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NOISE GENERATED BY
TURBULENT NON-PREMIXED FLAMES

DISSERTATION

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the Degree Doctor of Philosophy in the Graduate
School of The Ohio State University

By

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* * * * *

The Ohio State University
1969

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SYMBOLS

Symbol

a	Average radius of a uniformly contracting and expanding sphere
b	Breadth of thickened jet boundary layer
c	Speed of sound in air at standard condition
c_1	Degree of turbulence constant for a submerged jet
CF	Convection factor, $(1 - M_c(\cos \hat{\Theta}))^{-2}$
D	Inside diameter burner tubes
f	Frequency of emitted sound
f_c	Center frequency, see footnote on page 31
ΔH_c	Heat of Combustion
$\Delta \hat{H}_c$	Heat of Combustion per unit mass, $\Delta H_c / m$
I.D.	Inside Diameter
I_0	Reference sound intensity at 1000 hz at the threshold of hearing for the average human ear.
I_r	Acoustic intensity for a simple source,
l	Eddy mixing length
m	Molecular mass
M	Flame mach number, U_b / c

Symbol

M_c	Convective Mach number, $ u_{ox} - u_f /2c$
$\dot{M}_f$	Mean rate of fuel consumed
m	Mass, velocity ratio, u_2/u_1
$\dot{m}$	Mass flow rate
p	Acoustic pressure
p_o	Reference sound pressure at 1000 hz at the threshold of hearing for the average human ear
p_{rms}	Measured root-mean-square pressure amplitude
R	Radius of soap-bubble, see Figure 4
Re	Pipe Reynolds number, $\rho u D/\mu$
r	Radial distance
SPL	Sound-pressure level, $20 \log_{10} \frac{p_{rms}}{p_o}$, see footnote on page 31
SIL	Sound-intensity level, $10 \log_{10} \frac{I_{rms}}{I_o}$
s	Radial distance to flame surface of a premixed combustible gas, see Figure 4
t	Time
U	Average speed, convective speed, $ u_{ox} - u_f /2$
U_b	Laminar flame speed
u	Streamwise velocity of jet,
u'	Streamwise perturbation velocity in a jet boundary layer

Symbol

$\bar{u}$	Mean mass velocity, $\int \bar{u}^2 dy / \int \bar{u} dy$
$(uA)_f$	Fuel volume flow rate
V	Volume of combustion zone, characteristic velocity used in Strouhal number
ΔV	Volume element for a source type combustion process
v'	Transverse perturbation velocity in a jet boundary layer
w	Acoustic energy for a simple source, $\overline{ \dot{p} ^2}/\rho c^2$
x	Streamwise direction shown in Figure 6
y	Transverse direction shown in Figure 6
α	Mixture ratio,
$\hat{\alpha}$	Strouhal number, $f D/V$
ϵ	Expansion ration, $\frac{\text{volume of burnt gases}}{\text{volume of unburnt gases}}$
$\hat{\epsilon}$	Degree of turbulence in the jet mixing region, $\frac{db}{dx}$ (see Figure 5)
λ	Wave length of acoustical radiation
η	Acoustic combustion efficiency, $\pi/\dot{m}_f \Delta h_c$
θ	Azimuth angle for microphone position
$\hat{\theta}$	Acute angle to streamwise flow direction, $180-\theta$
μ	Viscosity coefficient
π	Acoustic power, $4\pi r^2 \overline{p} ^2/\rho c$

Symbol

ρ	Density at standard conditions
$\bar{\rho}$	Density ratio, ρ_1/ρ_2
ϕ	Co-latitude angle for microphone position
$\hat{\phi}$	Equivalence ratio, $(\dot{m}_f/\dot{m}_{ox})/(\dot{m}_f/\dot{m}_{ox})_{\text{STOICH}}$
χ	Impingement angle, see Figure 43

Subscripts

Air	Air
f	Fuel
o	Initial
O ₂	Oxygen
ox	Oxidizer
STOICH	Stoichiometric

Superscripts

Air	Air
O ₂	Oxygen
ox	Oxidizer
v	Vertical component
-	Time average
NPM	Non-Premixed
PM	Premixed

CHAPTER I

INTRODUCTION

Numerous investigations have dealt with the various acoustical phenomena associated with flames. The best known examples are those of the "singing flame." Wood¹ gives this account: "If a small jet of inflammable gas is ignited and inserted into a wider tube open at both ends, then under certain conditions a steady and intense tone is produced having the natural frequency of the wider tube. The first person who observed a singing flame may have been Dr. Bryan Higgins in 1777."

The singing flame is a resonant phenomena of combustion noise. Along with it rank the so-called regular combustion oscillations which are divided into two main groups: 1) those in which the acoustics of the system determine the frequency, for example, "screaming" of rocket engines and 2) those whose frequency is fixed by the combustion or fuel-feed processes like the "chugging" of liquid propellant rocket engines. Each group comprises a set of discrete frequencies: the first group is

characterized by high frequencies, whereas the second group occurs at low frequencies. In most cases these sound waves are extremely intense and cause pain to the human ear.

More recently, Bragg² has suggested investigation of the noise of non-resonant turbulent flames, that is, the irregular and unrepitative sound or true noise. This class of noise is typified by jet propelled aircraft. The energy radiated from these jets will become more intolerable with advances in more powerful jet engines. In his paper, Bragg presented an analysis based on the concept of a simple monopole source from which the order of magnitude of the noise could be estimated. Shortly afterward, noise generation of premixed open turbulent flames were studied³. In more recent times, researchers in England^{4,5} have become interested in the noise of premixed open flames. Here in the United States^{6,7,8}, studies of open turbulent fuel flames with applications to industrial burners have been made and very recently⁹ a paper has been published on optical studies of the generation of sound by open diffusion flames and open liquid-fuel spray combustion.

The object of this investigation is to contribute to the understanding of the noise generated by non-resonant flames. In particular, attempts have been made to determine the acoustical energy released by turbulent non-premixed flames. Furthermore, the mechanism of noise generation, its directionality and frequency spectrum will be related to the geometry, flow and chemistry of the combustion systems considered.

CHAPTER II

BACKGROUND

In order to understand and interpret the noise generated by non-premixed and premixed turbulent flames, the basic terms used in describing acoustic phenomena in jets will be set down.

A. Definition of Acoustical Terms

By definition the sound-pressure-level is:

$$SPL = 20 \log_{10} \frac{p_{rms}}{p_0} \quad [db]$$

where p_{rms} is the measured root-mean-square pressure amplitude and p_0 is the reference level. The reference level being 0.0002 dyne/cm^2 . The sound intensity level is:

$$SIL = 10 \log_{10} \frac{I_{rms}}{I_0} \quad [db]$$

where I_{rms} is the root-mean square sound intensity and I_0 the reference level taken as $10^{-9} \text{ erg/cm}^2\text{-sec}$. The decibel (abbreviated "db") represents a relative quantity; either relative to P_0 or I_0 for SPL or SIL.

The response of the ear¹⁰ is shown diagrammatically in Figure 1.

Figure 1 - Contours of equal loudness for the threshold of hearing and threshold of pain, for intensity level and frequency for the average ear.

The bottom curve, for zero loudness is the threshold of hearing, below which sound of that frequency is inaudible to the average ear. The upper contour is the threshold of pain, above which the feeling is more of pain than of sound

(and the result is usually damage to the ear). The reference pressure and intensity referred to earlier correspond to the values of p_{rms} and I_{rms} at 1000 Hertz at the threshold of hearing.

Some typical SPL are: 135 db for a F-84 at take-off (80 ft. from tail), 90 db inside a motor bus, 63 db for conversational speech. A typical SPL for the flames considered here is about 100 db at a distance of 1 m from and downstream of the issuing combustible gases.

B. Basic Acoustical Wave Theory

The three-dimensional equation of acoustic-wave* motion, which is valid for small amplitude waves and for a medium which is assumed to be inviscid and thermally non-conducting, is:

$$\nabla^2 p = \frac{1}{c^2} \frac{\partial^2 p}{\partial t^2} \quad (1)$$

where p is called the acoustic (or sound) pressure and is a function of time " t " and position " r "; and c is the speed of sound of the medium.

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*For details see Morse and Ingard, Theoretical Acoustics. McGraw-Hill, 1968, Chapter 6.

1. The Simple Source

As follows from equation (1), the wave equation for a uniformly contracting and expanding sphere (a monopole source), in spherical polar coordinates is:

$$\frac{1}{r^2} \frac{\partial}{\partial r} \left(r^2 \frac{\partial p}{\partial r} \right) = \frac{1}{c^2} \frac{\partial^2 p}{\partial t^2}$$

The general solution of this equation (except at $r=0$) is:

$$p = \frac{1}{r} F(r-ct) + \frac{1}{r} f(r+ct)$$

a wave going outward from the center and another going toward the center.

Consider a uniformly contracting and expanding sphere of average radius, a , such that the radial velocity U_r of its surface is

everywhere the same $U(t)$ (see Figure 2).

Fig. 2 - Uniformly contracting and expanding sphere of average radius "a".

Thus, $U(r) = U(t)$, and the momentum equation for an inviscid thermally non-conducting fluid to first order in p and U_r

is:

$$\rho \frac{\partial U_r}{\partial t} = - \frac{\partial p}{\partial r} \quad (2)$$

The volume flow rate of air away from the sphere is:

$$\frac{\partial V}{\partial t} = 4\pi a^2 u(t)$$

Therefore, (2) becomes:

$$\frac{\rho}{4\pi a^2} \frac{\partial}{\partial t} \left(\frac{\partial V}{\partial t} \right) = -\frac{\partial p}{\partial t} \quad (3)$$

If the vibrating sphere is very small (or if $a \ll \lambda$, where λ is the wavelength of the sound radiated), the sphere is called a simple source and the pressure wave radiating outwards at a distance "r" from the simple source is:

$$p = \frac{\rho}{4\pi r} \frac{d}{dt} \left(\frac{dV}{dt} \right) \quad (4)$$

At distance of $r > \lambda$ * the farfield acoustical energy, w, intensity, I_r , and power, Π , for a simple source become:

$$w = |\bar{p}|^2 / \rho c^2 \quad (5)$$

$$I_r = |\bar{p}|^2 / \rho c \quad (6)$$

$$\Pi = 4\pi r^2 I_r \quad (7)$$

---

*See Chapter 7, p. 311 of Morse and Ingard. Theoretical Acoustics. McGraw-Hill, 1968.

2. Remark on Interpretation of Complex Solutions

The solution to the wave equation $\nabla^2 p = \frac{1}{c^2} \frac{\partial^2 p}{\partial t^2}$ is usually given as a complex function such as $\sum A_n e^{-i\omega_n t}$. The convention used is to express physical quantities in terms of complex numbers, considering energy and other quadratic functions of the variables as the basic physical quantities and using their mean values averaged over time*. Thus, the expressions for acoustical energy, intensity and power are referred to the time average value of $|p|^2$ and where ever the lone p is used, it is to be interpreted as the root-mean-square amplitude.

C. Source Noise of Non-Resonant Combustion

1. Non-Premixed Flames

Bragg² has postulated that combustion noise can be considered as that of a monopole simple source, i.e., the pressure waves are radiated by isolated elements of gaseous fuel-oxidizer mixture as they ignite, burn and expand. Consider a small mass element of fuel m_f which burns at a constant pressure with a mass element of air αm_f , where α is the ratio of the mass of air to mass

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* See pp.9 and 251 of Morse and Ingard Theoretical Acoustics. McGraw-Hill, 1968.

of fuel. Upon burning, the products of combustion form ϵ times the initial volume of the unburnt gas and the surrounding medium is then pushed away. For hydrocarbon fuels this displaced mass is $(\epsilon-1)\alpha m_f$ (See Figure 3).

Figure 3 - Pictorial presentation of elementary constant pressure combustion.

The displaced mass, $(\epsilon-1)\alpha m_f$, manifests itself by radiating a pressure wave moving away from its source at the start of the burning process, and an expansion wave, equilibrating the mass movement at the end of burning (the effective pulsating sphere). From equation

(7), the power radiated from a simple monopole source is:

$$\begin{aligned}\overline{\Pi} &= 4\pi r^2 |\dot{p}|^2 / \rho c \\ &= |\overline{4\pi r \dot{p}}|^2 / 4\pi \rho c\end{aligned}$$

Further, with equation (4)

$$\overline{\Pi} = |\overline{4\pi r (\rho/4\pi r) \frac{d}{dt} (\frac{dV}{dt})}|^2 / 4\pi \rho c$$

Thus,

$$\overline{\Pi} = |\overline{\rho \frac{d}{dt} (\frac{dV}{dt})}|^2 / 4\pi \rho c \quad (8)$$

But the mass flow rate away from the source is $(\epsilon-1)\alpha \dot{m}_f$, which is equivalent to $\rho \frac{dV}{dt}$ for the simple source given in section 1, page 8. Thus,

$$\overline{\Pi} = |\overline{\frac{d}{dt} [(\epsilon-1)\alpha \dot{m}_f]}|^2 / 4\pi \rho c \quad (9)$$

Hence, the acoustical power radiated by a simple source is related to the time rate of change of the rate of fuel consumption. In a steady laminar flame the fuel mass flow rate is constant, and hence, the acoustical power radiated will be zero. For a turbulent flame, although the total fuel consumption is constant, the local

rate of fuel consumption fluctuates with time, and the acoustical power radiated for small volume elements are not zero ($\overline{p} \neq 0$).

From equation (4), the acoustical pressure for non-premixed flames may be written as:

$$p = \frac{(\epsilon - 1)}{4\pi r} \frac{d}{dt} (\alpha \dot{m}_f) \quad (10)$$

2. Premixed Flames

For a constant-pressure burning of a premixed combustible gas a spherical combustion wave within a soap bubble (See Figure 4),

Figure 4 - Pictorial representation of constant-pressure combustion of a premixed combustible gas expanding in a soap-bubble.

the work done by the burnt gas is transferred to the unburnt gas. The volume of burnt gas is $\frac{4}{3}\pi s^3$; using the definition for ϵ , the volume of the unburnt gas is $\frac{4}{3}\pi s^3/\epsilon$. The increase in volume upon burning is $\frac{4}{3}\pi s^3(1-1/\epsilon)$, and the rate of increase of volume is $\frac{dV}{dt} = (1-1/\epsilon)4\pi s^2 \frac{ds}{dt}$. Let U_b be the flame speed. Then, in time dt an amount of fresh gas $4\pi s^2 U_b dt$ is consumed. That is, in time, dt , a gas layer $(U_b dt)$ of combustible gas is consumed, but the actual flame front moves a distance $ds > U_b dt$ because of the expansion of the combustible products. Now the amount of gas in the spherical shell s and $s+ds$ after burning is $4\pi s^2 ds$, and the corresponding amount of unburnt gas is $4\pi s^2 ds/\epsilon$. This latter amount is the actual amount consumed by the flame. Hence, $4\pi s^2 U_b dt = 4\pi s^2 ds/\epsilon$. Thus, $\frac{ds}{dt} = \epsilon U_b$, a constant for the mixture. Thomas and Williams⁴ have demonstrated that this type of combustion corresponds to the simplified monopole radiating source. Then the acoustical pressure can be written as:

$$p = \frac{\rho}{4\pi r} \frac{d}{dt} \left[(1-1/\epsilon) 4\pi s^2 \frac{ds}{dt} \right]$$

$$= 2 \rho \epsilon (\epsilon-1) (s/r) U_b^2 \quad (11)$$

From (7) the power radiated becomes:

$$\begin{aligned} \overline{P} &= \overline{|4\pi r [2s\epsilon(\epsilon-1)^{3/2} U_b^2]^2|} / 4\pi s\epsilon \\ &= \overline{|8\pi s\epsilon(\epsilon-1)^{3/2} U_b^2|^2} / 4\pi s\epsilon \end{aligned} \quad (12)$$

3. Efficiency of Combustion Noise

For combustion in a turbulent burning zone, Bragg² defined the efficiency of combustion noise as the ratio of acoustical power to the energy released by the fuel per sec. His analysis is as follows: for an elemental volume, ΔV , in a turbulent burning zone, consider that its dimensions are of the same order as the thickness of a laminar flame, such that the conditions at any instant can be considered uniform for the whole of ΔV . Based on the "wrinkled laminar flame"^{12,13} concept for the turbulent flame, the rate of combustion in ΔV may vary from zero to that of a laminar flame. Let $\dot{M}_f$ represent the mean rate of fuel consumption per unit volume of the flame. Let $\bar{U}$ be the velocity at which the flame passes through ΔV . Hence, the mean rate of change of burning rate must be on the order of $\pm \dot{M}_f \frac{\Delta V \bar{U}}{d}$ where d is the flame thickness. The proportion of time such a rate is maintained is $\dot{m}_f / \sqrt{\dot{M}_f}$, the ratio

of the mean-burning rate in the combustion zone to that burning rate in a laminar flame where $\dot{m}_f$ is the total fuel supply rate and V the total volume of the zone.

Recalling the expression for the acoustical power radiated, $\overline{\Pi} = \left| \frac{d}{dt} \left(\xi \frac{dV}{dt} \right) \right|^2 / 4\pi \rho c$ where the expression in the brackets corresponds to the time average rate of rate fuel consumption (recalling that $\xi \frac{dV}{dt} \propto \dot{m}_f$). Thus, the power for $V/\Delta V$ elements in the combustion zone becomes:

$$\overline{\Pi} = \frac{(\xi-1)^2 \alpha^2}{4\pi \rho c} \left| \dot{M}_f \Delta V U/d \right| \frac{\dot{m}_f}{V \dot{M}_f} \frac{V}{\Delta V} \quad (13)$$

The energy released by the fuel is $\dot{m}_f \Delta h_c$ where Δh_c is the heat of combustion.

Then the efficiency to produce of combustion noise may be defined as the ratio $\eta = \overline{\Pi} / \dot{m}_f \Delta h_c$

Thus,

$$\eta = \frac{(\xi-1)^2 \alpha^2}{4\pi} \left(\propto \frac{\dot{M}_f}{\rho c} d \right) \left(\frac{\Delta V}{d^3} \right) \left(\frac{U^2}{\Delta h_c} \right) \quad (14)$$

By assumption $d^3 \sim \Delta V$, therefore, $\Delta V/d^3 \sim 1$. The product $\alpha \dot{M}_f d$ corresponds to the rate of air consumption per unit cross-section in a laminar flame, and is, therefore, equal to $S U_b$ where U_b is the laminar flame speed. Thus, $\frac{\alpha \dot{M}_f d}{\rho c} = \frac{S U_b}{\rho c} = \frac{U_b}{c} \equiv M$ the flame Mach number. For a turbulent combustion process occurring as a result of the mixing of two streams moving at different speeds, a reasonable estimate for U would be one half the difference between these speeds, $U = \frac{1}{2} |U_{ox} - U_f|$ where U_f and U_{ox} correspond to the speeds of the fuel and oxidizer streams, respectively. Therefore,

$$\eta = \frac{(\xi-1)^2}{16\pi} M |U_{ox} - U_f|^2 / \Delta h_c \quad (15)$$

Thus, the above efficiency indicates that the noise can increase because of the reactivity of the fuel manifesting itself in the flame Mach number, M , and due to the increase in flow velocities or pressure losses represented by $|U_{ox} - U_f|^2$.

D. General Properties of Turbulent Jets

1. Degree of Turbulence

It is well known* that when a gas jet is expelled into a medium (at rest or moving), tangential separation is experienced by parameters such as velocity, temperature, and specie concentration. The instability of such a tangential separation region causes eddies which move at random both transverse to, and along with the flowing stream in the turbulent (laminar) jet boundary layer. A simplified diagram of a jet is shown in Figure 5. When a jet is spreading through a medium at rest it is called a submerged jet.

In the theory and experimental measurements of turbulent jets, the spreading of the jet becomes an important parameter. It can be estimated in the following way: A region in the mixing zone of a turbulent

Figure 5 - Simplified diagram of a co-flowing jet.

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*G. N. Abramovich, The Theory of Turbulent Jets, M.I.T. Press Cambridge Massachusetts, 1963, Chapter 1.

jet boundary layer may be pictured as shown in Figure 6. When an eddy moves in the transverse direction, it reaches a layer whose mean velocity differs from its original layer; the change in mean velocities is:

$$\Delta \bar{u} = l \frac{\partial \bar{u}}{\partial y}$$

This velocity in turn is equivalent to a perturbation velocity u' ; hence,

$$u' \sim l \frac{\partial \bar{u}}{\partial y}$$

Using the assumptions that the transverse velocity perturbation is proportional to the streamwise perturbation ($-v' \sim u' \sim l \frac{\partial \bar{u}}{\partial y}$), and Prandtl's assumptions that $l(y) =$

constant and $\frac{db}{dt} \sim v'$, where b is the breadth of the jet mixing zone, we obtain

$$\begin{aligned} \frac{db}{dt} &= \frac{db}{dx} \frac{dx}{dt} \\ &= \frac{db}{dx} \cdot u \end{aligned}$$

Figure 6 - A region in the mixing zone of a turbulent jet boundary layer with eddies distant by a mixing length ' $l(y)$ '

Hence from similarity in velocity profiles in the mixing region, and thus similarity in geometric* dimensions

$$\frac{db}{dx} = c_1 \frac{u_1 - u_2}{u}$$

Thus, defining the term "degree of turbulence":

$$\hat{\epsilon} = \left| \frac{db}{dx} \right| = c_1 \frac{|u_1 - u_2|}{|u|}$$

where u may be taken as the mean mass velocity:

$$\bar{u} = \frac{\int_1^2 \rho u dy}{\int_1^2 \rho dy}$$

For co-flowing jets one proposed model¹⁴ is:

$$u = \frac{\rho_1 u_1 + \rho_2 u_2}{\rho_1 + \rho_2}$$

Thus,

$$\hat{\epsilon} = c_1 \frac{1+\bar{\rho}}{2} \left(\frac{1-m}{1+\bar{\rho}m} \right)$$

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*G. N. Abramovich, The Theory of Turbulent Jets, M.I.T. Press Cambridge Massachusetts, 1963, Chapter I.

where

$$\bar{S} = S_2 / S_1$$

$$m = u_2 / u_1$$

$$C_1 = 0.27, \text{ based on submerged jet experiments}$$

Thus, for a co-flowing jet, $\hat{\epsilon}$ represents a fluid dynamic mixing parameter which takes into account turbulence, and velocity and density differences of the gas streams pouring into the mixing zone from the different sides of this region.

2. Convection and Refraction

In a jet, the sound emitters are moving with respect to a lab-fixed coordinate system. As the sound emitter moves downstream (the convection), the shape of the sound waves propagated are shown schematically in Figure 7.

For subsonic convection speeds the acoustic waves are somewhat crowded downstream and more widely spaced upstream. This state augments the acoustical intensity in directions making an

Figure 7 - A sound source moving subsonically and emitting acoustical pulses.

acute angle with the flow. For a monopole sound emitter, the convection factor which multiplies the sound intensities in the absence of convection is* :

$$CF = (1 - M_c \cos \theta)^{-2}$$

where

$$M_c \equiv \frac{|U_{ax} - U_s|}{\text{speed of sound}} \cdot \frac{1}{2}$$

It is also well-known^{15,16,17} that sound waves are refracted when passing through an interface between two media (see Figure 8). Ribner¹⁵ shows that it multiplies the stationary sound intensity by $(1 + \cos^4 \hat{\theta})$. Thus, the total directivity may be summarized as a simple product of two factors namely,

$$(1 + \cos^4 \hat{\theta}) CF$$

Hence, if the measured acoustical power is to be converted back to a stationary system, we have

$$\begin{aligned} \pi &= \frac{\pi_{exp}}{CF} \cdot \frac{1}{(1 + \cos^4 \hat{\theta})} \\ &= \pi_{exp} \frac{(1 - M_c \cos \hat{\theta})^2}{(1 + \cos^4 \hat{\theta})} \end{aligned}$$

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*Chapter 11 of Reference 11.

Figure 8 - Refraction, I_2 , and reflection, I_r , of sound intensity, I_1 , at the interface between medium a_1 and a_2 .

CHAPTER III

APPARATUS AND EXPERIMENTAL PROCEDURES

Discussed in this chapter will be the type of experiments performed for this dissertation, and the experimental apparatus used.

A. Type of Experiments

Two types of combustion processes were examined: 1) flames of co-flowing non-premixed turbulent gas streams both of which issued through concentric tubes (usually the fuel through the outer annulus and the oxidizer through the center tube), 2) flames of impinging non-premixed turbulent jets of fuel and oxidizer. In order to supplement the co-flowing non-premixed flame experiments, a few experiments were made with premixed flames.

To evaluate the experiments the following type of information was recorded: type of fuel and oxidizer, (to indicate the effect of reactivity on noise generation), the diameter of the tubes, and impingement angles (to indicate the mixing and stream speed effect on the noise),

the mass flow rates of fuel and oxidizer (to indicate the effects of degree of turbulence and Reynolds number on the noise), the angular position (to indicate directionality effects of the noise source).

The mixtures studied were: 1) for co-flowing jets: hydrogen - air, hydrogen - oxygen, ethylene - air, and ethylene - oxygen. 2) for impinging jets: hydrogen - air. 3) for premixed flames: hydrogen - air, hydrogen - oxygen.

B. Flame Burners

The arrangement for the concentric tubes is shown in Figure 9. The outer annulus is 1.835 cm I.D. and the length, 63.4 cm. Two burners were used for the co-flowing experiments, and both were made of stainless steel. One burner has an 0.813 cm I.D. inner copper tube (this burner was used for most experiments), and the other had a thick wall 0.391 cm I.D. inner copper tube diameter. In all experiments, the burner was placed in a horizontal position on a stand four foot high.

The arrangement for the impinging jets is shown in Figure 10. The plate shown was made so that two stainless steel cylindrical tubes could be secured at impingement angles of 45° , 90° and 135° .

Both tubes had the same inside diameters (either $D = 1.1303$ cm, 0.8079 cm, or 0.4981 cm). For the experiments with impinging jets the mounting plate shown in Figure 10 was welded to a stand such that the jet streams were issuing in directions inclined to the vertical and the point of impingement was about four and a half feet from the ground.

Figure 9 - Diagram of a typical concentric burner.

Figure 10 - Diagram of the impinging flame burner stand.

C. Flow System

A standard flow control system was used to regulate the flow rates of fuel and oxidizer simultaneously. This system consisted of two flat plate orifice-type flow meters, regulators, and control valves. By varying the driving pressure and orifice-hole diameter, a wide range of flow rates could be obtained. A diagram of the system is shown in Figure 11.

Figure 11 - General arrangement of apparatus

D. Sound Measurements

The sound emitted by the turbulent flames was measured with a General Radio type 1551-P1 condenser microphone in conjunction with a General Radio type 1551-A Sound-Level Meter set to the 20 kc weighting position. The frequency spectra were obtained with a Singer Sonic Panoramic Sound Wave Analyzer. The calibration frequency response curves when the condenser microphone is used in conjunction with the sound-level meter set at the 20 kc position is practically uniform over the audio frequency range of interest for perpendicular or 0° incidence.

Before and after each experiment the microphone was calibrated with a type 1552-B sound-level calibrator to insure accuracy of measurements. The overall sound-pressure levels were recorded from the sound-level meter, and a polaroid photograph was taken of the frequency spectrum displayed on the oscilloscope of the wave analyzer.

All experiments used in the analysis were taken outdoors. Reflection from the building walls contributed less one db* to the measured sound-pressure level. The configurations of the microphone position are shown in Figure 12.

Figure 12 - Configuration of microphone positions a) concentric burner b) impinging burner.

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*It was found that at points far from the reference point, the measured sound-pressure level was down by more than 8 db - indicating a less than 1 db contribution of sound. Handbook of Noise Measurements, General Radio Company 5th ed. p. 69.

For the co-flowing jet experiments, $\phi = 90^\circ$, and θ (referred to as the microphone position) is allowed to vary. The x, y plane is the horizontal plane, and the jets are issuing into the -x direction. For the impinging jet experiments, the microphone was placed at $\theta = 90^\circ$, $\phi = 180^\circ$. The vertical plane is the x, y plane and the jets are issuing at angles inclined to the -x direction. For both systems the radial distance, r, was 95.098 cm.

CHAPTER IV
DISCUSSION OF RESULTS

A. Co-Flowing Non-Premixed Turbulent Streams of
Combustible Gases

Unless otherwise stated, the results to be presented in this chapter apply to the case in which the oxidizer flows thru the center tube and the co-flowing fuel passes thru the outer tube. The dimensions of the burner are: 0.819 cm I.D. for inner tube, and 1.835 cm I.D. for the outer tube; the length of the burner was 66.0 cm.

Figure 13 - Concentric burner for co-flowing streams of gases.

Three aspects of the experiments will be discussed for each fuel-oxidizer system: 1) the noise level, 2) the directivity of the noise sources, and 3) the center frequencies of the noise sources.

Noise level* means the measured sound-pressure levels of the flames and/or unignited flowing jets, the influence of fluid dynamic variables, such as, turbulence and mass flow rates, and geometry (here meaning diameter effects). Directivity means the preferred direction of the acoustical intensity radiated. Center frequency** means the characteristic frequency estimated from photographs taken of the frequency spectra emitted by the flames being considered.

The typical order of magnitude of the noise levels encountered is given by a hydrogen-air flame with mass flows of $\dot{m}_{H_2}=0.4436$ grams/sec, $\dot{m}_{air}=19.1$

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*To measure sound-pressure level, the sound-level meter controls should be set for as uniform a frequency response as possible (Handbook of Noise Measurements, General Radio Company 5th ed. p. 10). For all the measurements discussed in this dissertation, the controls for the sound-level meter were set at the 20 kc position which gives the most uniform frequency response possible for the meter.

**In taking photographs of the frequency spectrum produced by the noise sources, it was found that the acoustically excited frequencies banded together in such a way as to suggest a dominant frequency. It was felt that this frequency may be characteristic of the combustion process and, thus, recorded and referred to as "center frequency" or f_c , in dimensions of hertz.

grams/sec. The acoustical energy released by such a flame is approximately 50 times greater than that emitted by the cold gas jets which means it is 17 db. higher. The center frequencies for the noise from the burning jets lie at about 3000 hertz, while that of the cold jets occur at around 6000 hertz. The over-all sound-pressure level is 103 db. for the combustion, whereas it is only about 86 db. for the cold jets. However, both were loud enough to be quite uncomfortable to the ear. The frequency range of maximum energy release lies in the frequency range where the ear is most sensitive (1000-4000 hertz). At a distance of about 1 meter, the acoustical power is, however, only 0.2 watts, corresponding to an acoustical pressure of 28 dynes/cm². It appears that the order of magnitude of the acoustical powers and pressures generated is low. However, in terms of the human ear, the noise produced by these flames is quite loud.

1. Hydrogen and Air Non-Premixed Flames

1.1 Noise Level of Hydrogen-Air Non-Premixed

Flames:--The effects of fuel and oxidizer flow rate variations on sound-pressure level are shown in Figure 14. The ordinate represents the sound-pressure level, while

the abscissa shows the equivalence ratio. The former indicates the noise level, the latter, the leanness or richness of the combustible reactants of hydrogen and air. The microphone was placed 95.098 cm from the burner port; all future data will be for this distance. Since nearly all discussions on the noise level will be based on graphs of this kind, the ordinate and abscissa will not be mentioned each time. In Figure 14, each curve represents the experimentally-measured combustion sound-pressure level produced for a fixed air mass flow rate while the hydrogen mass flow rate is varied as shown. The graphs show that with increased oxidizer flow rates, the noise level is greatly increased. It should be pointed out that for each individual curve the ratio of acoustic energy of the hot jets to that of the cold jets is 48 to 78. The curves also show a coalescence at the higher air mass flow rates. When the flow rate of air is increased these curves will probably level off or even come down. These data indicate that for increased Reynolds number, the combustion noise increases.

1.2 Directivity of Hydrogen-Air Non-Premixed

Flames:--Figure 15 indicates the effect of directivity of hydrogen-air flames on the noise level. The sound-pressure

level has a maximum at directions $\theta = 120^\circ$ to 125° for an equivalence ratio of 0.77 and a maximum SPL at a direction $\theta = 140^\circ$ for an equivalence ratio of 0.22. Figure 16 shows the stationary power as a function of microphone position taking into account refraction and monopole convection for equivalence ratios of 0.77 and 0.22:

$$\Pi_{\text{stationary}} = \Pi_{\text{exp}} \frac{(1 - M_c \cos \hat{\theta})^2}{(1 + \cos^4 \hat{\theta})}; \quad \hat{\theta} = 180 - \theta$$

Figures 17 and 18 show the stationary power and measured power versus equivalence ratio. Figures 16 and 17 indicate that the combustion noise does not act like a collection of monopole sources. If it did, Figure 16 would show two straight lines with zero slope (if sources were stationed at burner port) or curves with a positive slope (if the monopole sources were displaced some distance from the burner port). Figure 19 shows the root-mean-square acoustical pressure versus equivalence ratio for the same conditions of the above experiments.

1.3 Center Frequencies for Hydrogen-Air Non-Premixed Flames:--By means of a spectral wave analyzer, center frequencies can be estimated as shown in Plate I.

The ordinate is a log db and the abscissa is log f.

Plate I shows a typical frequency spectrum for hydrogen-air flames at an equivalence ratio of 0.77, a microphone position of 118° , and an air mass flow rate of 19.1 g/sec. The center frequency is at about 3000 hertz.

Plate II shows the corresponding cold jet spectrum. It should be noted that the voltage scale for this jet spectrum was 10 times as great as that of the hot jet.

If the same scale had been used, the spectrum would hardly be visible on the photograph. From here on, all reference to center frequencies will imply data taken from such photos. Figure 20 shows the center frequencies at various microphone positions. It is seen that there is a shift of the center frequency from 6000 hertz for a cold jet to 2000-3000 hertz for a $\hat{\phi} = 0.77$ non-premixed flame with $\dot{m}_{\text{air}} = 19.1$ grams/sec. For leaner mixtures, the center frequency lies around 4000-4500 hertz at $\hat{\phi} = 0.22$ and at the same oxidizer mass flow rate. The trend of the curves seems reasonable, that is, the leaner flames ($\hat{\phi} = 0.22$) produce frequencies which are closer to those of the cold jet while rich mixtures ($\hat{\phi} = 0.77$) produce lower frequencies. Hence, the combustion of hydrogen in air lowers the frequencies produced by the un-reacted mixture, and increases

its sound-pressure level considerably. Also shown in Figure 20 are typical center frequencies at lower oxidizer mass flow rates. The trend of these data is to lower the f_c with decrease in $\dot{m}_{air}$. Hence, it can be concluded that f_c depends on the equivalence ratio, the microphone position as well as the oxidizer mass flow rate.

The observation that the center frequency decreases with decrease in air mass flow rate may be explained by the laws governing frequencies of more commonly known acoustical phenomena. Two well-known examples are the frequency relationships for aeolian tones and edge tones*. Both of these acoustical phenomena obey the law that the frequency is directly proportional to a characteristic velocity, and inversely proportional to a characteristic dimension, i.e.,: $f = \hat{\alpha} \frac{V}{D}$ where $\hat{\alpha}$ is the proportionality constant called Strouhal number, and is 0.185 for aeolian tones and 0.055 for edge tones; V is the characteristic velocity;

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*See Reference 1, Chapter XV for definitions and discussions of these two phenomenon.

and, D is the characteristic dimension. Assuming a similar type relation holds for the experiments being considered here, V corresponds to the average air speed of the jets, D is either the fuel tube diameter or oxidizer tube diameter, and f_c the characteristic frequency. Then, as the air mass flow rate decreases, V decreases, and thus, the center frequency decreases. This is the same trend that is observed in Figure 20.

Figure 21 shows the calculated Strouhal numbers for hydrogen-air flames using f_c as the characteristic frequency, the average air velocity as the characteristic velocity, and the fuel tube diameter for the flames' noise, and air tube diameter for cold jet noise as the characteristic dimension. This figure shows that for hydrogen-air flames the Strouhal number is not a constant, but decreases linearly with increasing equivalence ratio. Although a non-constant Strouhal number is inappropriate for aeolian and edge tones, it is likely that it is correct for the flames considered here. As the equivalence ratio changes, the combustion process changes, and most likely the characteristic length is different from the geometric tube diameter chosen. Perhaps some characteristic mixing length for the turbulent combustion process

is the proper choice for D which varies as the turbulent mixing process varies.

2. Hydrogen and Oxygen Non-Premixed Flames

2.1 Noise Level of Hydrogen-Oxygen Non-Premixed Flames:--Figure 21 shows a comparison between flames of hydrogen-air and hydrogen-oxygen mixtures for low and high oxidizer mass flow rates. At low oxidizer flow rates ($\dot{m}_{\text{air}} = 4.77$ grams/sec, $\dot{m}_{\text{O}_2} = 5.34$ grams/sec) the pipe Reynolds number of the oxidizers is practically the same ($R_{\text{e}}^{\text{air}} = 0.4377 \times 10^5$; $R_{\text{e}}^{\text{O}_2} = 0.4433 \times 10^5$; $U_{\text{air}} = 7567$. cm/sec; $U_{\text{O}_2} = 7669$. cm/sec). However, the noise of the hydrogen-oxygen flames are several db lower than that of hydrogen-air flames. The degree of turbulence in the hydrogen-oxygen jet mixing region is lower than that of the hydrogen-air jet mixing region (See Figure 23). But, at the higher flow rates where the oxidizer Reynolds number and the speed of the oxidizers at the exit are nearly equal for both systems ($U_{\text{O}_2} = 30,380$ cm/sec, $R_{\text{e}}^{\text{O}_2} = 1.756 \times 10^5$; $U_{\text{air}} = 30,250$ cm/sec, $R_{\text{e}}^{\text{air}} = 1.750 \times 10^5$) and, the degree of turbulence is lower, the hydrogen-oxygen flames are considerably louder than the hydrogen-air flames. One explanation for this noted noise-level increase could be that the burning process is

becoming rougher, i.e., there are enlarged pockets of combustible mixtures burning and superimposing an added noise-level to the flames. This concept will be more completely examined later.

2.2 Directivity of Hydrogen-Oxygen Non-Premixed Flames:--Figure 24 shows the noise level of the hydrogen-oxygen flames for variations in microphone position. For an equivalence ratio of 0.16, the preferred direction for the radiated sound is 105° , while for the equivalence ratio of 0.068, it is about 110° ; so a shift in directionality occurs with change in equivalence ratio. As in the case of hydrogen-air flames, the fact that the acoustic stationary power decreases with increasing microphone position θ (see Figure 25) indicates a non-monopole source of acoustical radiation.

2.3 Center Frequencies for Hydrogen-Oxygen Non-Premixed Flames:--The frequency spectrum for hydrogen-oxygen flames is illustrated in Figure 26; plate III shows a representative frequency spectrum for flames of hydrogen-oxygen. For the richer mixtures Figure 26 shows that the center frequencies varied from 5000 to 3000 hertz. However, for leaner mixtures the center frequencies were lower (around 4000-3000 hertz). This

behavior is opposite to that found for hydrogen-air flames (see Figure 20). When the mixtures were rich as in the hydrogen-air flames at lower oxidizer flow rates, the center frequencies are lower and they shift to a lower frequency with increasing θ . As with the hydrogen-air flames, it is found that the center frequencies for hydrogen-oxygen flames are also a complicated function of equivalence ratio, microphone position, and oxygen mass flow rate for the co-flowing gaseous systems considered here.

3. Ethylene and Air/Oxygen Non-Premixed Flames.

As with the hydrogen, air and oxygen were used as oxidizers. The ethylene-air flames proved to be a highly unstable system, i.e., the system would remain burning only for restricted flow conditions; during data-taking there was an audible and constant non-periodic popping sound causing rapid fluctuations in meter readings, making data-taking difficult. Only after letting the system burn for a while did the sound smooth out for certain flow settings; this is presumably due to the burner port heating up and keeping the flame attached. It was also observed that a large amount of carbon deposits remained on the burner and burner stand

after experimentation. With the ethylene-oxygen flames, data taking was considerably easier and cleaner than with ethylene-air flames. Although noise level measurements were not difficult, it is still felt that there existed a certain amount of instability. The instability was not observed so much by what was heard with the ear, but what was recorded with the meter readings. At certain flow conditions, there existed meter fluctuations, although not nearly as large or as rapid as in ethylene-air flames.

3.1 Noise Level of Ethylene-Air/Oxygen Non-Premixed Flames:--The noise levels of ethylene-air flames and ethylene-oxygen flames are compared with the noise levels of hydrogen-air and hydrogen-oxygen flames in Figure 27. At the low turbulence levels ($Re^{ox} \sim 0.44 \times 10^5$) the hydrogen-oxygen flames produce the least noise of the four, next being that of ethylene-oxygen, while hydrogen-air and ethylene-air flames are the noisiest. The degree of turbulence in the mixing region is greatest, however, for ethylene-oxidizer mixtures. For higher Reynolds numbers, hydrogen-oxygen flames have the highest sound-pressure levels, but it should be noted that they are not much higher than that of the ethylene-oxygen flames whose Reynolds number for the oxygen flow is half

as large as that of the hydrogen-oxygen flames. Therefore, at the higher Reynolds number, the ethylene-oxygen flames should be of higher noise level.

3.2 Directivity of Ethylene-Air/Oxygen Non-Premixed Flames:--The directionality of the noise emitted by non-premixed ethylene-oxygen flames is shown in Figure 28. This figure shows that at an equivalence ratio of 0.34 the peak of the sound-pressure level occurs at a direction $\theta \sim 107^\circ$; and at $\hat{\phi} = 0.18$, $\theta \sim 95^\circ$. Once again, this shows a shift of angular preference with increase of fuel flow. But, in contrast to the hydrogen-oxygen flames (Figure 24) and the hydrogen-air flames (Figure 15), the leaner mixture for ethylene-oxygen flames has its peak sound-pressure level at a microphone position of a smaller θ rather than a larger θ .

The calculations of the acoustic stationary power, based on a monopole-source sound model are shown in Figure 29. This figure shows as in all non-premixed flames that the acoustical stationary power is not constant with microphone position, consequently, the noise sources from the flames do not seem to be of a monopole structure.

3.3 Center Frequencies for Ethylene-Air/Oxygen

Flames:--The center frequencies for ethylene-air flames and ethylene-oxygen flames are plotted in Figure 30; plate IV shows a typical frequency spectrum for these flames. For ethylene-oxygen flames at an oxygen mass flow rate of 10.18 grams/sec, the center frequency is always 6000 hertz. While for ethylene-air flames, the center frequency ranged between 600-800 hertz. For ethylene-oxygen flames at an oxygen mass flow rate of 5.24 grams/sec, the center frequency is 600 hertz. It appears that the center frequency is a complicated function of the equivalence ratio, oxygen mass flow rate, and microphone position. As shown in Figures 20 and 26, the trend that the center frequency changes greatly with oxidizer mass flow rate is observed for hydrogen-oxygen flames, but not for hydrogen-air flames. For some reason, this center frequency shift is limited only to fuel oxygen systems.

The considerably greater changes in center frequencies when the fuel is reacted with oxygen rather than air may be due to the fact the fuel-oxygen flames have considerably higher burning speeds than do flames of fuel-air. Based on the simple monopole source of noise

model for combustion, the frequency is inversely proportional to the laminar flame speed, i.e.,

$$\begin{aligned}\pi &\sim \overline{S_u A}^2 f^2 / \rho c \\ &\sim \text{constant } f^2 U_b^2\end{aligned}$$

and hence,

$$f \sim \frac{1}{U_b} \sqrt{\pi / \text{constant}}$$

This type of relation qualitatively agrees with observed center frequency data reported, and therefore, the concept of a monopole source of combustion noise is also supported.

4. Roles of Fuel and Oxidizer Flows Reversed for Flames of Non-Premixed Hydrogen-Air Co-Flowing Jets

A few experiments were performed with flames of turbulent non-premixed hydrogen-air where the roles of the fuel and oxidizer were reversed. Instead of flowing the air through the center tube and fuel through outer tube of the concentric burner, the fuel was passed through the center tube and the air was passed through the outer tube. These experiments resulted in a lifted flame which produced a noise level of 13.5 db greater than the

flame of the former experiments where the flame remained attached to the burner part.

Gaydon¹⁸ has mentioned that lifted flames from concentric burners act like a premixed flame. As the gases issue from the annuluses there exists an entrainment which eventually mixes the gases through the lifted non-burning region. Then the resulting mixture of gases burns. If the noise-level of the lifted flames is considered as a premixed flame, and the noise level from the non-lifted flame is considered as a non-premixed flame a basis of comparison results from the simple monopole models derived in Chapter II.

From equations 10 and 11 the acoustic pressure for a non-premixed flame is proportional to $(\epsilon-1)$, therefore, we write this proportionality as:

$$p^{NPM} \sim (\epsilon-1)$$

while for a premixed flame the acoustical pressure is proportional to $\epsilon(\epsilon-1)$, so that this proportionality is written as:

$$p^{PM} \sim \epsilon(\epsilon-1)$$

Therefore,

$$p^{PM} \sim \epsilon p^{NPM}$$

Based on this relation, a sound-pressure level for a premixed flame was calculated with the sound-pressure level measurement of the non-premixed flame mentioned earlier, and the calculated value was compared to the measured sound-pressure level of the lifted or premixed flame. The result was that the calculated noise level for the premixed flame was 15.07 db above the non-premixed flame. This is only 1.57 db greater than the difference between the measured premixed and non-premixed flames. Thus, the simple monopole model of combustion noise is supported.

5. Premixed Turbulent Flames

In order to supplement the data for the non-premixed flames, experiments with premixed flames were also performed. The burner was of the same construction as that for the above non-premixed experiments except the inner tube was removed near the base of the burner so that the fuel and oxidizer could mix before combustion. The mixtures examined were hydrogen-air and hydrogen-oxygen.

To prevent blow-off of the hydrogen-air flames, the ignitor had to stay on continuously. Since the ignitor consisted of a small spark, it did not contribute signi-

ificantly to the noise. With hydrogen-oxygen, only a few experiments with very lean mixtures were possible because richer mixtures would cause flash-back of the flame.

5.1 Noise Level of Premixed Turbulent Flames:--

Figure 31 shows a comparison of the noise levels of the premixed flames and those of non-premixed flames. While the premixed flames emit less noise than the non-premixed flames when the mixtures are lean, they are noisier for richer and near stoichiometric mixtures.

By using the relationship between non-premixed flames and premixed flames discussed in section 4.0, the trend of the data presented in Figure 31 can be verified. Based on the concept that the premixed acoustic pressure can be approximated by the expansion ratio times the non-premixed acoustic pressure, the sound-pressure level for the premixed flames are related to the non-premixed flames as:

$$(SPL)^{PM} = 20 \log_{10} \epsilon p^{NPM} / p_0$$

or

$$(SPL)^{PM} = (SPL)^{NPM} + 20 \log_{10} \epsilon$$

This relation is in qualitative agreement with the results shown in Figure 31. Based on this agreement, the simple monopole source as the model for combustion noise is supported.

5.2 Directivity of Premixed Turbulent Flames:--

The directionality of the emitted sound from the flames is

illustrated in Figure 32. The maximum of the acoustical energy seems to occur at angles somewhere between 90° to 110° for hydrogen-air flames, while for hydrogen-oxygen flames it is not so well defined; it probably lies between 90° - 104° .

The stationary acoustical power for hydrogen-air and hydrogen-oxygen flames is shown in Figure 33. As with the non-premixed cases, the acoustical stationary power is not a constant with change in microphone position for hydrogen-air and hydrogen-oxygen flames. This indicates that the flames are a non-monopole source of sound.

5.3 Center Frequency for Premixed Turbulent Flames:--

For the premixed flames, the center frequency versus microphone position is shown in Figure 34. (Plate V is a typical frequency spectrum for premixed hydrogen and air flames.)

For hydrogen-air flames the center frequency gradually decreases with increase in microphone position, θ . The range is 1500-900 hertz for the air mass flow rate of 19.1 grams/sec and the equivalence ratio of 0.78. In the case of the hydrogen-oxygen flames, there is the same gradual decrease in the center frequency, f_c , with decrease in θ , but for a frequency range of 2000-1400 hertz, for $\dot{m}_{O_2} = 20.55$ grams/sec, $\hat{\phi} \sim 0.166$.

Moreover, for leaner premixed hydrogen-air flames with an air mass flow rate of 19.1 grams/sec, the center frequency decreases to 600-300 hertz. This is just the opposite of what happened for the non-premixed hydrogen-air flames (Figure 20). There, for leaner mixtures, the center frequency came closer to the cold jet frequency (i.e., f_c became larger). However, for nearly the same equivalence ratio, the center frequencies are slightly lower (900-1500 hertz) for the premixed as compared to the non-premixed flames (2000-3000 hertz).

6. Acoustical Combustion Efficiency and a Note on the Effect of Microphone Position on Center Frequencies for Turbulent Flames

The next parameter to consider is the acoustical combustion efficiency. From Chp. 2, Sec. C.3, the percent of energy appearing as noise is:

$$\eta = \pi / \dot{m}_f \Delta H_c / \dot{m}_f$$

Figure 35 shows the acoustic combustion efficiency as a function of the fuel volume flow rate. The results show that most values of the non-premixed flames form a linear relationship between η and $(uA)_f$, except hydrogen-oxygen flames at a low Reynolds number. In this case, the com-

bustion efficiency decreases with increasing fuel volume flow rate. The second aspect of Figure 35 is that the data for the premixed flames follow a cubic relation between η and $(uA)_f$, i.e.,

$$\eta \propto (uA)_f^3$$

Another way to correlate the data is shown in Figure 36. Here, the combustion efficiencies for hydrogen-air, hydrogen-oxygen, and ethylene are shown versus the fuel mass flow rate. This figure indicates that perhaps the correct correlating parameter is $(\dot{S}u)_f$. A linear relation between η and $\dot{m}_f$ is drawn in Figure 36 to show this relationship.

At this time, further observations concerning the roughness of burning should be mentioned. The higher noise level of the hydrogen-oxygen flames in comparison to the hydrogen-air flames at an increased air mass flow rate was attributed to what was termed a rougher burning process. By this, it was implied that pockets of unburnt fuel and oxidizer were being formed, and the larger these pockets became, the rougher the burning became and the louder the noise level per pocket became upon burning.

Experimentally, the roughness was recognized by the fluctuating sound-level meter readings as evidenced by the ethylene-oxygen flame data presented in Figure 35.

The experiments with premixed flames also exhibited a roughness for the experiments where $\theta \neq 116.5$. They are shown in Figure 37. Here, again, the data points far from the correlation line of $\eta = C_1(uA)_f^3$ are those points where considerable meter deviations occurred, indicating a rough burning process and what was observed as a fluctuating sound. The rough burning process (the non-periodic burning of enlarged pockets of combustible gases) need not have the exaggerated character as evidenced by the experiments with non-premixed ethylene-oxygen flames as shown in Figure 35, or the results of the premixed flame experiments shown in Figure 37, but the burning process may gradually increase in roughness as the Reynolds number is increased (and here a simultaneous relative stream speeds increase). Moreover, this unsteady combustion process may take place so rapidly that the rough burning will be a superposition of additional acoustical energy, and will be sensed only as an overall loudness, i.e., higher db. From the above experiments the strength of the roughness seems to be strongly coupled with the fuel and oxidizer turbulence, and burning speed of the mixtures. As shown in Figure 27, for the same oxidizer Reynolds number ($Re^{ox} \sim 0.44 \times 10^5$),

the noise from the hydrogen-oxygen flames is less intense than hydrogen-air flames. Also, the jet mixing region for the hydrogen-oxygen flames has a lower degree of turbulence than that of the hydrogen-air flames. But, at higher levels ($R_{e}^{ox} \sim 1.75 \times 10^5$) the noise of the hydrogen-oxygen flames was considerably louder than the hydrogen-air flames. For the latter case, the greater noise of the hydrogen-oxygen flames can be attributed to the fact that, even if for each case, the size (or volume) of the accumulating pockets is the same, for the hydrogen-oxygen flames the burning rate is greater and the rate of change of the rate of fuel consumption could be greater.

For a single source it was found that the acoustical power is related to the rate of change of the rate of fuel consumed in the following manner:

$$\pi \sim \left| \frac{d}{dt}(\dot{m}_f) \right|^2$$

or, in terms of frequency:

$$\begin{aligned} \pi &\sim \overline{g u A^2} f^2 \\ &\sim \text{constant } u_b^2 f_c^2 \end{aligned}$$

From Figures 20 and 26 we see that at the higher oxidizer mass flow rates ($\dot{m}_{air} = 19.1$ grams/sec, $\dot{m}_{o_2} = 21.19$ grams/sec), the center frequencies for the hydrogen-oxygen flames

are generally higher than the center frequencies for the hydrogen-air flames. This fact, together with the fact that the flame speed for hydrogen-oxygen is greater than the flame speed for hydrogen-air flames, shows that the simple monopole relationship between acoustical power, flame speed and frequency implies the same result as was found experimentally. At the lower oxidizer flow rates, the center frequencies for the hydrogen air flames are considerably greater than the center frequencies of hydrogen-oxygen flames. Thus, the trends indicated in Figure 22 are consistent with the monopole source-of-noise model.

Now consider the trends of data indicated in Figure 27. For the higher oxidizer mass flux, the center frequencies for ethylene-oxygen flames are higher than either those of hydrogen-oxygen or hydrogen-air flames. This fact leads to the conclusion that ethylene-oxygen flames should radiate the greater acoustical power if the oxidizer mass flow rates were equal to those of the hydrogen-air and hydrogen-oxygen flames. This was also the conclusion drawn from the experimental measurements of the sound-pressure levels for these flames.

At the lower oxidizer flow rates, the ethylene flames have approximately the same center frequencies as

the hydrogen-oxygen flames and lower center frequencies than the hydrogen-air flames. For comparison of noise levels between ethylene-oxygen and hydrogen-air flames, the trends obtained experimentally (i.e., the hydrogen-air flames have the larger noise level of the two) agree with the qualitative predictions of the monopole source-of-noise model as demonstrated with the hydrogen-oxidizer flames. However, for the comparison of the noise levels between ethylene-oxygen and hydrogen-oxygen flames the fact that the degree of turbulence of the ethylene-oxygen flame is the larger of the two implies that the ethylene-oxygen flame has the larger burning zone of the two*. Thus, for a collection of monopole sources, the expression of acoustical power is written as:

$$\Pi \sim \frac{V}{\Delta V} \overline{suA}^2 f_c^2$$

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*Since the degree of turbulence is the rate of jet boundary layer spread with distance, it can also be considered as a relative indicator of the volume of the combustion zone.

where

$V \sim$ Volume of the combustion zone

$\Delta V \sim$ Source volumes

Hence, since ethylene-oxygen flames have a greater degree of turbulence than those of hydrogen-oxygen flames, it should also have the greater value of $V/\Delta V$ and, thus, be the louder of the two flames. Hence, the trend indicated by the data presented in Figure 27 is consistent with the monopole model of noise.

Another aspect of the results given in this chapter was that, generally, the center frequency shifted to lower levels for increasing values of the microphone position θ . This shift to the lower frequencies may be explained by the observation that the hot jet has a lower frequency than the cold jet. Plates VI, VII, show shadowgraphs of the cold jet, and that of a burning hydrogen-air mixture. From Plate VII it is seen that near the port of the burner there is a laminar flame. It can also be seen that there is a confined turbulent burning*

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*Looking at this region closely, acoustic waves can be seen radiating from this region.

with the inner core air jet, farther downstream a thicker turbulent combustion zone is formed. Hence, it can be said, that as the microphone is moved to larger and larger angles θ , it collects more noise from the combustion zone and, hence, the lower frequencies of the combustion zone predominate.

7. Burner Diameter Effects on Co-Flowing Turbulent Non-Premixed Flames.

7.1 Diameter Effects on Noise Level:--the last set of data for the concentric burners concerns the change in diameter of the inner tube from the standard 0.813 cm to 0.391 cm. Figure 38 shows a comparison of the noise levels from hydrogen-air flames for both of these burners for mass flows of 10.04 grams/sec and 7.21 grams/sec, respectively. It is readily seen that the noise level from the burner with smaller inner diameter tube is higher even though its initial air mass flow rate is less. Evidently, at equal air mass flow the noise level of the smaller tube will even be greater than shown. What should be noted is that the Reynolds number and speed is considerably larger for the smaller inner diameter tube; in fact, the speed of the air at the exit is sonic. The Reynolds number is also slightly larger for the 0.391 cm burner.

Hence, all the previous data would suggest that this is the expected trend for increased noise-levels. Thus, Figure 38 shows that for equal oxidizer flows, 1) the decrease in inner tube diameter produces more noise than burners of inner tubes of a larger diameter, 2) that the trend of noise-level changes with change in equivalence ratio is the same as for the 0.813 cm inner diameter tube and, 3) that the ratio of acoustical power due to combustion to the cold jet acoustic power is on the order of 100.

7.2 Diameter Effects on Directivity:--Figure 39, showing the directivity of the 0.391 cm burner, indicates that there is an acoustic intensity preference around $\theta \sim 135^\circ$ for $\hat{\phi} \sim 0.60$ (illustrating the same features as for the 0.813 cm burner). To illustrate further how the peak intensities occur, Figure 40 is presented showing sound-pressure-level for the range of fuel flow rates used. These figures (i.e., 39 and 40) clearly show directivity effects. Figure 41 shows the stationary power vs. θ ; again, the distribution of Π_s with θ is not what would be expected from a collection of monopole sources.

7.3 Diameter Effects on Center Frequency:--The center frequency for the 0.391 cm burner (Figure 42) is similar to that of the 0.813 cm burner in that the

flames of the richer mixtures of hydrogen-air have lower center frequencies, while the flames of the leaner mixtures have the higher frequencies. The order of magnitude of center frequencies observed are f_c 3000-5000 hertz for $\phi \sim 0.6$. The center frequency of the cold jet was 15000 hertz. As in previous cases studied, the center frequencies decrease with increase in microphone position θ ; the reason is probably the same as was given in section 6. Hence, the observed data trends for noise-levels, directionality, and center frequencies are all very much like what was observed for the nominal 0.813 cm concentric burner.

B. Impinging Non-Premixed Turbulent Flames

In order to supplement the study of co-flowing non-premixed flames, impinging streams of hydrogen and air were also investigated. A diagram of the arrangement of this burner is shown in Figure 43. Fuel and air streams flow separately from cylindrical stainless steel tubes having the same diameters. Both tubes are inclined to the vertical by the same

Figure 43 - Diagram of Impinging Burner Experiments for Hydrogen and Air.

angle. The microphone is placed 95.098 cm away from the point of impingement of the jets. The orientation of the microphone is $\theta = 90^\circ$, $\phi = 180^\circ$ as shown in Figure 43. The diameters of the tubes were 1.1303, 0.8079, and 0.4981 cm. I.D. The impingement angles were 45° , 90° , 135° .

Figure 44 illustrates the effect of the Reynolds number for an impingement angle $\chi = 45^\circ$ and $D = 1.1303$ cm. Like for co-flowing non-premixed hydrogen and air flames (Figure 14), of the sound level of the impinging flames is increased when the Re^{air} is increased. The tendency of the curves to come closer with increased air flux is also evident. The ratio of the resulting acoustic energy of hot to cold jets is $w/w_{jet} = 44.7$ for $\chi = 45^\circ$, $\dot{m}_{air} = 19.1$ grams/sec, $D = 1.13$ cm; and $w/w_{jet} = 45$ for $\chi = 90^\circ$, $D = 1.13$ cm. $\dot{m}_{air} = 15.03$ grams/sec. This ratio is practically the same as that obtained for the co-flowing hydrogen-air flames at high Reynolds numbers. It should also be noted that these curves show a maximum sound-pressure level is reached for the values of equivalence ratios considered, whereas the noise level curves for the flames of co-flowing jets showed a steady increase in sound-pressure level for the same values of fuel and oxidizer flow.

Diameter effects are illustrated in Figure 45. It is readily seen that for a fixed impingement angle, a decrease in the tube diameter increases the noise level. Thus, when the diameters of the burner tubes is decreased by a factor of 2.26, the acoustical energy released is increased by a factor of 5 for $\dot{m}_{\text{air}} = 19.1$ grams/sec, $\chi = 45^\circ$.

The effect of impingement angle variation for all three diameter tubes is shown in Figures 46, 47, and 48. It is apparent that for a fixed mass flux (or equivalence ratio) increase in impingement angle decreases the noise level. According to Braggs' theory, this trend is correct. In Braggs' expression for the estimated acoustical power:

$$\pi \propto |u_{ox} - u_f|^2$$

for the impinging flow case, u_{ox} , u_f should be replaced by their vertical components; $u_{ox} \cos (\frac{1}{2}\chi)$ and $u_f \cos (\frac{1}{2}\chi)$. Thus, with increasing χ the vertical components become smaller and smaller. Hence, for the combustion of impinging non-premixed jets, the simple model of a monopole source of combustion appears to apply.

The calculation of the acoustic combustion efficiency, η , for the impinging cases become most interesting when compared with the results for the co-flowing jets

for the same fuel volume flux. This comparison is shown in Figure 49 which shows the acoustic combustion efficiency as a function of the volume flux of the fuel. The trend for the flames of imping jets is opposite to that of the flames of co-flowing jets. The co-flowing non-premixed flames follow a relationship of

$$\eta \propto (uA)_f$$

whereas, the results for flames of impinging jets follow a relationship of

$$\eta \propto 1/(uA)_f$$

The order of magnitude of η for flames of impinging jets is between 0.5×10^{-6} to 1.6×10^{-6} for $D = 1.13$ cm., and 2.5×10^{-6} to 6.0×10^{-6} for $D = 0.4981$ cm. (these latter values would be representative for $D = 0.8079$ cm. also). Thus, for approximately the same Reynolds number, it may be said that the combustion of impinging jets yield slightly higher η 's compared to the combustion of co-flowing jets of non-premixed flames of hydrogen and air.

In Figure 50, the acoustic combustion efficiency is compared to: $|u_{Air}^v - u_{H_2}^v|^2$

where

$$u_{Air}^v = u_{Air} \cos(1/2\gamma)$$

$$u_{H_2}^v = u_{H_2} \cos(1/2\gamma)$$

It is readily apparent that the data for $D = 1.13$ cm. at $\dot{m}_{air} = 19.1$ grams/sec. closely follow the linear relation between η and $(u_{ox}^v - u_f^v)^2$, and thus, Braggs' theory. For $D = 0.4981$ cm, $\dot{m}_{air} = 19.1$ grams/sec, the data of η for $|u_{Air}^v - u_{H_2}^v|^2$ also follows a corresponding linear relation, but only at higher $|u_{Air}^v - u_{H_2}^v|^2$. For its smaller values of $|u_{Air}^v - u_{H_2}^v|^2$, the trend is still to increase in η with increase in $|u_{Air}^v - u_{H_2}^v|^2$, but not at a linear rate as did the data of the other burners*.

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* It should be noted that for $D = 0.4981$ cm. and $\dot{m}_{air} = 19.1$ grams/sec, the air exit speed is sonic, and the exit pressure is above atmospheric. These conditions may lead to additional acoustic cold jet noise that is not ordinarily present for the subsonic flows.

As the relative speed between the jets of fuel and oxidizer is further increased (or as u_f is decreased), this trend must reverse and go to pure jet noise, but for the range of values tested the model is valid.

As with the co-flowing cases, frequency spectra were taken of the emitted sound of the flames from impinging jets. Some typical spectra are shown in Plates VIII-XII. From these, it can again be seen that the combustion process excites the lower frequency range and broadens the band of frequencies in comparison to the pure jet noise.

The results of the spectrum analysis are shown in Figures 51, 52, 53, and 54. Here we see that for a fixed diameter burner, an increase of the impingement angle generally tends to lower the value of the center frequency (Figures 51, 52, 53 for $D = 0.4981, 0.8079,$ and 1.13 cm, respectively). It is also seen that an increase in the equivalence ratio decreases the value of f_c . For a fixed impingement angle and burner diameter, decrease in fuel mass flux generally decreased the center frequency, too (see Figure 54). Typical values for center frequencies are 600 to 1000 hertz for $D = 1.13$ cm, $\dot{m}_{air} = 15.03$ grams/sec, and 1600 to 2000 hertz for $D = 0.8079$ cm, $\dot{m}_{air} = 19.1$ grams/sec.

A few additional remarks should be made concerning the combustion of impinging non-premixed jets. In many cases the meter readings of the sound-pressure-levels fluctuated. These fluctuations were slow in nature (unlike the rapid and large amplitude fluctuations of the co-flowing cases discussed earlier), and usually never more than 0.5 to 0.8 db. difference in amplitude. Both, the high and low readings were recorded, and a linear average was taken as the final point. In exploratory experiments, gases other than air, were used to impinge upon the hydrogen jet. It was found that by using air, oxygen, or nitrogen, the overall sound-level was practically the same. It is, therefore, suspected that the impinging stream acts to "stir" the hydrogen with the surrounding air, and not necessarily react with it.

CONCLUSIONS

The following general conclusions can be summarized from the present study of the noise generated by turbulent non-premixed flames.

(1) The simple monopole source-of-noise model for combustion qualitatively describes the general acoustical features of the noise emitted from turbulent premixed and non-premixed, co-flowing and impinging type of flames considered in this investigation. But, due to the extended lengths of the flames of co-flowing non-premixed jets, the flames tested do not act like a simple collection of monopole sources-of-noise.

(2) For the premixed flames and non-premixed co-flowing turbulent flames investigated, a somewhat pessimistic view concerning the reduction of combustion noise is elicited from the experimental findings, namely,

that in the former case the acoustical combustion efficiency obeys a "fuel volume flux" cubic law, $\eta \sim \text{constant} \times (uA)^3$, and in the latter case the acoustic combustion efficiency obeys a linear "fuel volume flux" law, $\eta \sim \text{constant} \times (uA)$. Thus, unlike the velocity law for emitted acoustical energy for subsonic jets¹, small changes in the fuel velocity yield small reductions in emitted acoustical energy.

In addition to the above general conclusions, the following remarks can be made in regard to some of the more specific results of the present investigation.

(1) In addition to the Reynolds number of the gas jets, the combustion "roughness" or "instability" increases the acoustic energy produced by the flames. The instability process is pictured as occurring in the following way: by increasing the Reynolds number of the jets, and by also increasing the relative speeds between the jets, the burning process is delayed because of the higher relative speed between the fuel and oxidizer and larger eddies of reactants which must burn. The delayed burning allows accumulations of unburnt reactants to form a premixed combustible volume of gas. When the mixed reactants ignite and burn, they send forth a pressure pulse whose strength is dependent upon the reactivity of the mixture (the flame speed), and

the initial volume of the unburnt gas. The acoustic energy emitted from the flames of non-premixed jets is hypothesized to be a superposition of a basic non-premixed combustion source(monopole in structure) and a premixed combustion source (monopole in structure). The rapidity with which additional acoustic noise due to the premixed sources occur, depends on the relative speed between the fuel and oxidizer streams (more rapid and periodic for the lower Reynolds number and smaller relative speeds, less rapid and non-periodic with higher jet Reynolds numbers and greater relative jet speeds).

(2) For turbulent co-flowing non-premixed hydrogen air flames, the calculated Strouhal number decreased in value with increase in equivalence ratio. This variation of Strouhal number suggests that there is a characteristic length (most likely, a mixing length) for combustion, which may be related to the noise emitted from turbulent flames.

(3) The acoustical energy produced by burning combustible gas jets was considerably greater than that emitted by the unburned jets, and the ratio of energies was found to be from 10 (for low Reynolds number hydrogen-air flames) to 550 (for high Reynolds number ethylene-oxygen flames).

In all cases, the combustion process showed a dominant frequency. The frequency spectrum of the flames was generally broader than the cold jet spectrum and in a lower frequency range.

(4) The acoustic combustion efficiency varied from 5×10^{-8} (for low Reynolds number hydrogen-oxygen co-flowing non-premixed flames) to 2×10^{-5} (for high Reynolds number ethylene-oxygen co-flowing non-premixed flames).

(5) A correlation parameter for acoustic combustion efficiency for the turbulent co-flowing flames considered in this investigation was found to be the fuel mass flux.

(6) For all the non-premixed flames considered, a decrease in burner diameter (oxidizer diameter for flames of co-flowing jets; fuel and oxidizer diameters for flames of impinging jets) increased the overall noise level.

(7) For the flames of impinging jets of hydrogen-air the acoustic combustion efficiency was found to be given by the relationship

$$\eta \sim \text{Constant} \times 1/(UA)_f$$

It was also found that

$$\eta \sim \text{constant} \times |u_{\text{air}}^{\vee} - u_{\text{Hz}}^{\vee}|^2$$

which corresponds to the simple monopole idea of combustion noise.

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Fig. 14 Sound-Pressure-Level Curves for $H_2 + Air$ Flames for Various Oxidizer Mass Flow Rates

Fig. 15 Sound Pressure Level Curves at Various Microphone Positions for $\hat{\phi} = 0.31$ and 0.77 for $\text{H}_2 + \text{Air}$ Flames

Fig. 16 Stationary Monopole Acoustical Power for $H_2 + \text{Air}$ Flames

Fig. 17 Stationary Monopole Acoustical Power vs.
 Equivalence Ratio for $H_2 + Air$ Flames

Fig. 18 Experimental Acoustical Power vs. Equivalence Ratio for $H_2 + \text{Air}$ Flames

Fig. 19 Root-Mean-Square Acoustical Pressure vs. Equivalence Ratio for $H_2 + Air$ Flames

Fig. 20 Center Frequencies for $H_2 + Air$ Flames vs. θ

Fig. 21 Strouhal Number vs. ϕ for H_2 +Air Flames

Fig. 22 Comparison of Noise Levels for Non-Premixed Flames

Fig. 23 Degree of Turbulence vs. ϕ

Fig. 24 Noise-Level for $H_2 + O_2$ Flames vs. θ

Fig. 25 Stationary Monopole Acoustical Power vs. θ for $\text{H}_2 + \text{O}_2$ Flames

Fig. 26 Center Frequencies vs. θ for $H_2 + O_2$ Flames

Fig.27 Comparison of Noise Levels for Non-Premixed $H_2 + O_2$, $H_2 + Air$, $C_2H_4 + O_2$ and $C_2H_4 + Air$ Flames

Fig. 28
 Sound-Pressure-Level vs. θ for $\text{C}_2\text{H}_4 + \text{O}_2$ Flames

Fig. 29
 Stationary Monopole Acoustical Power
 vs. Θ for $C_2H_4 + O_2$ Flames

Fig. 30 Center Frequency vs. Equivalence Ratio for $C_2H_4 + O_2$ and $C_2H_4 + \text{Air}$ Flames

Fig. 31
 Comparison of Noise Levels of Non-Premixed and Premixed Flames

Fig. 32
 Directionality of Premixed $\text{H}_2 + \text{Air}$ Flames

Fig. 33
 Stationary Monopole Acoustical Power
 vs. θ for $\text{H}_2 + \text{Air}$ Premixed Flames

Fig. 34
 Center Frequency for Premixed H_2+Air Flames vs. θ

Fig. 35 Acoustic Combustion Efficiency vs. Fuel Volume Flow Rate for Non-Premixed and Premixed Flames

Fig. 36
Acoustic Combustion Efficiency vs. Fuel Mass Flow Rate

Fig. 37
Acoustic Combustion Efficiency vs. Fuel Volume
Flow Rate for Premixed Flames

Fig. 38
Comparison of Noise Levels for Different
Diameter Concentric Burners

Fig. 39
 Directionality of $H_2 + Air$ Non-Premixed Flames
 for 0.391 cm Concentric Burner

Fig.40 Effect of Fuel Mass Flow Rate and θ on the Noise Level of $H_2 + \text{Air}$ Non-Premixed Flames for 0.391 cm Concentric Burner

Fig. 41
 Stationary Monopole Acoustical Power vs.
 θ for $H_2 + \text{Air}$ Non-Premixed Flames for
 0.391 cm Concentric Burner

Fig. 42
 Center Frequency for $H_2 + Air$ Flames
 for 0.391 cm Concentric Burner

Fig. 44
Sound-Pressure-Level Curve for Impinging
 $H_2 + Air$ Flames at $\chi = 45^\circ$, $D = 1.1303 \text{ cm}$
for Various Oxidizer Mass Flow Rates

Fig. 45
 Diameter Effects on Noise Level of Impinging $\text{H}_2 + \text{Air}$ Flames

Fig. 46
 Impingement Angle Effect on Noise-Level of
 Impinging $H_2 + \text{Air}$ Flames: $D = 0.4981 \text{ cm}$.

Fig. 47 Impingement Angle Effect on Noise-Level of Impinging $\text{H}_2 + \text{Air}$ Flames: $D = 0.8079$

Fig. 48
 Impingement Angle Effect on Noise-Level
 of Impinging $\text{H}_2 + \text{Air}$ Flames: $D = 1.1303 \text{ cm}$.

Fig. 49 Comparison of Acoustic Combustion Efficiencies of Non-Premixed Co-Flowing Flames and Non-Premixed Impinging Flames

Fig. 50
 Combustion Efficiency vs. $|u'_{\text{air}} - u'_{u_0}|^2$
 for Non-Premixed Impinging $\text{H}_2 + \text{Air}$ Flames

Fig. 51
 Effect of Impingement Angles on Center Frequency
 for Impinging $H_2 + \text{Air}$ Flames: $D = 0.4981 \text{ cm.}$

Fig. 52
 Effect of Impingement Angles on Center Frequency
 for Impinging $H_2 + \text{Air}$ Flames: $D = 0.8078 \text{ cm}$.

Fig. 53
 Effect of Impingement Angles on Center Frequency
 for Impinging $H_2 + \text{Air}$ Flames: $D = 1.1303 \text{ cm}$.

Fig. 54

Effect of Air Mass Flow Rate on Center
 Frequency for Impinging $H_2 + \text{Air}$ Flames:
 $D = 1.1303$; $\chi = 45^\circ$

Plate I Co-flowing non-premixed hydrogen and air for nominal concentric burner: $\phi = 0.77$, $\dot{m}_{air} = 19.1$ grams/sec, SPL = 102.5 db., $f_c = 3000$ hertz.

Plate II Air jet for nominal concentric burner: $\phi = 0.0$, $\dot{m}_{air} = 19.1$ grams/sec, SPL = 86.2 db., $f_c = 6500$ hertz.

Plate III Co-flowing non-premixed hydrogen and oxygen for nominal concentric burner: $\phi = 0.16$, $\dot{m}_{o_2} = 21.19$ grams/sec, SPL=105.1 db., $f_c = 5000$ hertz.

Plate IV Co-flowing non-premixed ethylene and oxygen for nominal concentric burner: $\phi = 0.17$, $\dot{m}_{o_2} = 10.18$ grams/sec, $f_c = 6000$ hertz.

Plate V Co-flowing premixed hydrogen and air for nominal concentric burner: $\phi = 0.77$, $\dot{m}_{\text{air}} = 19.1$ grams/sec, $f_c = 1200$ hertz.

Plate VI Shadowgraph of an Air Jet for nominal burner: $\phi = 0.0$, $\dot{m}_{air} = 19.1$ grams/sec.

Plate VII Shadowgraph of ignited hydrogen and air for nominal burner: $\phi = 0.77$, $\dot{m}_{air} = 19.1$ grams/sec.

Plate VIII Impinging non-premixed hydrogen and air for $D=1.130$ cm, $\chi = 45^\circ$ at $\phi = 0.976$, $\dot{m}_{\text{air}}=15.03$ grams/sec, SPL=92.5 db, $f_c=700$ hertz.

Plate IX Impinging non-premixed hydrogen and air for $D=1.130$ cm, $\chi = 135^\circ$ at $\phi = 0.284$, $\dot{m}_{\text{air}}=15.03$ grams/sec, SPL=93 db, $f_c=1000$ hertz.

Plate X, Air Jet for $D=0.4981$ cm, $\chi = 45^\circ$ at $\phi = 0.0$, $\dot{m}_{air}=19.1$ grams/sec, SPL=90.3 db, $f_c=7000$ hertz.

Plate XI Impinging non-premixed hydrogen and air for $D=0.8079$ cm, $\chi = 45^\circ$, $\phi = 0.4090$, $\dot{m}_{air}=19.1$ grams/sec, SPL=100.5 db, $f_c=2000$ hertz.

Plate XII Impinging non-premixed hydrogen and air for $D=0.8079$ cm, $\gamma=135^\circ$, $\phi=0.7695$, $m_{\text{air}}=19.1$ grams/sec, $\text{SPL}=97.5$ db, $f_c=2000$ hertz.