis and would tend to be the last product consumed. The simulation of HRI operation with airflows only slightly less than stoichiometric should, therefore, be adequate.
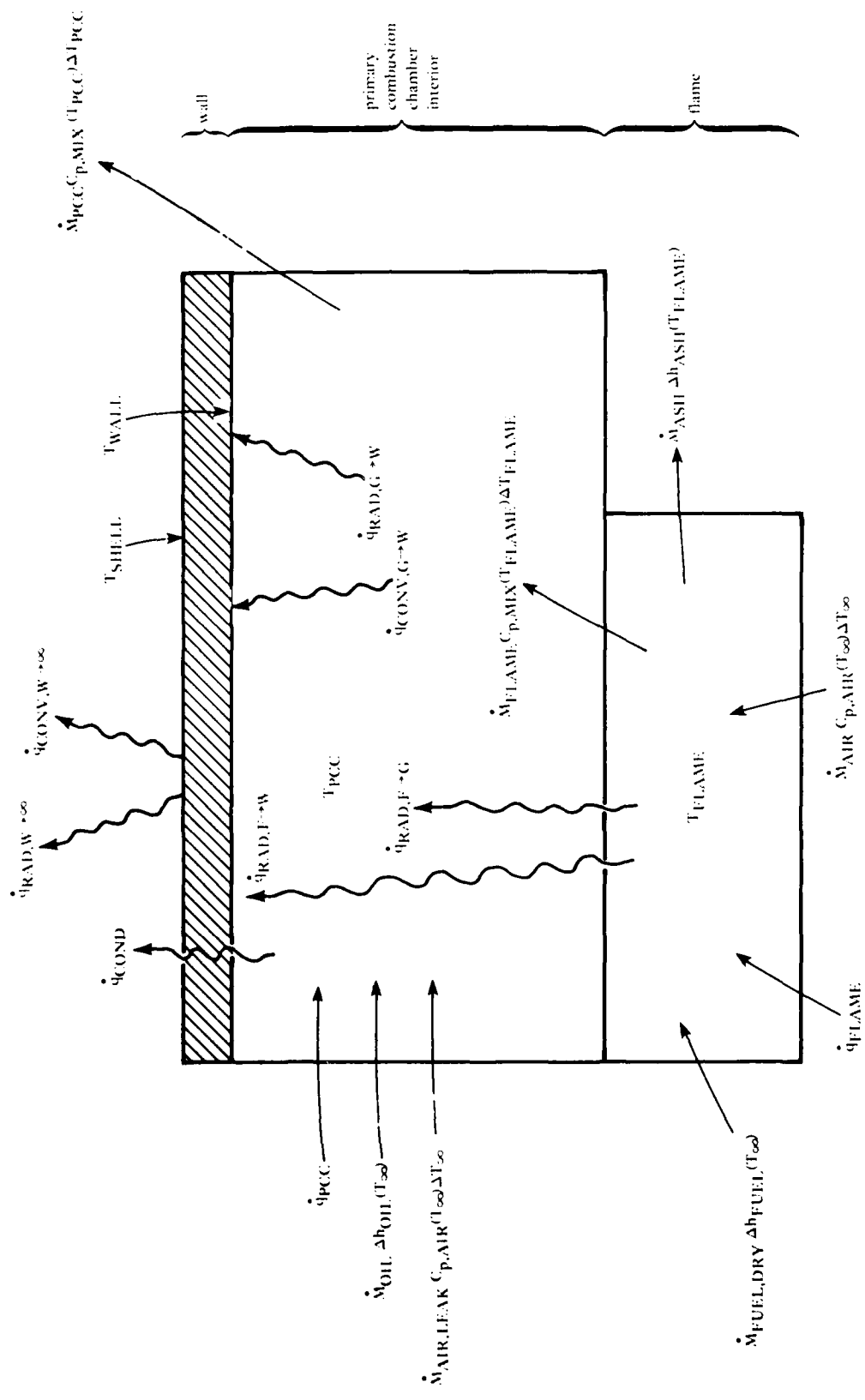
C-11

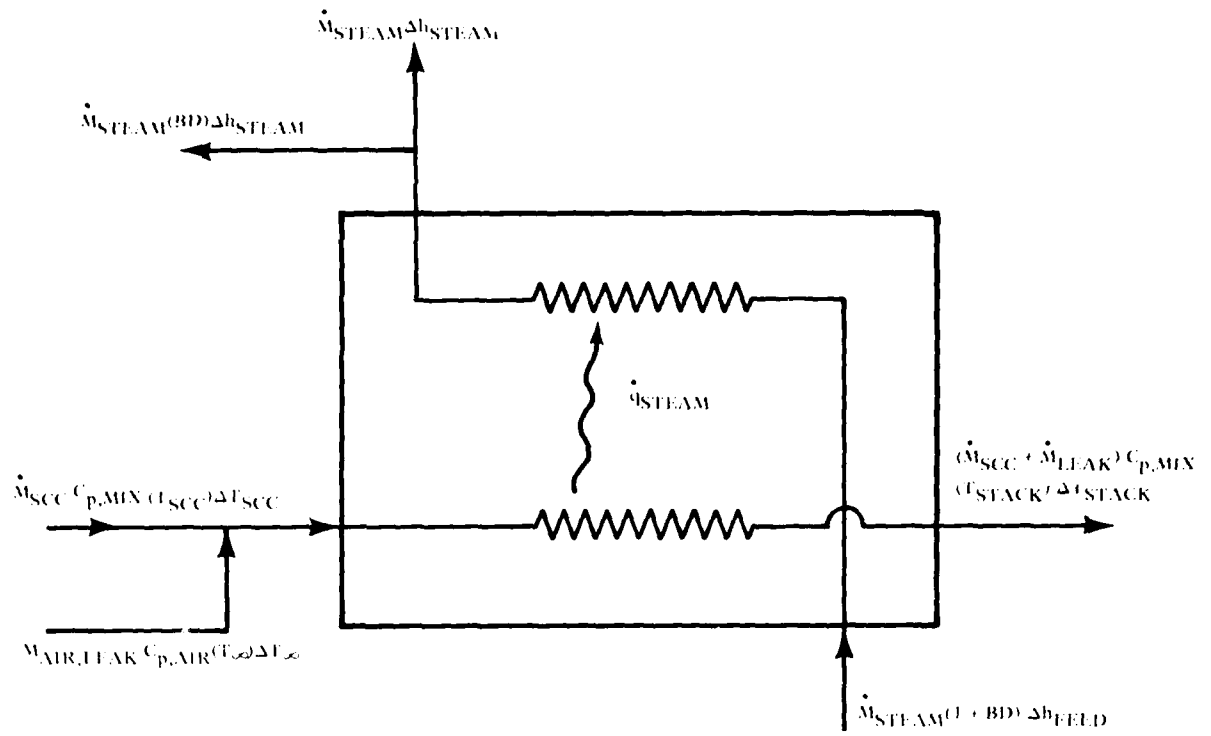Figure C-1. Conservation of energy in primary combustion chamber.

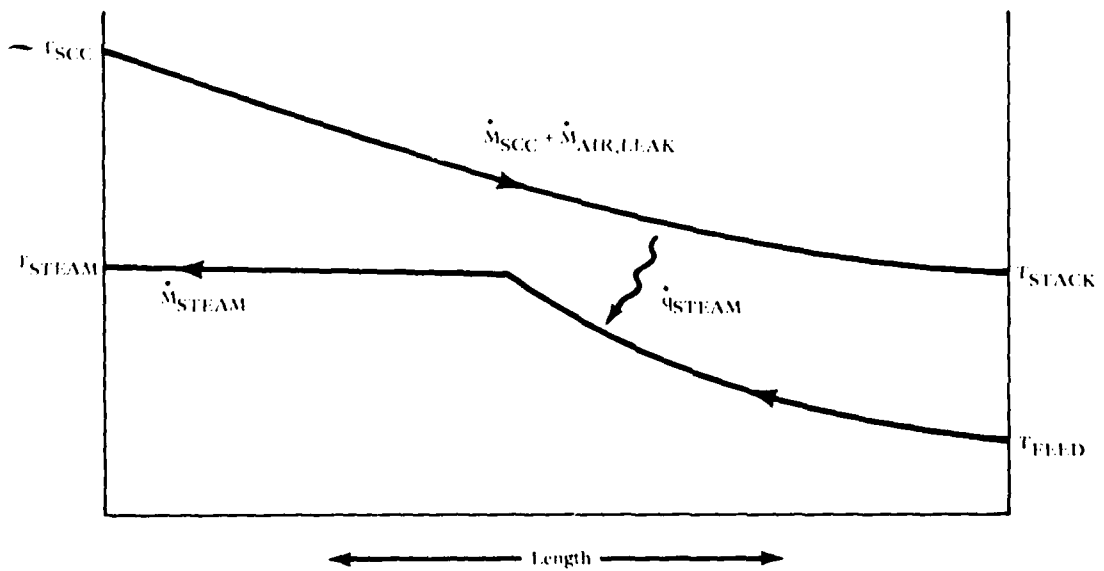Figure C-2. Conservation of energy in heat recovery boiler.

Figure C-3. Typical temperature variations through heat recovery boiler.

Notes:
(1) Feed rate = $0.152(10)^8$ Btu/hr
(2) Temps measured while operating with excess air during acceptance tests (Ref 10)

predicted PCC temperature

capacity of air blowers

measured PCC temperature

measured SCC temperature

predicted SCC temperature

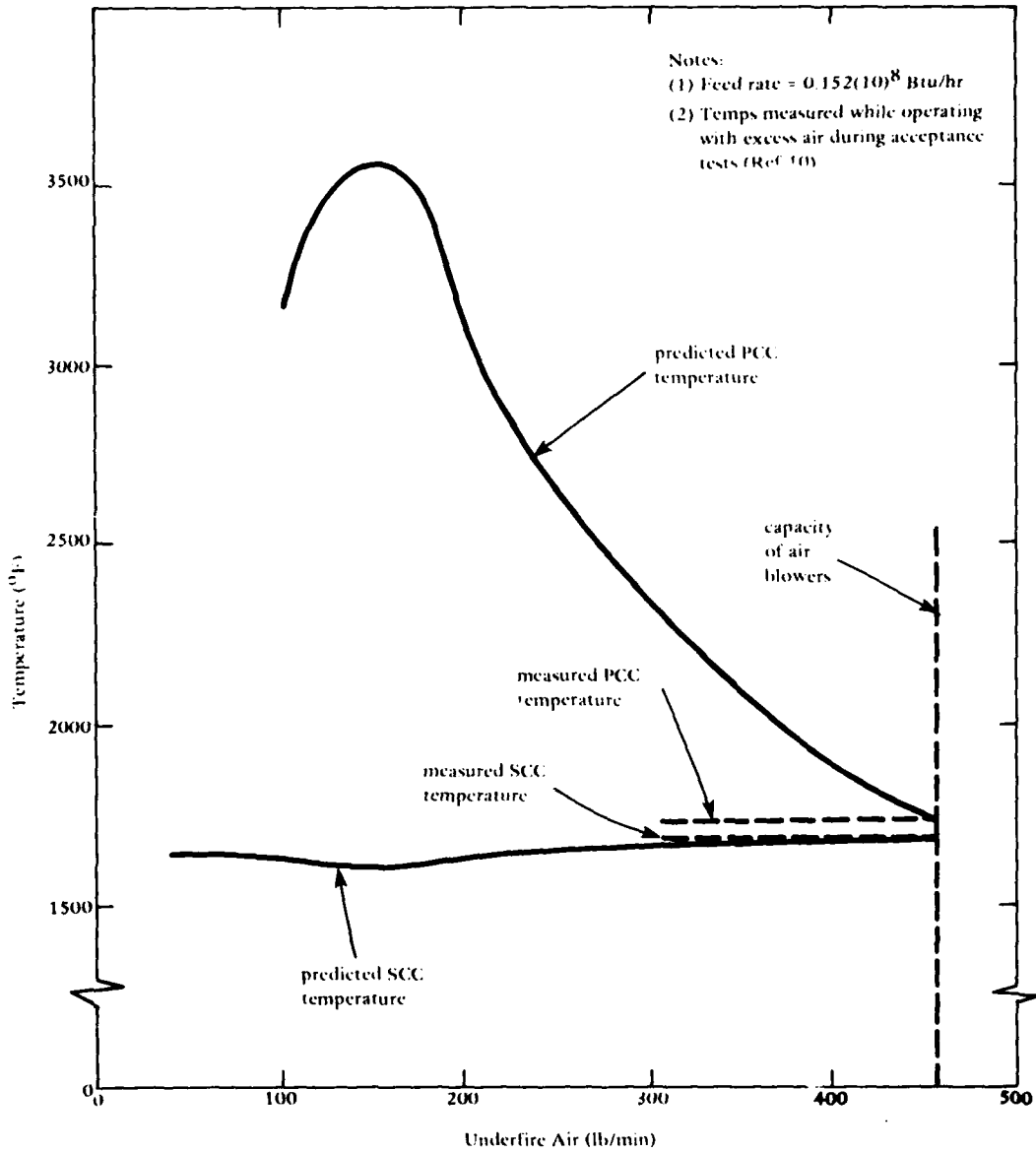Temperature (°F)

Underfire Air (lb/min)

Figure C-4. Evaluation of mathematical simulation of NAS Jacksonville HRI operating with excess combustion air.

C-14

Appendix D

EFFICIENCY CRITERIA USED TO DEFINE PERFORMANCE
OF HEAT RECOVERY INCINERATORS

The performance of the heat recovery incinerator was evaluated by applying the heat loss method suggested for steam generating units by the American Society of Mechanical Engineers (Ref 5). "The efficiency is equal to 100% minus a quotient expressed in percent. The quotient is made up of the sum of all accountable losses as the numerator, and heat in the fuel plus heat credits as the denominator." Or, in mathematical form, Equation 2,

$$\eta \equiv 1 - \frac{\Sigma \ LOSSES}{\Sigma \ INPUT}$$

Not all losses are included in the summations, and some are slightly different from those suggested by Reference 5 in order to be compatible with the mathematical simulation.

Losses

Heat lost vaporizing moisture with waste $= \dot{M}_{MOIST} \ \Delta h_{fg}$

Vaporization of water generated by burning hydrogen in waste

$$= \ 18 \ n_2 \ \Delta h_{fg} \ \dot{M}_{FUEL}$$

Carbon carried out with ash $= - \ \dot{M}_{ASH} \ C_{ASH} \ Q_{CO_2}$

Sensible heat in ash $= \ C_{P,ASH} \ \dot{M}_{ASH} \ \Delta T_{FLAME}$

Heat transfer through walls of PCC $= \ KA_{PCC}(T_{PCC,WALLS} - T_{PCC,SHELL})$

Heat transfer through walls of SCC $= \ KA_{SCC}(T_{SCC,WALLS} - T_{SCC,SHELL})$

Carbon monoxide in stack gases $= - \ n_9 \ \dot{M}_{FUEL} \ (Q_{CO_2} - Q_{CO})$

Sensible heat in stack gases

$$= (\dot{M}_{SCC} + \dot{M}_{AIR,LEAK})\ C_{P,MIX}(T_{STACK})\ \Delta T_{STACK}$$

Loss of steam due to blowdown $= \dot{M}_{STEAM}\ [BD/(1 - BD)]\ \Delta h_{STEAM}(T_{STEAM})$

Inputs

Chemical plus sensible energy in waste

$$= \dot{M}_{FUEL}\ HHV_{FUEL,DRY} - \dot{M}_{AIR}\ C_{P,AIR}(T_\infty)\ \Delta T_\infty$$

$$+ \left(\dot{M}_{SCC} + \dot{M}_{AIR\ LEAK}\right)\ C_{P,AIR}(T_\infty)\ \Delta T_\infty$$

Enthalpy of combustion air $= \dot{M}_{AIR}\ C_{P,AIR}(T_\infty)\ \Delta T_\infty$

Chemical plus sensible energy in oil $\cong \dot{M}_{OIL}\ HHV_{OIL}$

Enthalpy of boiler feed water $= \dot{M}_{STEAM}(1 + BD)\ \Delta h_{FEED}(T_{FEED})$

Sensible heat of products of combustion entering boiler

$$= (\dot{M}_{SCC} + \dot{M}_{AIR\ LEAK})\ C_{P,MIX}(T_{SCC})\ \Delta T_{SCC}$$

The power required to run accessories is input directly.

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