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ENGINEERING RESEARCH INSTITUTE
UNIVERSITY OF MICHIGAN
ANN ARBOR

PROGRESS REPORT NO. 1

INVESTIGATION OF OIL-FIRED RADIANT BURNERS

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2. Air sufficient for stoichiometric combustion can be induced by operation of an air-atomized fuel nozzle in a converging inlet section with the shape of an Archimedes spiral.

3. Atomization of fuel oils can be simplified and coking reduced or eliminated by vaporization and partial combustion in a precombustion chamber. This allows the use of very simple fuel nozzles constructed from standard pipe "tees" with air atomization. Uniform cup temperatures of 2500°F are consistently obtained in full-scale open radiant burners, at stoichiometric ratios with the No. 4 fuel oil. Temperatures in enclosed burners are considerably higher and must be reduced by operating with lean mixtures to prevent structural failure.

4. The "petal" effect in radiant cup burners can be eliminated by redesign of the cup inlet. Schlieren photographs of two-dimensional burners show that the "petals" are caused by divergences of flow around a sharp corner at the cup inlet, leaving a cold region of no gas flow next to the cup wall. Aerodynamic streamlining of the cup inlet eliminated these petals and produced a uniform cup temperature.

In addition to these achievements, an understanding of the basic variables in the operation of radiant burners with liquid fuels has been obtained and a systematic evaluation of these variables is currently been made. The information obtained in the study of round cup burners has been applied to construct experimental burners which might be installed as a continuous radiant strip in a furnace wall. Only enough work has been done to show that this is a promising concept.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

PROGRESS REPORT NO. 1

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INTRODUCTION

The long range objective of the combustion research sponsored by the Selas Corporation of America at the University of Michigan is the development of an effective, oil-fired radiant burner which can also be operated with gaseous fuels. The immediate objective is to obtain an understanding of those factors which contribute to the effectiveness of an oil-fired radiant burner.

The problem was attacked simultaneously in several ways. Separate investigations of components parts, such as fuel atomization nozzles, air induction systems, and burner designs were conducted. As rapidly as information was obtained from these component studies it was integrated into experimental burner designs. Such work is still continuing as detailed quantitation relationships among the components become apparent.

Concurrently with this program, full-scale burners were operated with propane and various liquid fuels, both to test the proposed designs and to familiarize personnel with the characteristics and desirable operating features of the burners.

This report summarizes the qualitative and quantitative information obtained to date.

ANALYSIS OF THE PROBLEM

The controlled transfer of heat in Selas radiant burners is accomplished by thermal radiation from a hot ceramic surface. In gas-operated burners, combustion occurs in the cup itself, thus maintaining the high surface temperature of the ceramic. The surface temperature can be controlled by changing the fuel rate.

Liquid fuels introduce additional variables into the operation of radiant burners with a uniform surface temperature. Several alternative methods of introducing liquid fuels were considered.

1. Liquid fuel can be introduced, through gas ports in existing burner tips. Since this did not seem very promising, no experiments using this procedure were performed.

2. The burner tip could be replaced by a spray nozzle located in the center of the cup. Preliminary experiments indicated that burning of the fuel did not achieve radiant cup temperatures.

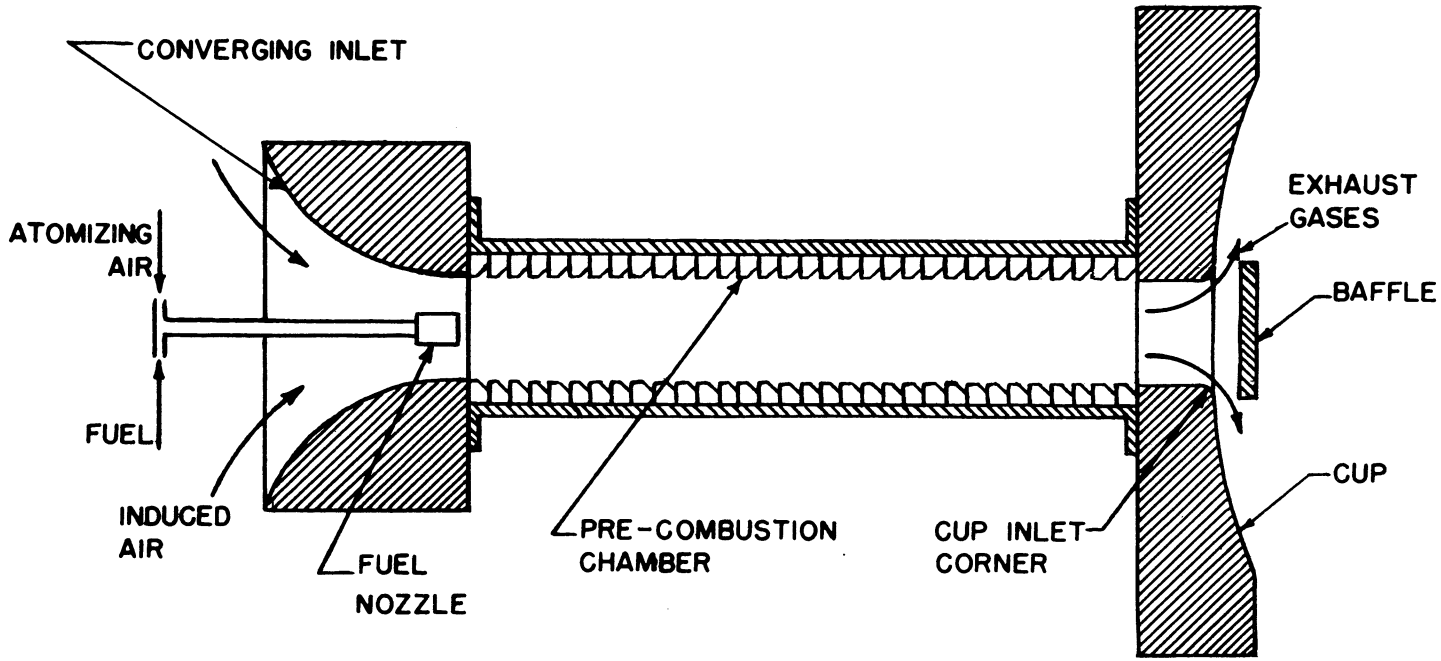
3. Liquid fuel could be atomized upstream of the burner tip and the fuel-air mixture would pass through both the annular holes and the central duct of present burner tips. Experiments using this technique indicated that burning occurred upstream of the burner tip. This indicated the fourth possibility.

4. Atomized liquid fuel and air enter a precombustion chamber where partial combustion occurs and the hot exit gases, in addition to surface combustion, maintain the high cup temperature. This technique greatly reduces the magnitude of the fuel-atomization problem. Successful burning of No. 4 fuel oil in 12-inch radiant burners was obtained with uniform cup temperatures. The burner tip was replaced by a flat ceramic baffle so that the parabolic cup shape was satisfactory. However, as the result of a suggestion by Mr. A. A. Furczyk of the Selas Corporation, research was initiated on a fifth possibility.

5. Fuel and air are partially burned in a precombustion chamber, the geometric shape is such that baffles and burner tips are eliminated. Preliminary experiments performed with propane in two-dimensional burners indicate that hot ceramic surfaces can be maintained without the use of baffles or burner tips.

While other techniques for burning liquid fuels in radiant burners undoubtedly exist experimental work was direct only along the above-mentioned lines. Since the greatest success in the exploratory research was achieved with the fourth technique, a statement of the variables is outlined below (See Fig. 2 for Nomenclature).

Required: Uniform cup temperature to be varied by varying the rate of fuel flow. Air sufficient for combustion to be induced by operation of air-atomizing fuel nozzle.



OIL-FIRED RADIANT BURNER- INDUCED AIR TYPE
FIGURE 2

A. Air Induction Systems

The amount of induced air is believed to be a function of

1. shape of converging inlet,
2. external shape and location of a given fuel nozzle,
3. pressure drop downstream of inlet nozzle (due to burning as well as baffle position),
4. Mass and velocity of driving fluid (air and fuel).

B. Fuel Nozzle

The droplet size and distribution is believed to be a function of

1. internal nozzle shape and dimensions,
2. fuel properties (viscosity, density, etc.),
3. for many air to fuel ratio at the nozzle as well as the absolute quantities of air and fuel in a given nozzle,
4. the velocity of the driving fluid, and
5. physical properties of the driving fluid.

C. Precombustion Chamber

The effectiveness of the precombustion chamber, as indicated by the pressure drop through the burner and the temperature at the exit of the precombustion chamber, is believed to be a function of

1. overall fuel-air ratio,
2. mass velocity of air and fuel, based on burner id,
3. velocity distribution at burner inlet and exit,
4. length and diameter of combustion chamber,
5. burner wall shape, surface, and temperature, and
6. degree of vaporization of fuel as well as droplet size and distribution.

D. Combustion Chamber Exit and Cup

The cup temperature in addition to being affected by all the above-mentioned variables is also believed to be a function of

1. shape of the inlet corner,
2. shape and location of baffle,
3. shape and surface roughness of cup, and
4. baffle and cup materials.

EXPERIMENTAL PROCEDURE AND RESULTS

I. INVESTIGATION OF OIL-FIRED RADIANT BURNER, INDUCED AIR TYPE

The initial phase of this research project was exploratory in nature but much of the experimental work is applicable to this type of burner. Therefore, the experimental work is presented according to the preceding analysis rather than the order in which the experiments were performed. Thus, the experiments which led to the development of this type of burner are included along with later experiments.

A. Air Induction Systems

1. Shape of Converging Inlet. In order to prevent friction losses, the converging inlet should approximate the path of the fluid streamlines. The pressure gradient on any fluid particle moving in a curved path of radius r may be found from the equation

$$\frac{dp}{dr} = \frac{\rho v^2}{r} ,$$

where p is the pressure, v the velocity, and ρ the density of the fluid particle. Since in three-dimensional flow a constant acceleration of the fluid particle is achieved when the fluid particle follows the path described by a spiral of Archimedes ($r = k\phi$), most of the experimental work was performed using a converging inlet of this shape and circular cross section. However, a limited number of experiments was performed with two-dimensional inlets whose longitudinal shape was elliptically approximated by the equation

$$(x^2/a^2) + (y^2/b^2) = 1 .$$

At a value of $a/b = 2$, a was varied from 3 to 10 inches, and at $a/b = 3/2$, a was varied from 3 to 10. With a circular shape, approximated by the equation $x^2 + y^2 = r^2$, r was varied from 3 to 10. Using two-dimensional Archimedian spiral-shaped inlets ($r = k\phi$), values of k (in inches) from $12.8/\pi$ to $44.8/\pi$ were tested. These early experiments indicated that as long as a streamlined shape was used the amount of air induced was not affected greatly by the shape of the inlet. Considerably more experimental data would be required to establish this point firmly.

The variation between streamlined and nonstreamlined shapes, however, is of considerably more importance than the variation between streamlined shapes represented by slightly different equations. Some experimental data confirming this is presented in Table I.

TABLE I

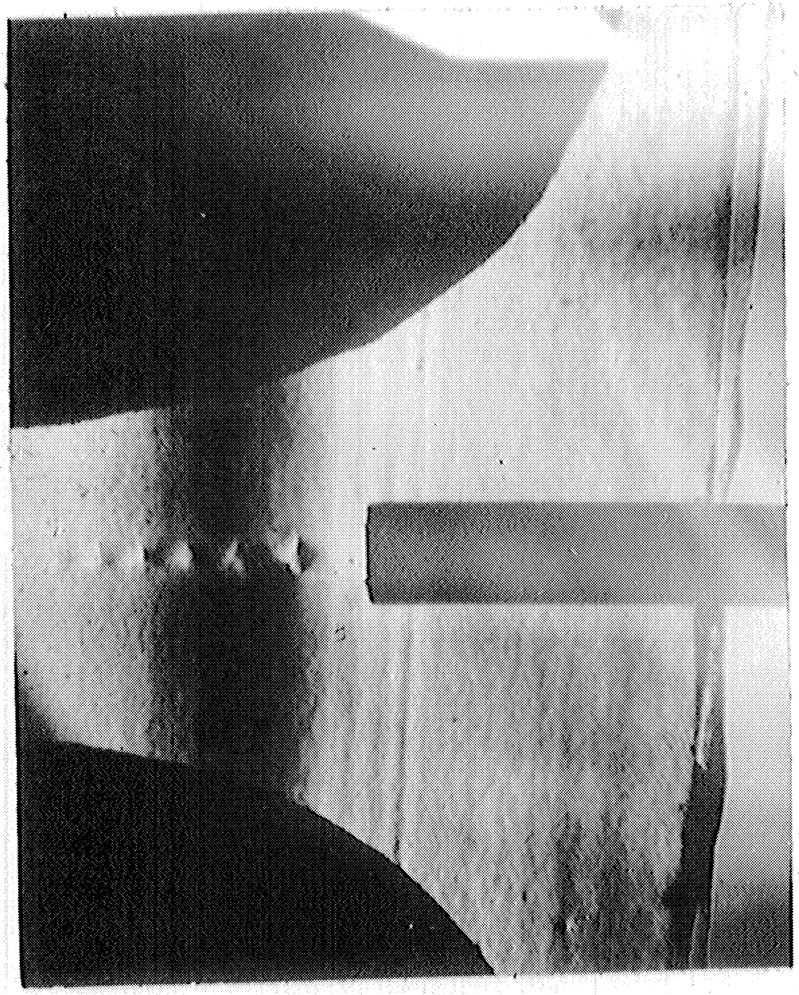
EFFECT OF INLET SHAPE ON AMOUNT OF AIR INDUCED

Shape	Ratio - lb Induced Air/lb Driving Air	
	No Burning	Burning
Chrome Burner Tip (Selas Part No. 31177-07)	0.05	
Cylindrical Tube (dia = 2 in)	5-15	
Reducing Coupling, 5 in. to 3 in dia. as installed on "Inspirator Radiant Oil Burner", Selas Sketch No. 1, 1-27-54, J.H.)		4.75
Archimedes Spiral (3 in. dia exit)	20-26	6.0

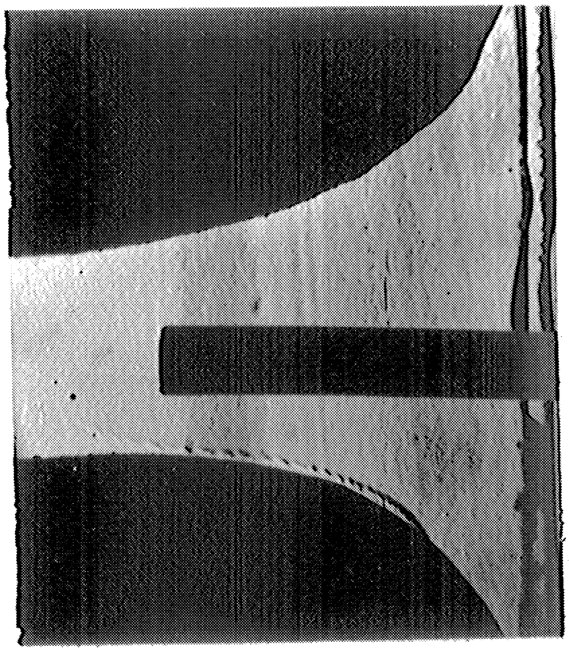
As may later be seen, the amount of induced air is also a function of the driving air pressure (or velocity), hence the variation in the nonburing experiments. The two burning experiments were conducted under comparable conditions.

Some schlieren photographs were made of inlet sections constructed from porous firebrick in the shape of Archimedian spirals. In Fig. 3, the driving air is issuing from a 1/4-inch diameter copper tube at a supersonic velocity as evidenced by the appearance of the Mach diamonds (shock and expansion wave reflections) in the jet. This supersonic velocity always occurs whenever the driving air pressure is greater than 1.895 times the pressure in the space where discharge occurs.

In Fig. 4, 5, and 6, the driving air velocity is subsonic. In these photographs propane was introduced in the boundary layer of the inlet section by means of a small plastic tube embedded in the surface. Since,

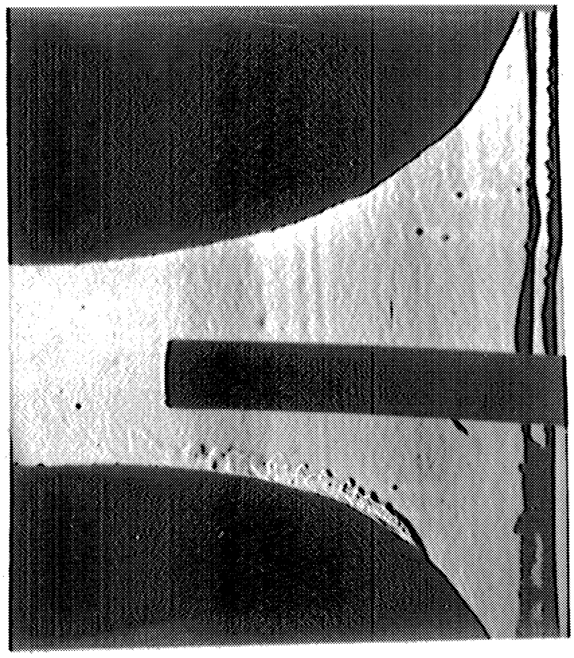


CONVERGING INLET SECTION
(SUPERSONIC DRIVER)
FIGURE 3



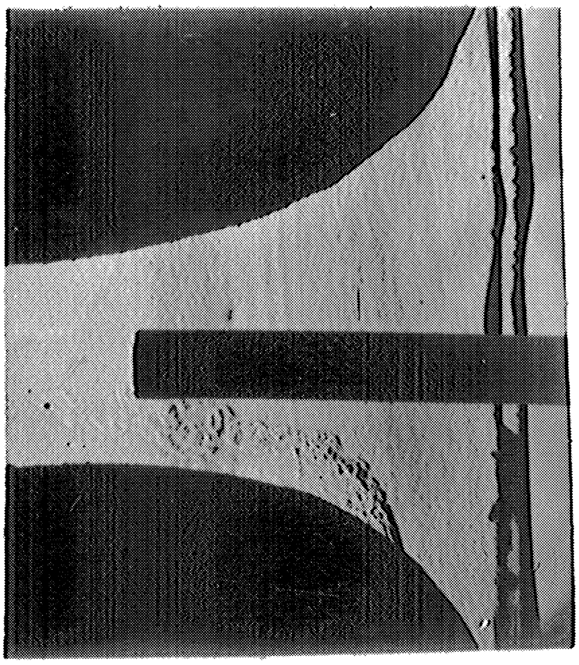
LAMINAR NEAR WALL

FIGURE 4



TURBULENT AWAY FROM WALL

FIGURE 5



TURBULENT NEAR CENTER

FIGURE 6

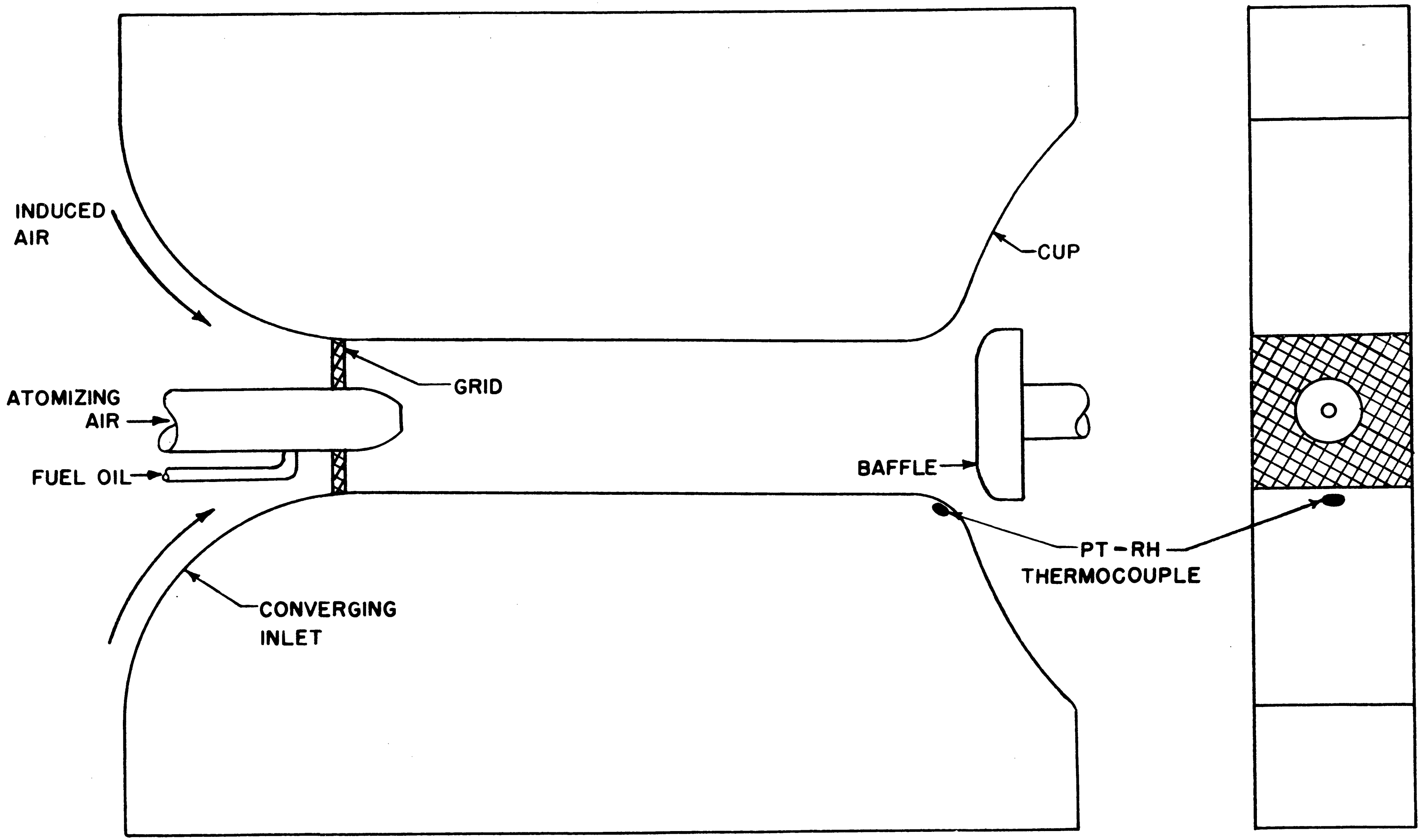
CONVERGING INLET SECTION WITH PROPANE INTRODUCED THROUGH WALL in a schlieren photograph, the variation in contrast is directly proportional to the first derivative of the fluid density with distance (i.e., the density gradient), the use of a gas with a greater density than air (in this case, propane) permits observation of flow patterns which normally do not exhibit large density gradients. Figure 4 indicates that the flow near the surface (the boundary layer) is laminar, while Fig. 5 and 6 indicate that turbulence exists farther away from the wall.

The fact that turbulence exists in the center of the induced air stream is not disadvantageous, however. Figure 7 is a sketch of the two-dimensional burner shown in Fig. 4, 5, and 6 with the addition of a liquid fuel nozzle. Experiments were performed with the grid present (as shown in Fig. 7) and without the grid present and it was found that the temperature at the exit of the precombustion chamber, as measured with a platinum-rhodium thermocouple imbedded in the wall, was 90°F greater when the grid was removed. Since the presence of the grid reduces the scale of the turbulence, poorer mixing and the resultant lower temperature occurred. The amount of air induced by the system was also lowered due to the friction drop across the grid.

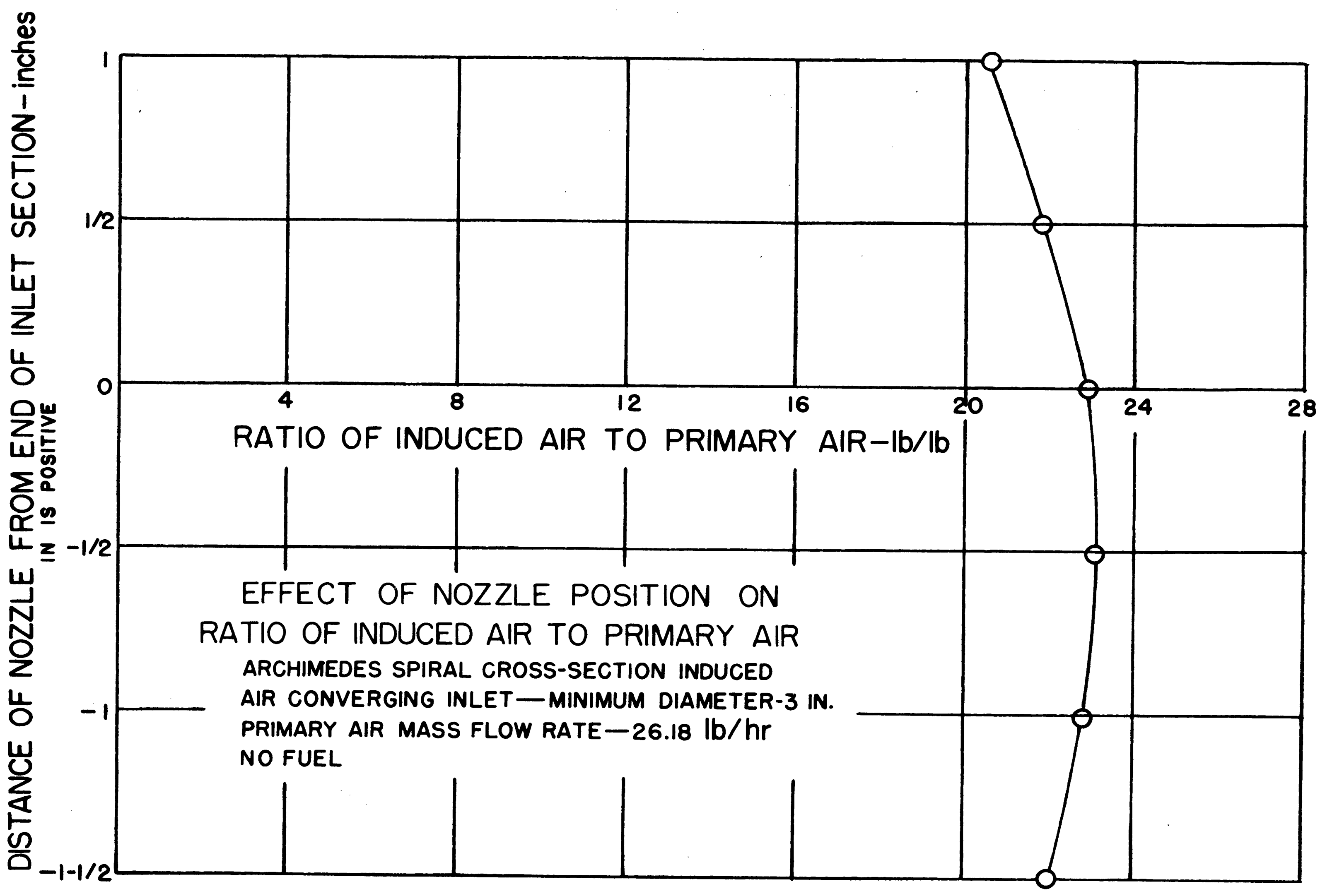
Thus, it has been shown that the shape of the inlet section used affects the amount of air which can be induced. Based on the present data, an inlet with an Archimedian spiral shape results in the greatest amount of air being induced. With inlets constructed to this shape, the scale of the turbulence and the amount of air induced affect the temperature at the exit of the precombustion chamber.

2. <u>External Shape and Location of Fuel Nozzle</u>. No quantitative experiments were performed to determine the effect of external nozzle shape on the amount of air induced. However, there is little doubt that the insertion of smaller, streamlined fuel nozzles results in the induction of larger quantities of air than the use of larger, wake-producing fuel nozzles.

Some experiments were performed in order to determine the effect of fuel nozzle location on the amount of air induced. The fuel nozzle used was the "Bloom" nozzle received with the "Inspirator Radiant Oil Burner" (Selas Sketch No. 1, 1-27-54, J. H.). This nozzle was mounted on 1/2-inch pipe and inserted in an inlet section with the shape of an Archimedes spiral. The exit of the inlet section was connected a 3-inch id precombustion chamber. The static pressure at the end of the inlet section was measured at the wall with an inclined water manometer. Since the total or stagnation pressure was equal to atmospheric pressure it was possible to compute the mass of air induced from this measurement with the aid of Bernoulli's equation, if the assumption of a uniform velocity distribution was made. The mass of air was maintained constant and determined with a rotameter to be 139 SCFH. The experimental data are shown in Table II and the results are plotted in Fig. 8.



TWO-DIMENSIONAL RADIANT BURNER
FIGURE 7



DISTANCE OF NOZZLE FROM END OF INLET SECTION-inches
IN IS POSITIVE
1
1/2
0
-1/2
-1
-1-1/2
4
8
12
16
20
24
28
RATIO OF INDUCED AIR TO PRIMARY AIR-lb/lb
EFFECT OF NOZZLE POSITION ON
RATIO OF INDUCED AIR TO PRIMARY AIR
ARCHIMEDES SPIRAL CROSS-SECTION INDUCED
AIR CONVERGING INLET—MINIMUM DIAMETER-3 IN.
PRIMARY AIR MASS FLOW RATE—26.18 lb/hr
NO FUEL

FIGURE 8

TABLE II

EFFECT OF FUEL NOZZLE LOCATION ON AMOUNT OF AIR INDUCED

Fuel Nozzle = "Bloom" nozzle
Atomizing Air Flow = 139 SCFH

Distance of Fuel Nozzle From End of Converging Inlet Section (in inches induced air flows in + direction)	Static Pressure at End of Inlet (inches of water, gage)	lb induced air / lb driving air
0	-0.54	22.77
+1/2	-0.49	21.73
+1	-0.44	20.53
-1/2	-0.55	23.03
-1	-0.54	22.77
-1.5	-0.50	21.95

These data indicate that the location of the fuel nozzle is not too significant but the optimum location for the burners tested is 1/2-inch before the end of the inlet section.

3. Pressure Drop Downstream of Inlet Nozzle. That the amount of air induced is a function of the pressure drop downstream of the inlet was illustrated in Table I where, for a given geometrical system, the amount of air induced was increased by a factor of 4 when no burning occurred. The pressure drop through the precombustion chamber is the sum of the pressure losses due to burning and these caused by the baffle at the exit.

Ideally, the pressure loss due to burning can be considered equivalent to the internal drag divided by the burner cross-sectional area which, for a drag coefficient equal to 1, is equal to $1/2\rho v^2$, where ρ and v are the density and velocity at the burner inlet. However, the internal drag coefficient is not always equal to 1 since variations in mass velocity result in different burning conditions. The pressure drop due to burning for the precombustion chambers used at mass velocities up to 50 lb/sec/sq ft has been determined in a previous investigation.[1]

The pressure drop due to the baffle or burner tip must be considered in addition to the pressure drop due to burning in the precombustion chamber. The pressure drop through a Selas burner tip (Selas Part No. 31177-07) at different flow rates is shown in Fig. 9, which also includes the results of tests to determine the effectiveness of this part as an ejector. Figure 9 includes plots of driving air flow rate, induced air flow rate, and induced pressure as a function of driving air pressure. As previously indicated in Table I, this part is not very effective as an ejector since 20 lb of driving air are required to induce 1 lb of induced air. It is realized, of course, that this part was not intended to be operated as an ejector.

Experiments to be discussed later indicated that higher cup temperatures were obtained when the burner tip was replaced by a flat baffle. Later experiments indicated that the exit of the precombustion chamber should have rounded corners. However, before these latter experiments were completed, other tests were performed using flat baffles and square corners in order to determine the pressure loss caused by the baffle. The results of these experiments with cylindrical tubes (inlet and exit have square corners) are shown in Fig. 10 and 11. Figure 10 shows the effect of pressure drop on the ratio of total air to primary air for various ratios of tube diameter to nozzle orifice diameter. To nozzle orifice was used as the source of driving air.

Figure 11 presents a correlation of pressure drop across the baffle as a function of the mass rate of air through the tube, tube diameter, and baffle spacing. The data were closely correlated by the equation

$$\left(\frac{d}{D_T}\right)^{2.2} = \left(k\frac{W_t^{\,2}}{\Delta p}\right)\left(\frac{1}{D_T^{\,4}}\right) \quad ,$$

under isothermal conditions where

$$\left(\frac{W_T^{\,2}}{\Delta p}\right)\left(\frac{1}{D_T^{\,4}}\right)$$

[1] Weir, Alexander, Jr., Ind Eng Chem 45, 1637, (1953).

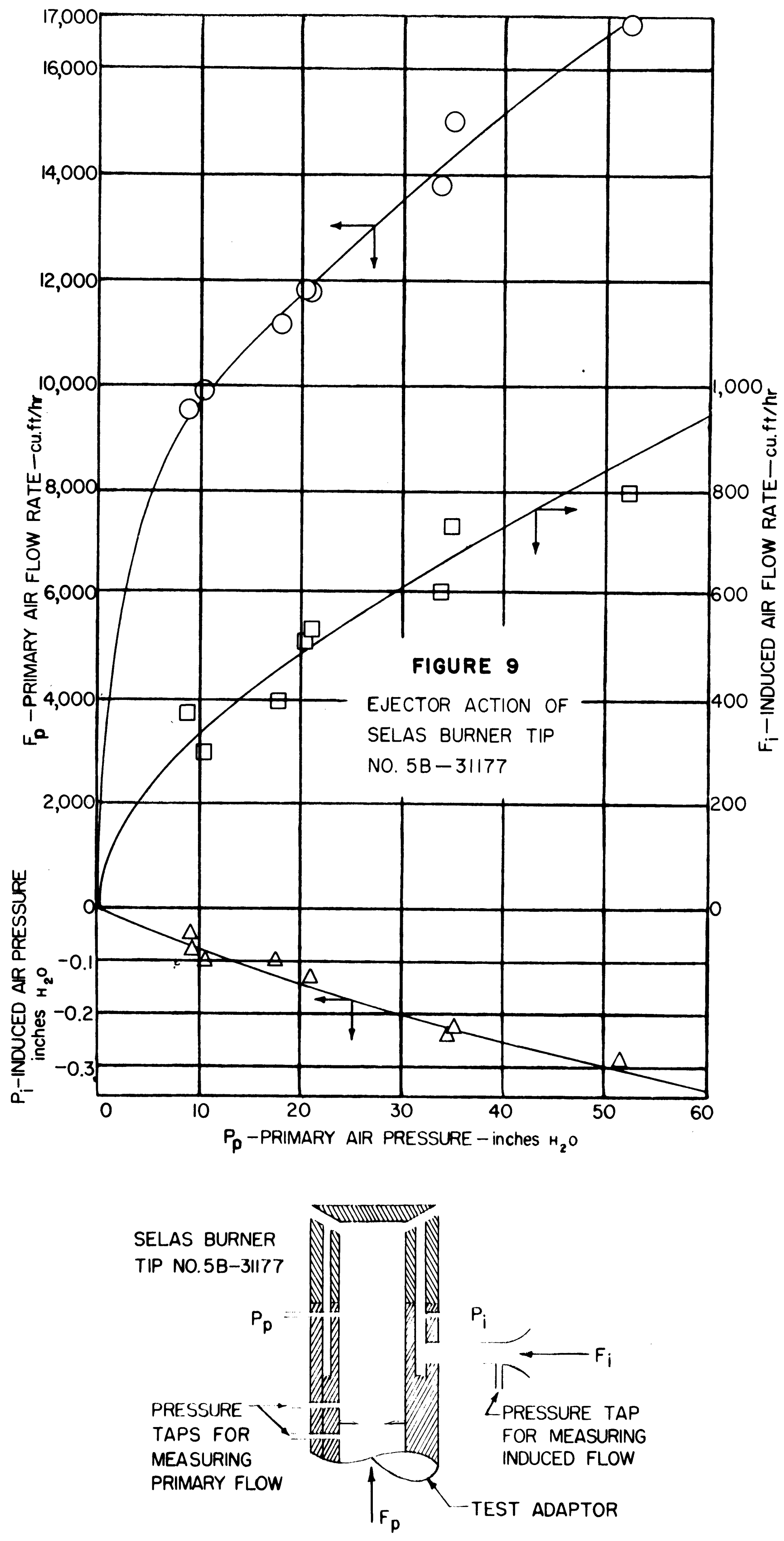


FIGURE 9

EJECTOR ACTION OF SELAS BURNER TIP NO. 5B—31177

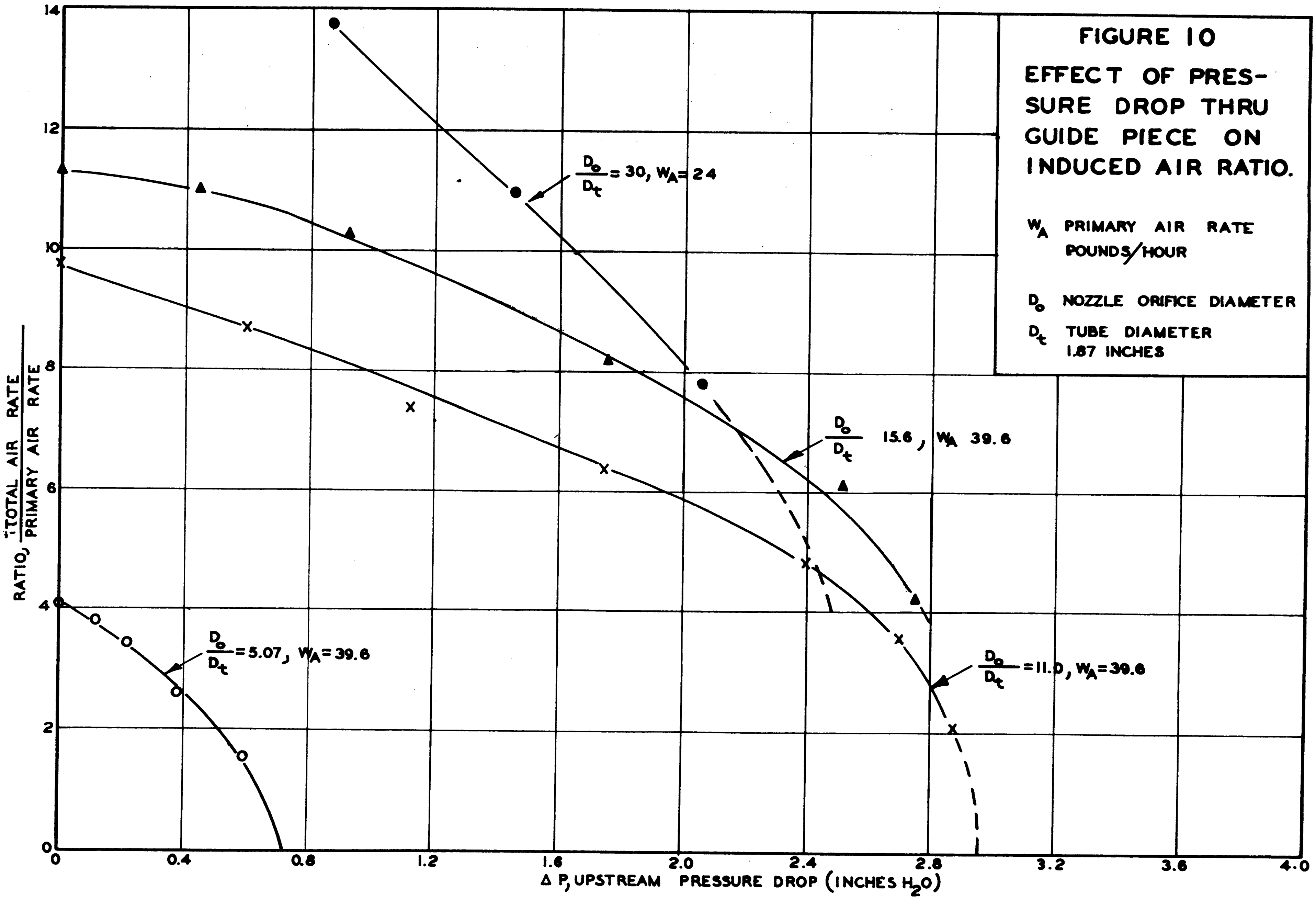


FIGURE 10

EFFECT OF PRESSURE DROP THRU GUIDE PIECE ON INDUCED AIR RATIO.

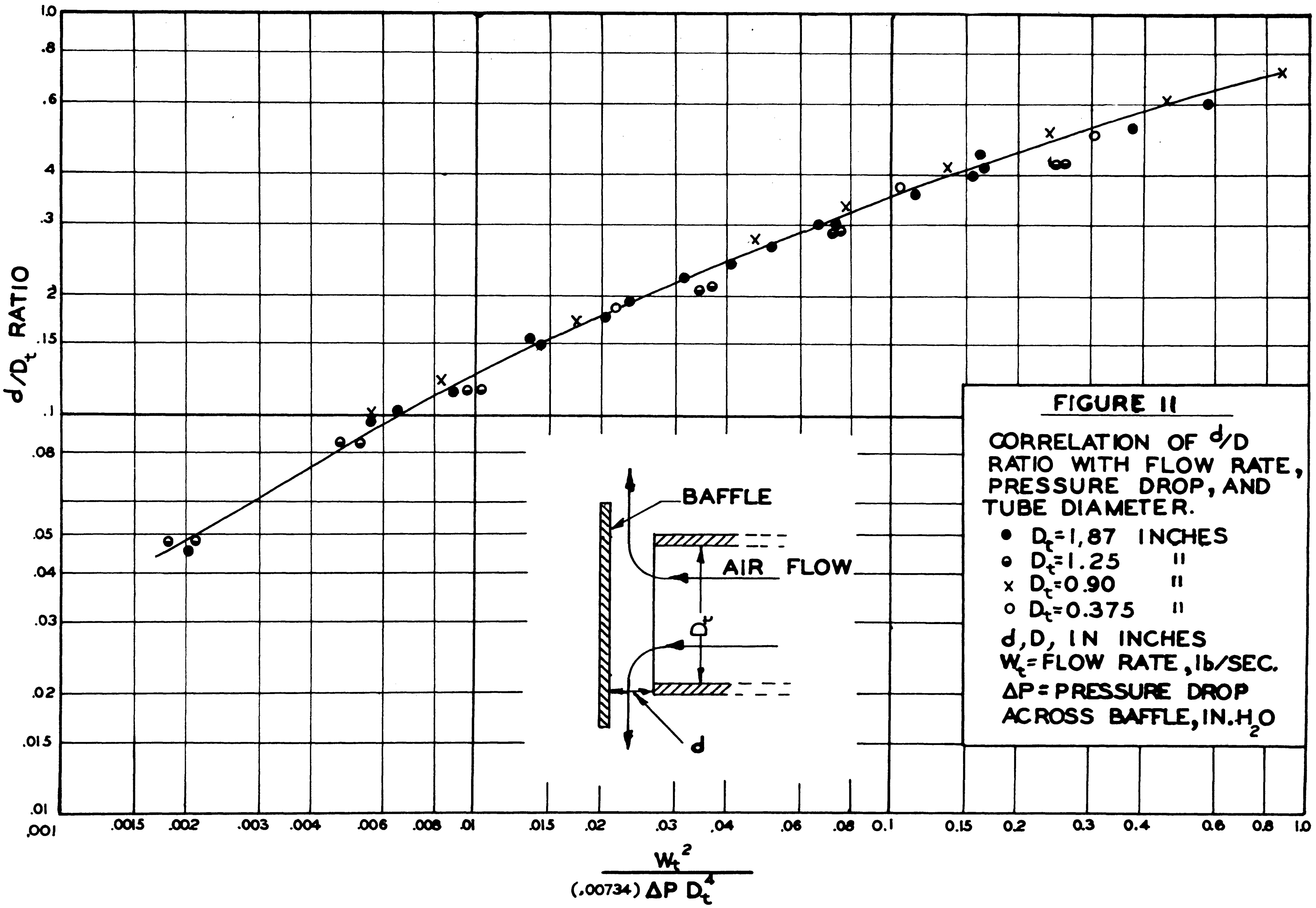


FIGURE 11
CORRELATION OF d/D RATIO WITH FLOW RATE, PRESSURE DROP, AND TUBE DIAMETER.
● D_t=1.87 INCHES
⊖ D_t=1.25 "
x D_t=0.90 "
o D_t=0.375 "
d, D, IN INCHES
W_t= FLOW RATE, lb/SEC.
ΔP = PRESSURE DROP ACROSS BAFFLE, IN. H_2O
BAFFLE
AIR FLOW
D_t
d
d/D_t RATIO
$\frac{W_t^2}{(.00734)\,\Delta P\, D_t^4}$
1.0
.8
.6
.4
.3
.2
.15
.1
.08
.05
.04
.03
.02
.015
.01
.001
.0015
.002
.003
.004
.006
.008
.01
.015
.02
.03
.04
.06
.08
0.1
0.15
0.2
0.3
0.4
0.6
0.8
1.0

represents the number of velocity heads lost thru the baffle (i.e., velocity head in the tube divided by pressure head across the baffle), where

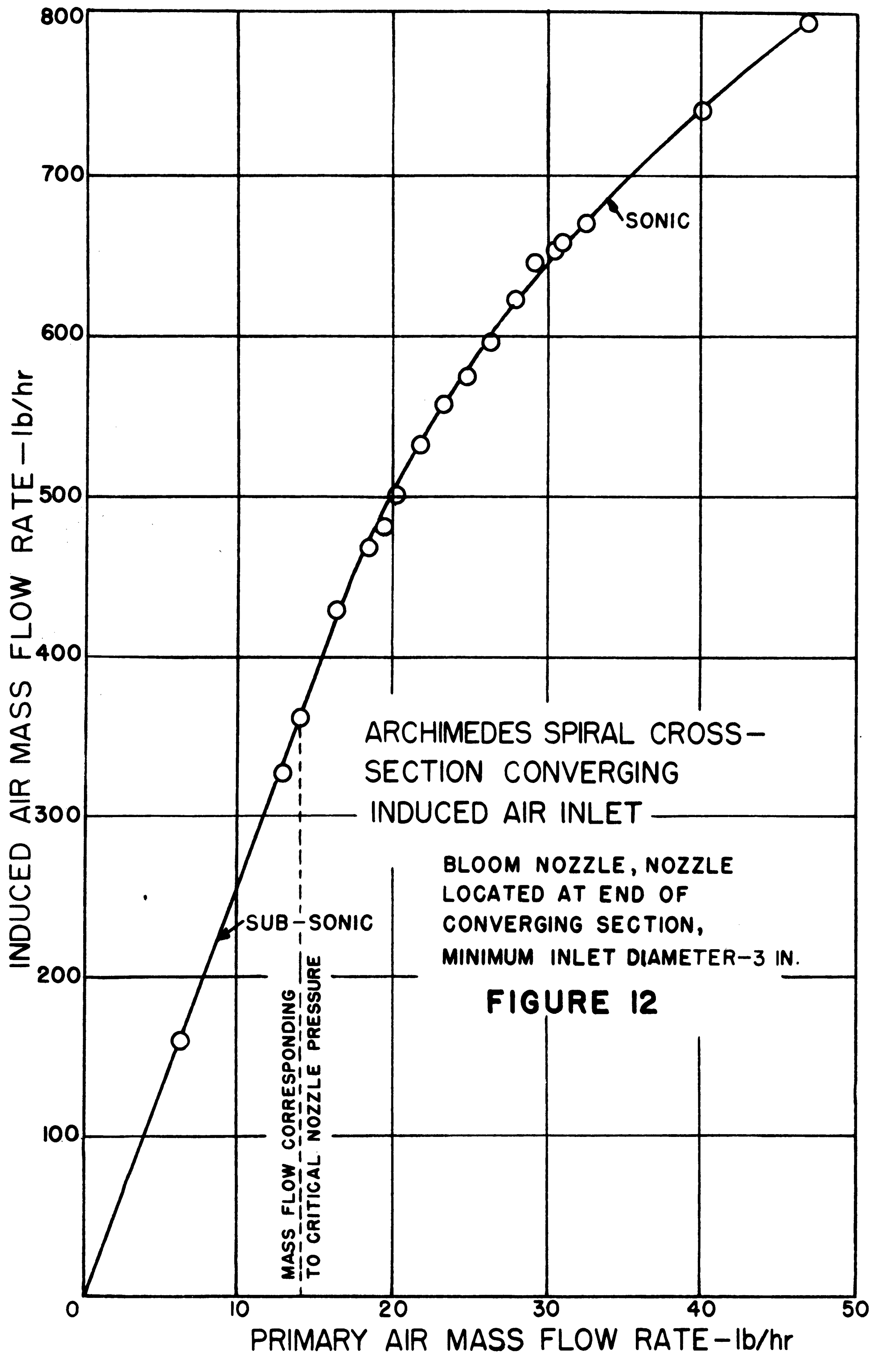
k = $f(\rho)$
W_t = total air flow, lb mass/sec
ΔP = pressure drop across baffle, inches of water
d = distance of baffle from end of tube, inches
D_t = tube diameter, inches
ρ = air density, lb mass/cu ft (.0734)

Thus, while experimental data are available for the pressure drop due to burning in precombustion chambers with no baffles, and the pressure drop due to baffles placed at the exit of square-cornered tubes, additional experimental data on the affect of pressure loss on the amount of air induced would be desirable.

4. Mass and Velocity of Driving Fluid. Figures 4 and 5 indicated the difference in flow pattern issuing from the nozzle when the velocity was subsonic or supersonic. For any mass flow it is possible for the exit velocity to be subsonic or supersonic depending upon the exit density and area of the nozzle. For a given nozzle, however, the exit velocity will be supersonic when the driving pressure is 1.895 times greater than the exit pressure. Since sonic velocity is established in the throat of the nozzle it possible to compute the mass flow which will pass through the nozzle at any pressure greater than the critical pressure, or in other words, the driving air pressure and driving air mass flow rate are no independent variables.

Some experiments were performed with the "Bloom" nozzle in an Archimedes spiral-converging inlet. The apparatus was that used to obtain the data in the previously presented Table II with the nozzle being located at the end of the converging section. The results of these experiments are presented in Fig. 12 and 13. Figure 12, which is a plot of the primary air flow rate versus the ratio of induced air to primary air, indicates that in the supersonic exit velocity region a straight line is obtained, the induced air to primary air ratio decreasing as the mass rate of driving air is increased. However, at the same primary air-flow rate, the ratio of induced to primary air is greater when the exit velocity is supersonic than when the exit velocity is subsonic. All of these experiments were conducted under nonburning conditions.

Additional experiments were performed with the cylindrical tube inlet and different diameter driving air orifices. The results of these experiments, which were performed primarily with subsonic driving velocities, are indicated in Fig. 14.



INDUCED AIR MASS FLOW RATE – lb/hr
800
700
600
500
400
300
200
100
0
10
20
30
40
50
PRIMARY AIR MASS FLOW RATE – lb/hr
SONIC
SUB – SONIC
MASS FLOW CORRESPONDING TO CRITICAL NOZZLE PRESSURE
ARCHIMEDES SPIRAL CROSS-SECTION CONVERGING INDUCED AIR INLET
BLOOM NOZZLE, NOZZLE LOCATED AT END OF CONVERGING SECTION, MINIMUM INLET DIAMETER–3 IN.

FIGURE 12

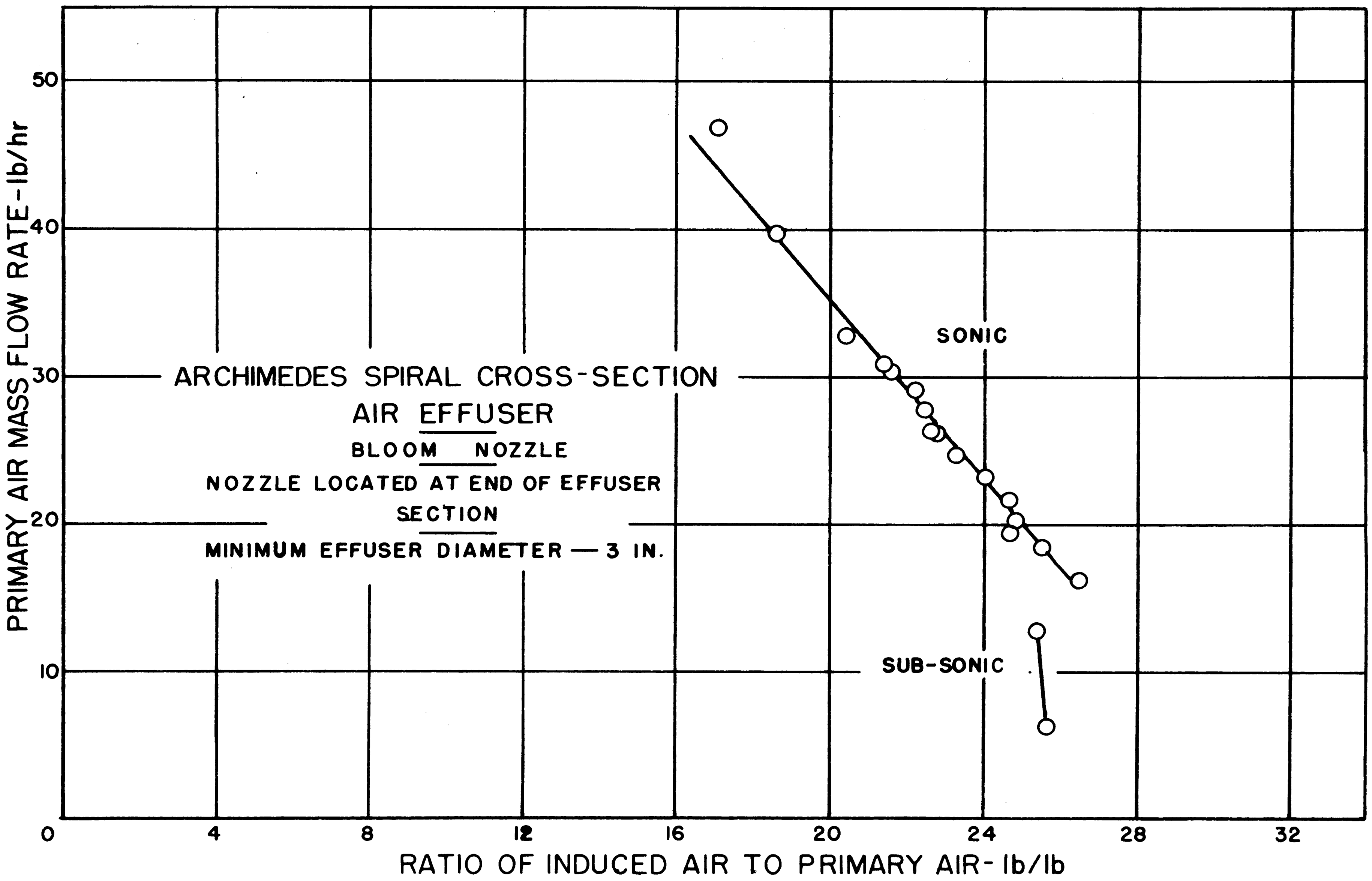

ARCHIMEDES SPIRAL CROSS-SECTION
AIR EFFUSER
BLOOM NOZZLE
NOZZLE LOCATED AT END OF EFFUSER
SECTION
MINIMUM EFFUSER DIAMETER — 3 IN.
SONIC
SUB-SONIC
PRIMARY AIR MASS FLOW RATE - lb/hr
RATIO OF INDUCED AIR TO PRIMARY AIR - lb/lb
0
4
8
12
16
20
24
28
32
10
20
30
40
50

FIGURE 13

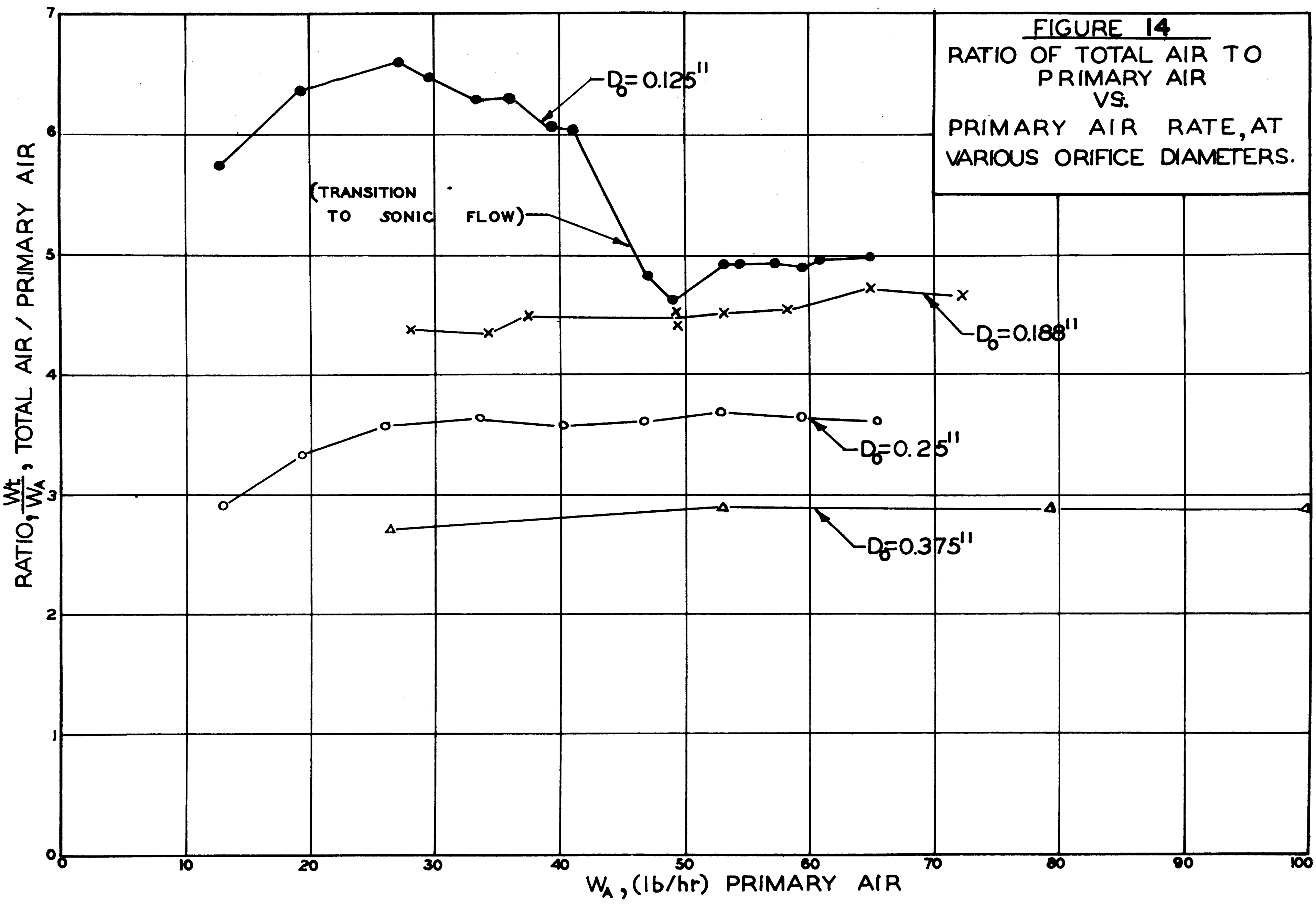


FIGURE 14

RATIO OF TOTAL AIR TO PRIMARY AIR VS. PRIMARY AIR RATE, AT VARIOUS ORIFICE DIAMETERS.

For a given fuel nozzle and inlet section the experimental data thus far obtained indicate that the ratio of induced to primary air decrease as the mass flow rate and pressure of the driving air are increased.

However, as Fig. 13 indicates, the exit area or diameter of the fuel nozzle in a given inlet section is an important variable in determining the amount of air induced. In orther words, the ratio of the mass of primary air to the mass of induced air is a function of the ratio of the exit area of the fuel nozzle to the total area, as well as a function of the velocities and pressures of the included and primary air flows. By using the conservation of mass, momentum, and energy, Morrison[2] was able to eliminate the pressure terms and obtain a general relationship involving the area and mass ratios, temperatures and Mach numbers of the induced and primary air streams in terms of the downstream Mach number. This relation is

$$\frac{\left[M_D \frac{A_D}{A_i}\frac{a_i}{a_D} + \frac{A_i}{A_2} M_i\right]^2 \left[\frac{m_D}{m_2}\frac{T_{oD}}{T_{oi}} + \frac{M_i}{M_2}\right]\frac{T_{oi}}{T_i}}{\left[1 + \gamma \frac{A_D}{A_2} M_D^2 + \gamma \frac{A_i}{A_2} M_i^2\right]^2} = \frac{M_2^2\left(1+\frac{\gamma-1}{2}M_2^2\right)}{\left(1+\gamma M_2^2\right)^2} = \left(\phi(M_2)\right) ,$$

in which

A_D = area of driver nozzle
A_i = area of induced flow at driver discharge
A_2 = area of mixing duct
a_D = speed of sound in driving air flow
a_i = speed of sound in induced flow
M_D = Mach number of driving gas flow
M_i = Mach number of induced gas flow
M_2 = Mach number of total mixed flow
m_D = mass flow of driver gas
m_i = mass flow of induced gas
m_2 = total mass flow
T_i = static temperature of induced gas flow
T_{oi} = stagnation temperature of induced gas flow
T_{oD} = stagnation temperature of primary gas flow
γ = ratio of heat capacities, for air $\gamma = 1.4$

2 Morrison, R. B., "A Shock Tube Investigation of Detonative Combustion", Ph. D. Thesis, University of Michigan, 1952, and later works.

For one-dimensional, compressible fluid flow, increased values of ϕM occur due to friction (or pressure drop) as well as by the addition of heat to the stream e.g., burning in the precombustion chamber. The maximum value of ϕM occurs at Mach = 1. Thus, the values of mass ratio, area ratio, temperature, and Mach number must be such that the left-hand side of the equation is less than the maximum value of ϕM. Since this equation can be made valid for nonadiabatic flow, it would be possible to apply this equation to the precombustion chamber as well as to the converging inlet section to obtain design curves. For example, if a certain the Btu/hr output of a radiant burner is desired this corresponds to a certain amount of total air flow at a stoichiometric for any other desired fuel air ratio. For a given precombustion chamber, this amount of air would determine M_2, and hence ϕM. Then, either the area ratio or mass ratio could be found which would correspond to this value of ϕM for the available upstream stagnation conditions. Experimental measurements of combustion efficiencies and friction losses are, of course, still required.

B. Fuel Nozzle

1. Internal Nozzle Shape and Dimensions. Past experience indicated that small droplet sizes could be most easily achieved by atomization with either air or steam rather than with fluid pressure nozzles. To aid in understanding the mechanism photographs were obtained of the interaction of air at sonic velocity with a solid jet of liquid, using a two-dimensional nozzle with transparent sides. One typical photograph is shown in Fig. 15, which was obtained using an open shutter camera and a 1 microsecond duration light source. Photographs of this type illustrated the disintegration of the liquid jet, and indicated methods of improving the spray.

One such method of for increasing the percentage of small droplets in the spray was found to be precipitation type nozzle. A drawing of a two-dimensional nozzle of this type may be seen in Fig. 16. In this nozzle, the fuel is atomized; the larger drops are precipitated; and then the precipitated on the flat sides of the nozzle and were carried into the spray without being reatomized. Presumably this difficulty would not occur in a three dimensional nozzle.

Three-dimensional precipitation nozzles, however, are relatively difficult to construct. A simpler type from a fabrication standpoint is an annular atomizing nozzle. In this nozzle the liquid is injected through an annular ring around the jet of air, instead of there being a solid jet of liquid in the center of the air jet. For the same liquid and air rates, the annular liquid injection, because of the thinner cross section of the liquid. A sketch of an annular atomizing nozzle is shown in Fig.17. Preliminary tests of this type of nozzle in radiant burners with ceramic precombustion mass flow which will correspond to any given value of the upstream pressure. For an "annular-atomizing nozzle", it was found that the amount of air passed by this nozzle, without fuel flow, was that predicted by a previously developed correlation by Weir[4]. The comparison of the experimental points with this nozzle with the predicted values is presented in Fig. 23.

In Fig. 24 data obtained with the nozzles using different fuel rates are presented. As may be seen, the introduction of liquid fuel does not appreciably change the amount of air induced by the nozzle.

4. Velocity of the Driving Fluid. No investigation of the effect of driving-fluid velocity on droplet size and distribution has been performed. As previously mentioned, nozzles have been run with subsonic as well as supersonic velocities of air impinging upon the liquid fuel in the nozzle itself. It is believed that further work should be done on this subject.

5. Physical Properties of the Driving Fluid. To date, all of the fuel nozzles have been driven with air as the atomizing fluid. However, for spraying viscous fuels such as "Bunker C", steam might offer some advantages as far as atomization is concerned. Also steam might be the only available or most economical fluid for atomization.

An exploratory experiment in which water as a coarse spray was injected into a radiant precombustion chamber fired with gas showed a cooling effect and a tendency toward instability, but neither was serious. The water-vapor cloud surrounding the of droplets in a steam-atomized spray might present a more serious problem.

In addition, it is believe that water-vapor presence has a deleterious effect on flame-propagation velocities, hence a balance must be made between the better atomization obtained with steam and the decreased combustion performance due to the presence of water vapor.

C. Precombustion Chamber

1. Overall Fuel-air Ratio and Mass Velocity of Air and Fuel. As previously mentioned, experimental data on the 3-inch diameter ceramic combustion chamber has been obtained during a previous investigation.[5] Maximum temperatures were obtained with stoichiometric fuel-air ratios, the temperature increasing as the mass velocity was increased.

---

4 Weir, A., Jr., "Two-and Three-Dimensional Flow of Air Through Square-Edged Sonic Orifices", Ph. D. Thesis, University of Michigan, 1954.

5 Weir, A., Jr., Ind Eng Chem, 45, 1637, 1953.

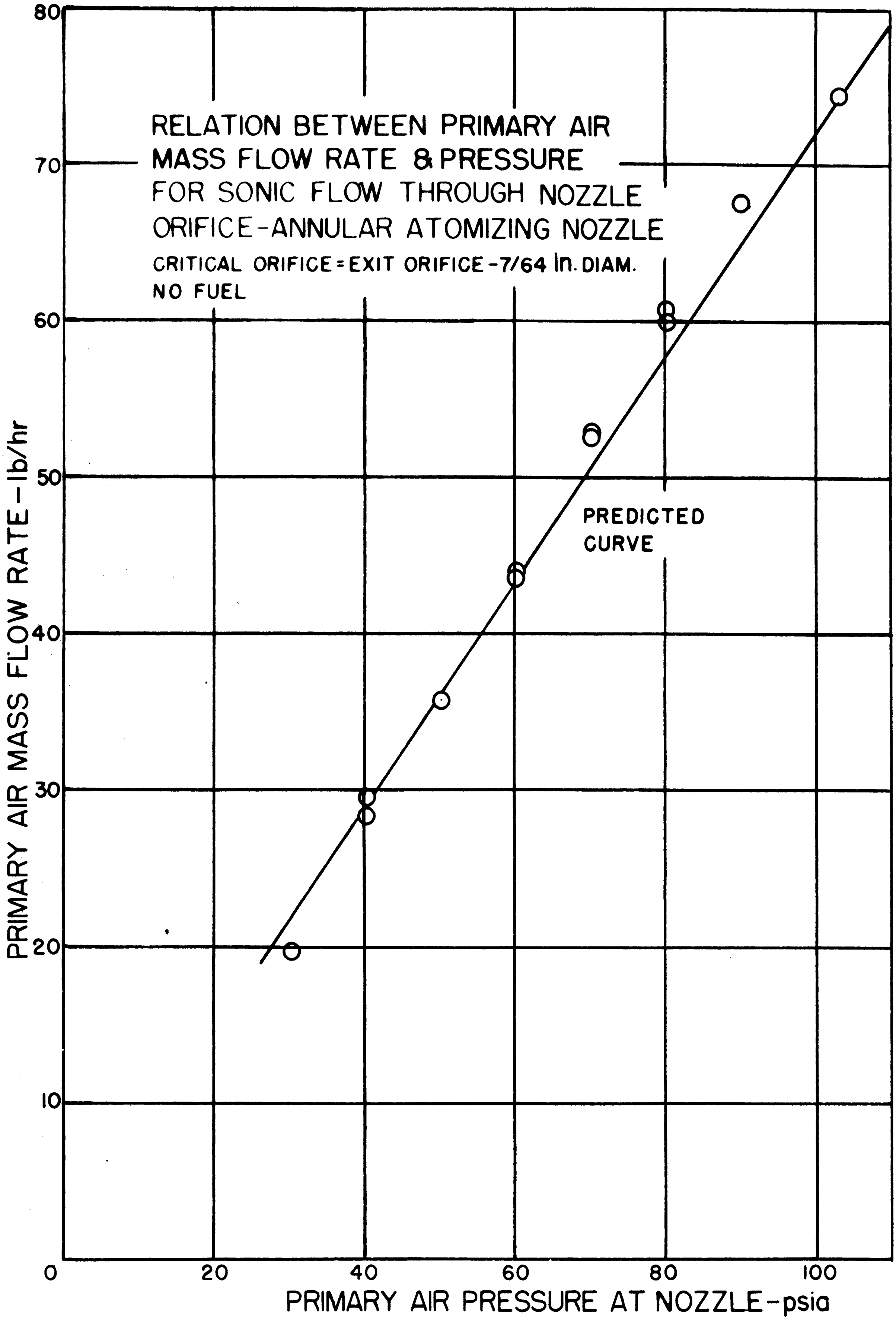

RELATION BETWEEN PRIMARY AIR
MASS FLOW RATE & PRESSURE
FOR SONIC FLOW THROUGH NOZZLE
ORIFICE-ANNULAR ATOMIZING NOZZLE
CRITICAL ORIFICE = EXIT ORIFICE - 7/64 in. DIAM.
NO FUEL
PREDICTED
CURVE
PRIMARY AIR MASS FLOW RATE - lb/hr
PRIMARY AIR PRESSURE AT NOZZLE - psia
0
10
20
30
40
50
60
70
80
100

FIGURE 23

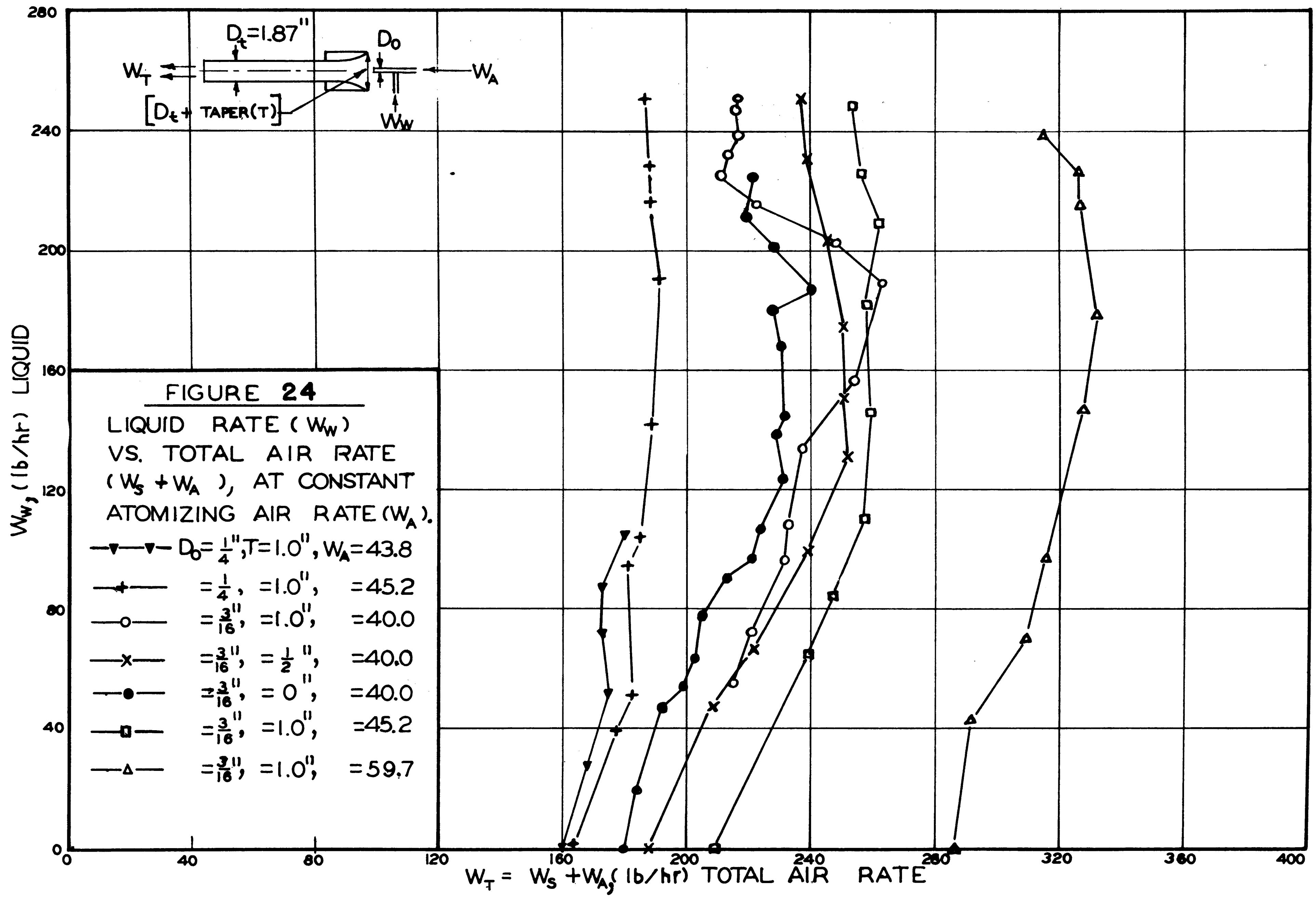


FIGURE 24
LIQUID RATE (W_W)
VS. TOTAL AIR RATE
($W_S + W_A$), AT CONSTANT
ATOMIZING AIR RATE (W_A).
$D_0 = \frac{1}{4}''$, $T = 1.0''$, $W_A = 43.8$
$= \frac{1}{4}$, $= 1.0''$, $= 45.2$
$= \frac{3}{16}''$, $= 1.0''$, $= 40.0$
$= \frac{3}{16}''$, $= \frac{1}{2}''$, $= 40.0$
$= \frac{3}{16}''$, $= 0''$, $= 40.0$
$= \frac{3}{16}''$, $= 1.0''$, $= 45.2$
$= \frac{3}{16}''$, $= 1.0''$, $= 59.7$
$D_t = 1.87''$
D_0
W_T
W_A
W_W
[D_t + TAPER(T)]
W_W, (lb/hr) LIQUID
$W_T = W_S + W_A$, (lb/hr) TOTAL AIR RATE
0
40
80
120
160
200
240
280
320
360
400

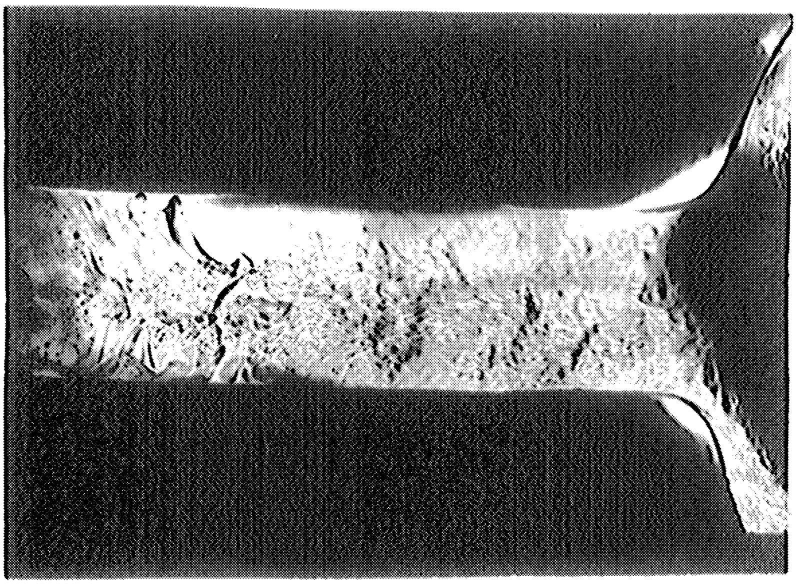
2. Velocity Distribution at Burner Inlet. With the induced air burner, the velocity distribution at the inlet to the precombustion chamber is different than with combustion chambers operated with premixed gaseous fuel and air. In Fig. 25, several schlieren photographs of two-dimensional ceramic precombustion chambers in operation are shown. In the top photograph, gaseous propane is being fed through the fuel nozzle; in the other photographs, No. 4 fuel oil is being used in a "tee" nozzle. As may be seen, a nonuniform velocity distribution at the inlet to the precombustion chamber is obtained.

A quantitative inbestigation of the effect of inlet velocity distribution on combustion-chmaber performance has not bee made.

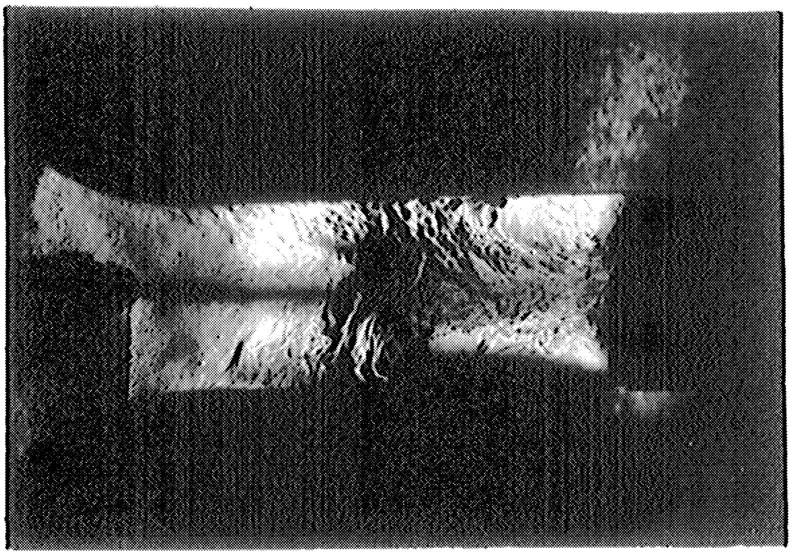
3. Length and Diameter of Combustion Chamber. Most of the experiments were performed with a 3-inch internal diameter precombustion chamber in full-scale radiant burners. Experiments were performed using No. 4 oil to study the effect of length of the combustion chamber on burning. The length was varied from 12.5 to 17 inches. It was found that increasing the length of the combustion chamber beyond 1 foot has a slight, but not appreciable, beneficial effect on burning. At higher mass velocities with propane (ca over 15 lb/sec/sq ft) a length of 15 to 18 inches was previously found to be required. Additional experiments were performed with No. 6 oil in small scale (9-inch diameter cup) radiant burners. In these experiments, longer combustion-chamber lengths were required for No. 6 oil than were required for Nos. 4 or 2 fuel oil.

4. Burner Wall Shape, Surface, and Temperature. Experiments were performed to determine the effect of surface roughness on burning in a 9-inch ID ceramic precombustion chamber. A smooth internal surface was compared to one with beveled edges. The temperatures were not significantly different, but better mixing and combustion occurred with the beveled edge burner as evidenced by the fact that the burner could be run without blow-off on leaner mixtures. Other data obtained on a different project confirm this observation.

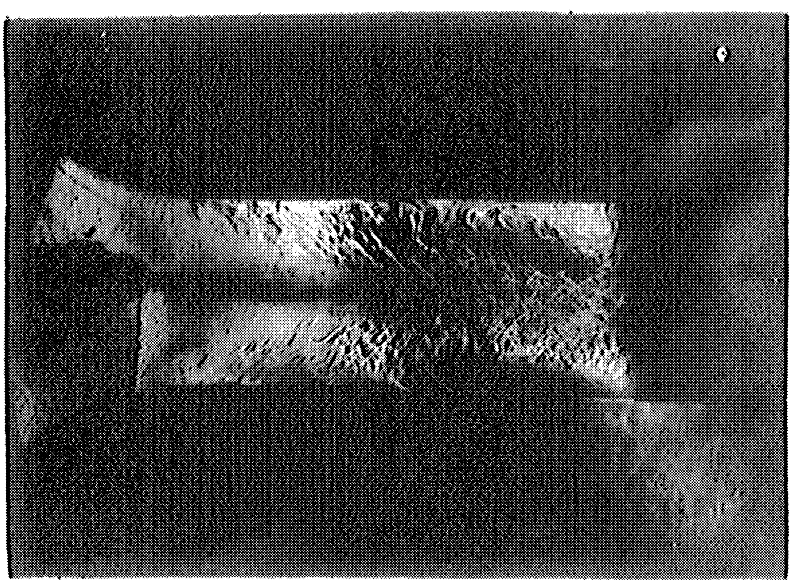
No experiments have been performed to determine quantitatively the effect of the precombustion-chamber wall temperature on the burner cup temperature or on combustion in the precombustion chamber. However, the combustion-chamber exit wall temperature was about 200°F greater than the burner cup temperature with the full scale radiant burners tested to date.



PROPANE



NO. 4 OIL, NOZZLE PRESSURE = 43.3 PSIA



NO. 4 OIL, NOZZLE PRESSURE = 56.4

SCHLIEREN PHOTOGRAPHS OF TWO-DIMENSIONAL BURNER

FIGURE 25

5. Degree of Vapcrization of Fuel and Droplet Size and Distribution. Since considerable backgroundinformation on this subject is available, detailed investigation of the effect of these variables on burner operation will be made when specific burner designs has been decided. As the result of the experience with these problems, it is felt that improved combustion can generally be accomplished by vaporizing any specific fuel, and for any given burner design, smaller droplets are preferable to large droplets.

D. Combustion Chamber Exit and Cup

1. Shape of the Inlet Corner. Original experiments were performed with a two-dimensional, 1/3-scale burner using gaseous propane as fuel. Uing the test sections shown in Fig. 26 schlieren pictures indicated that the "petal" effect noted or radiant burners was due to an area where no flow occurred. This area was first observed in the schlieren pictures obtained with a 10,000-volt spark source shown in Fig. 27.

It has been found that this vacant region is a function of the geometric shape and size of exhaust ports, and the velocity of exhausting gases. The effect of geometric shape was investigated from two standpoints. First, the sharp corners (Fig. 26) which were believed to be diverting the gas flow away from the walls were streamlined; and secondly, the distance (x) at the exhaust outlet was varied. A group of pictures taken with the streamlined corners (Fig. 28) indicated that the gases seemed to flow much better along the ceramic with this design, and that the region of no gas flow increased with increasing distance (x). Complete data were not obtained with these pictures but the mass flow is believed to be between 0.1 and 0.2 lb per minute and the maximum temperature was found to be approximately 2500°F. This temperature was higher than any previously observed values. Schlieren photographs of burners of different shapes are shown in Figs. 29, 30, 31 and 32. The cup temperatures and mass flow rates are given below each photograph.

The fact that the size of this region increased with increasing distances at the exhaust port (Fig. 27) was evidence that it is also a function of velocity. As the distance increased, the exhaust area increased and hence the velocity decreased. A further verification that the gas does not penetrate to the ceramic at low velocities was obtained with other pictures. For example, a comparison of the pictures in Fig. 31, E_7 and E_8, both taken at the same distance but with the mass flow in E_8 about double that in E_7, shows no vacant spot observed in E_8. The increased mass flow in E_8 gave a proportional increase in velocity.

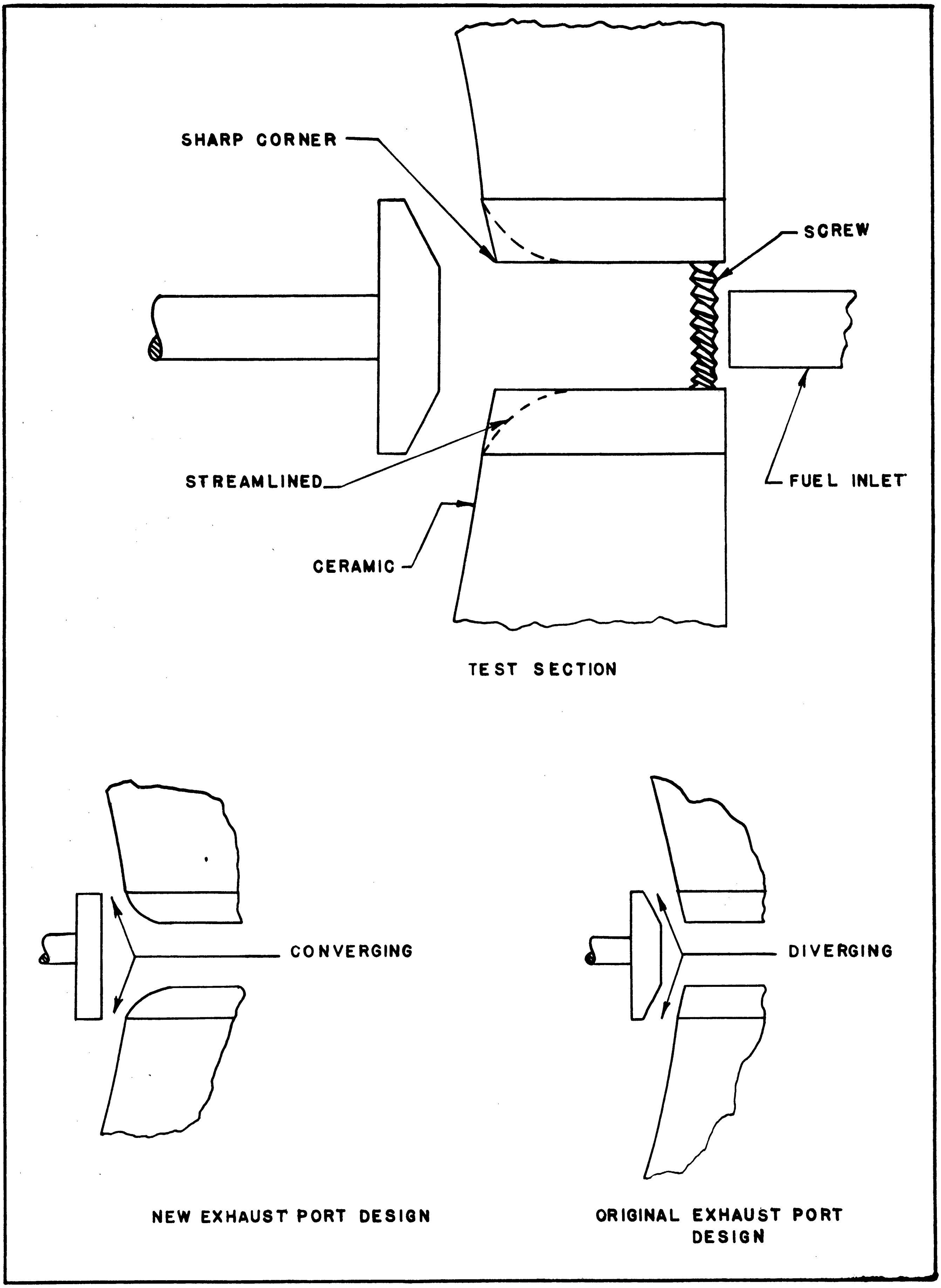
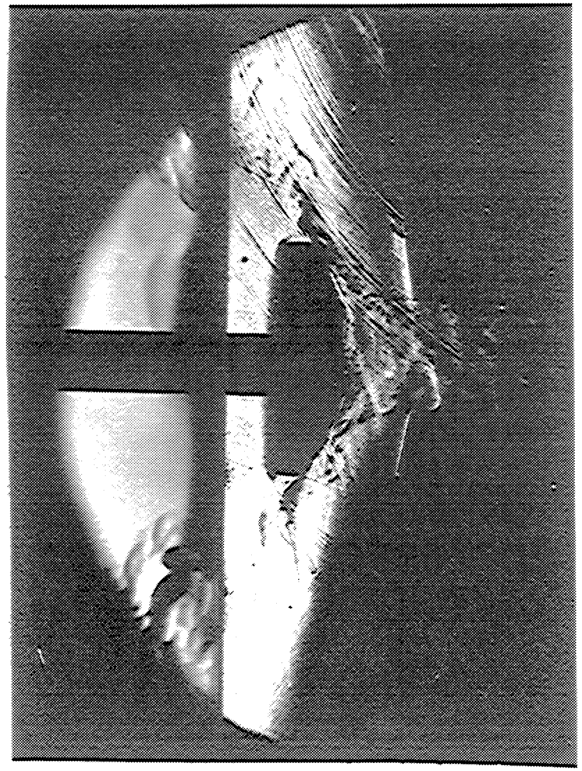

SHARP CORNER
SCREW
FUEL INLET
STREAMLINED
CERAMIC
TEST SECTION
CONVERGING
DIVERGING
NEW EXHAUST PORT DESIGN
ORIGINAL EXHAUST PORT DESIGN

FIGURE 26

A1

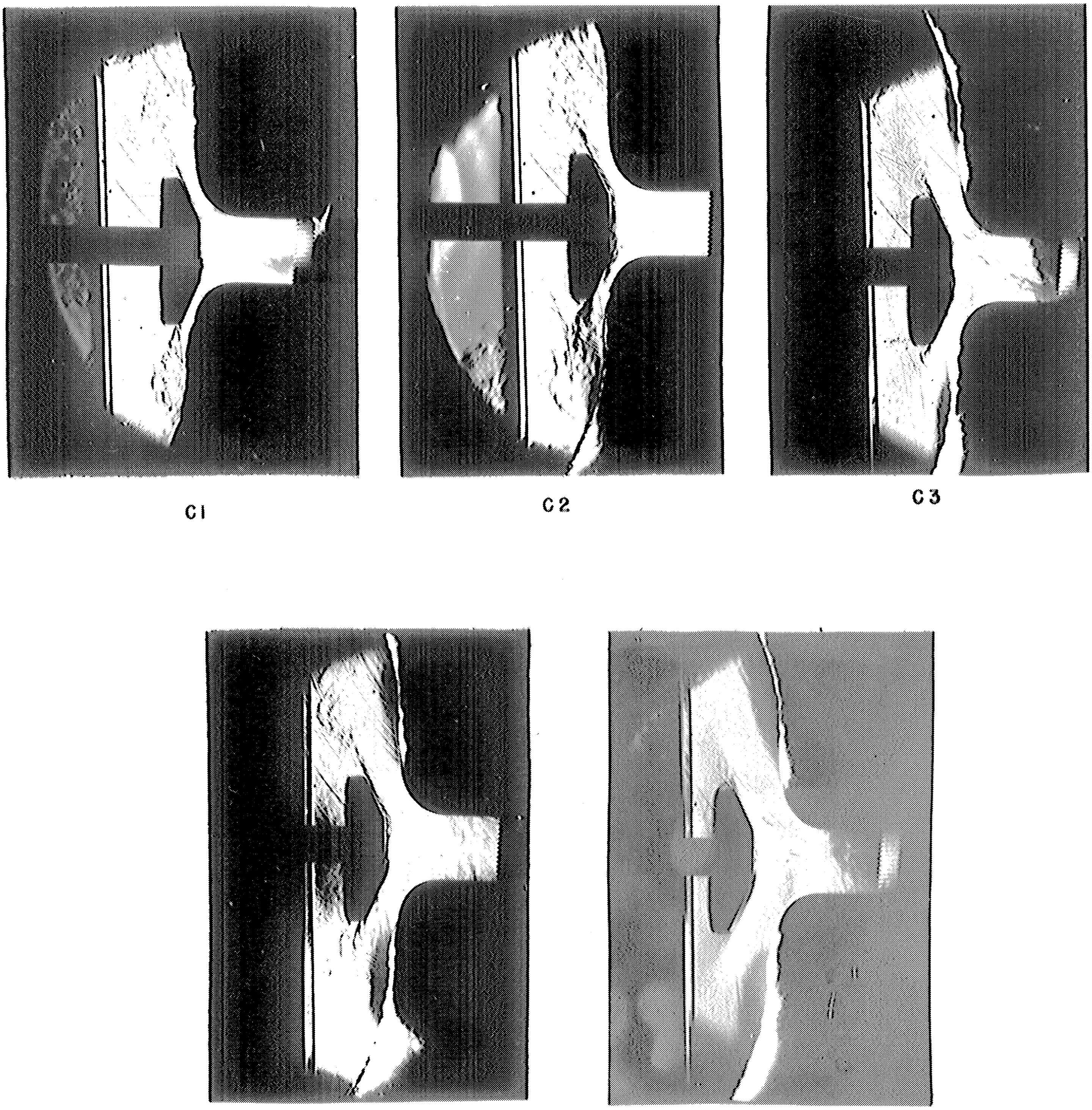
A2



A3

SHARP CORNER, BEVELED BAFFLE

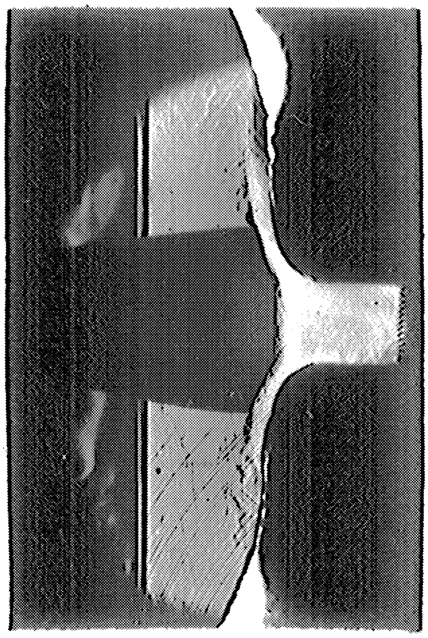
FIGURE 27



MAXIMUM TEMPERATURE (C3 OR C4) APPROXIMATELY 2500° F
MASS FLOW RATE BETWEEN 0.1 AND 0.2 LBS. PER MIN.

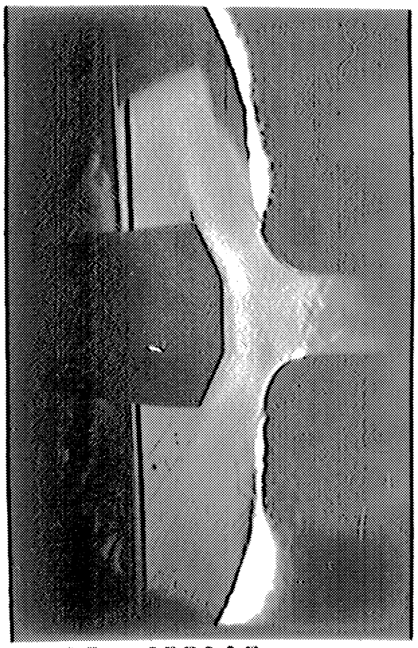
STREAMLINED CORNER, BEVELED BAFFLE

FIGURE 28

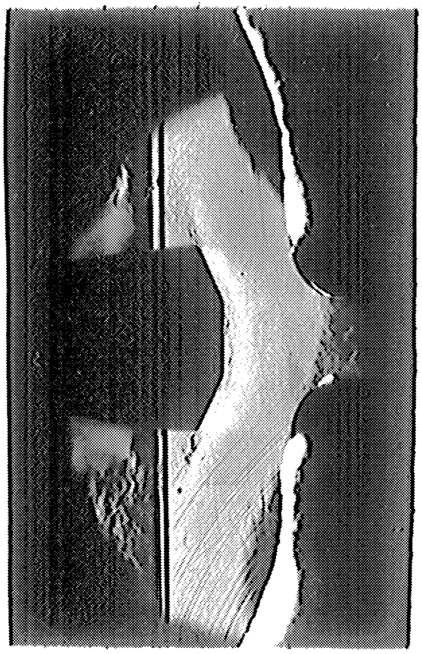


D1 2225° F
0.22 LBS. PER MIN.

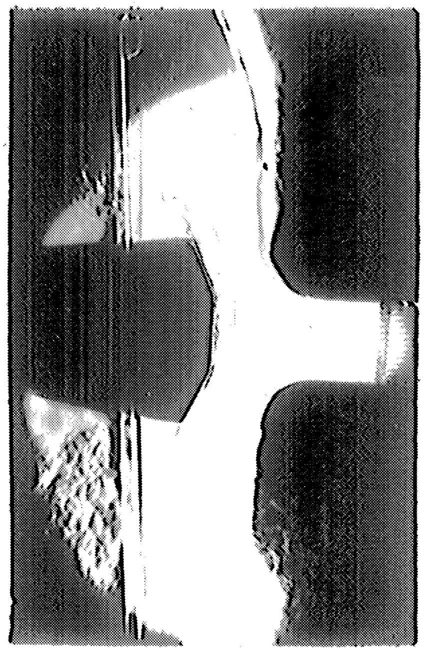
D2 2300° F
0.22 LBS. PER MIN.



D3 2350° F
0.22 LBS. PER MIN.



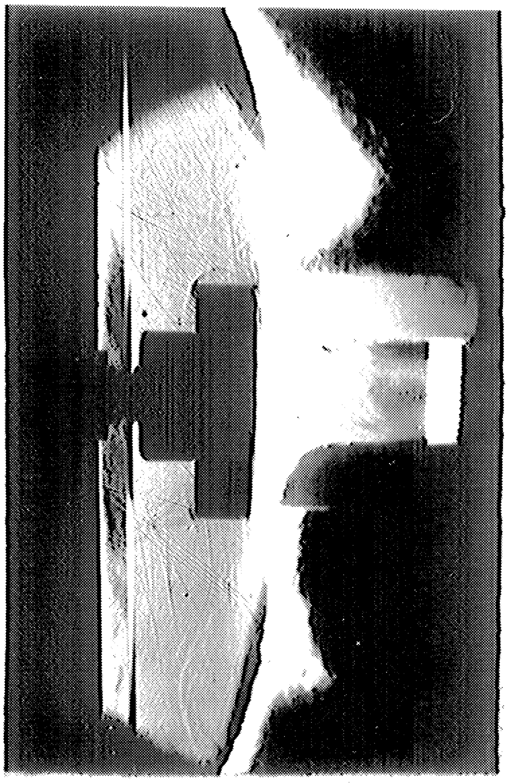
D4 2300° F
0.22 LBS. PER MIN.



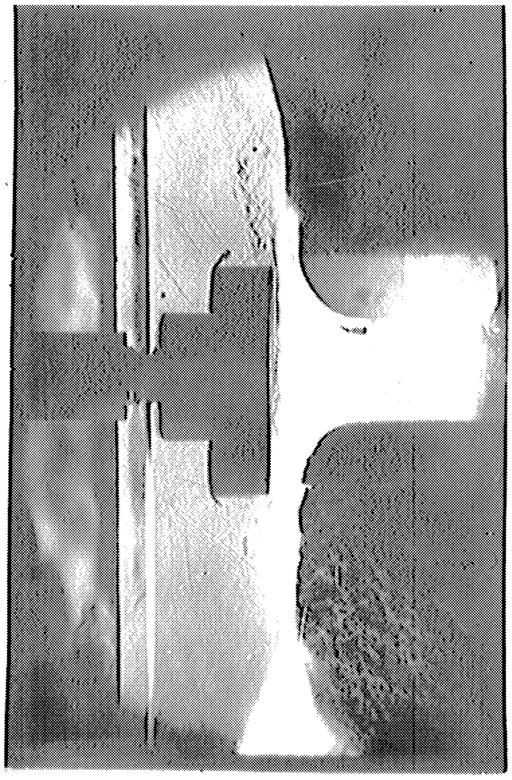
D6 2500° F
0.45 LBS. PER MIN.

STREAMLINED CORNER, LARGE BAFFLE

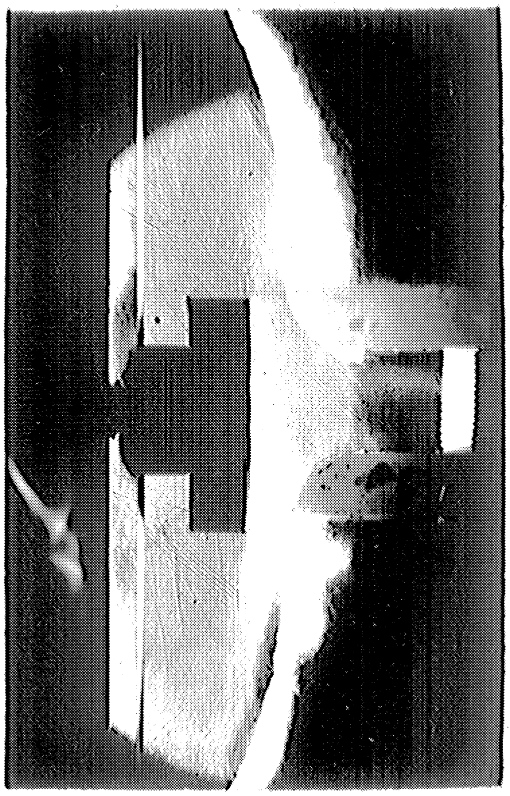
FIGURE 29



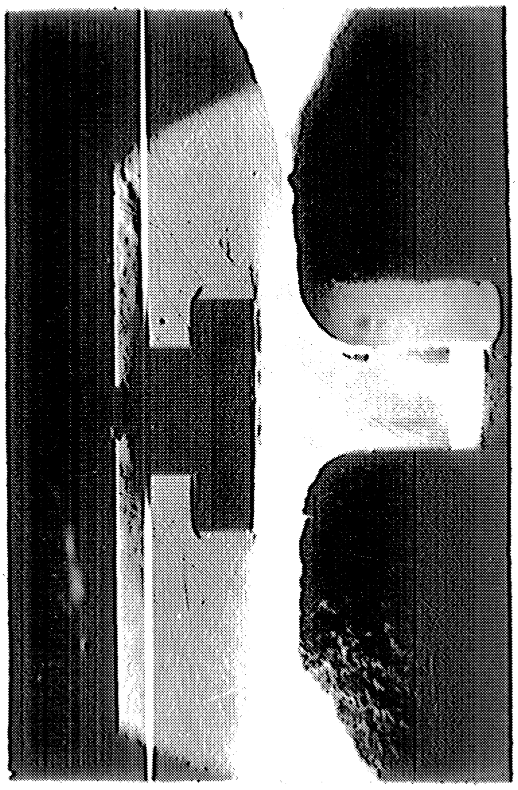
E1 2300° F
0.17 LBS. PER MIN.



E2 2440° F
0.43 LBS. PER MIN.

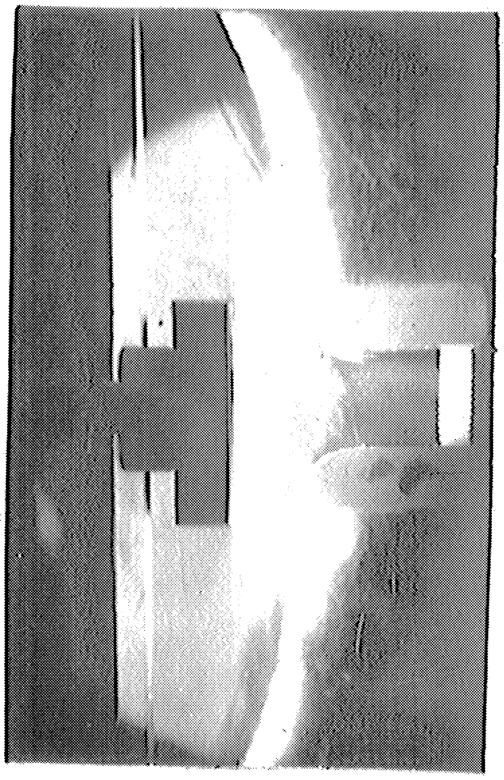


E3 2410° F
0.17 LBS. PER MIN

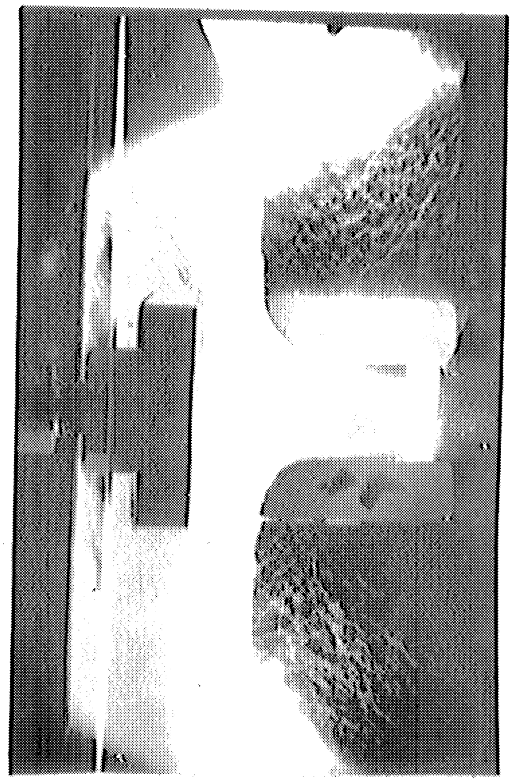


E4 2460° F
0.43 LBS. PER MIN.

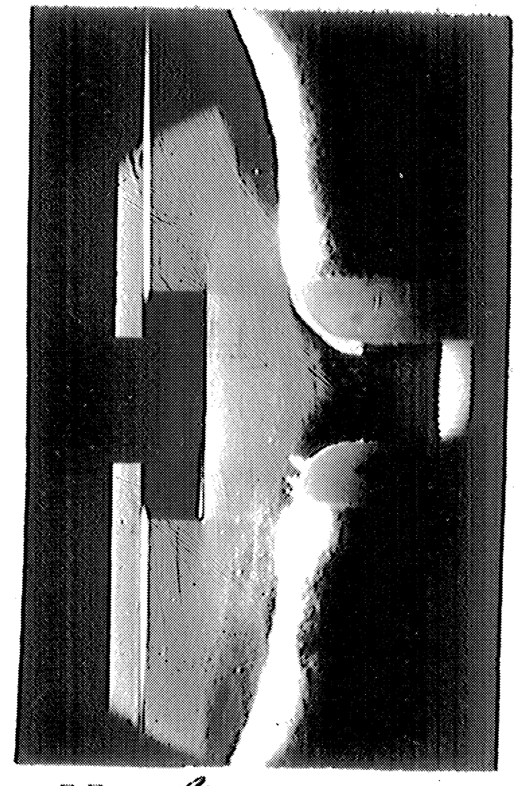
STREAMLINED CORNER, FLAT BAFFLE
FIGURE 30



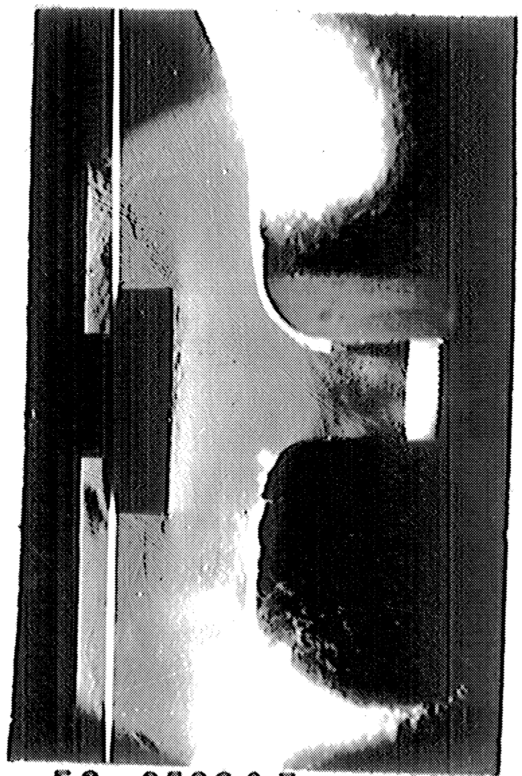
E5 2430 °F
0.17 LBS. PER MIN.



E6 2560 °F
0.43 LBS. PER MIN.



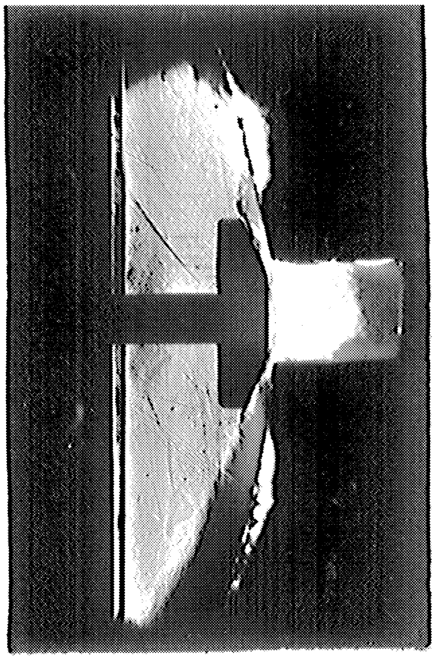
E7 2425 °F
0.17 LBS. PER MIN.



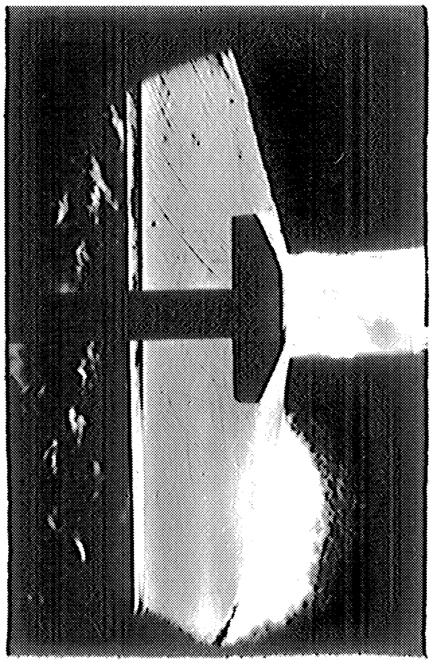
E8 2520 °F
0.43 LBS. PER MIN.

STREAMLINED CORNER, FLAT BAFFLE

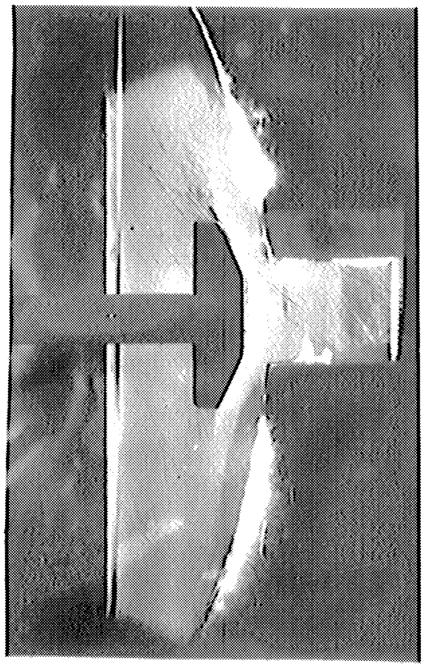
FIGURE 31



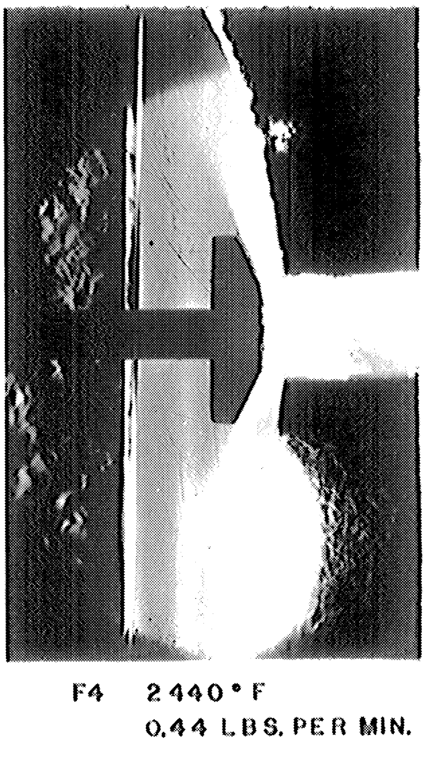
F1 2290°F
0.17 LBS. PER MIN.



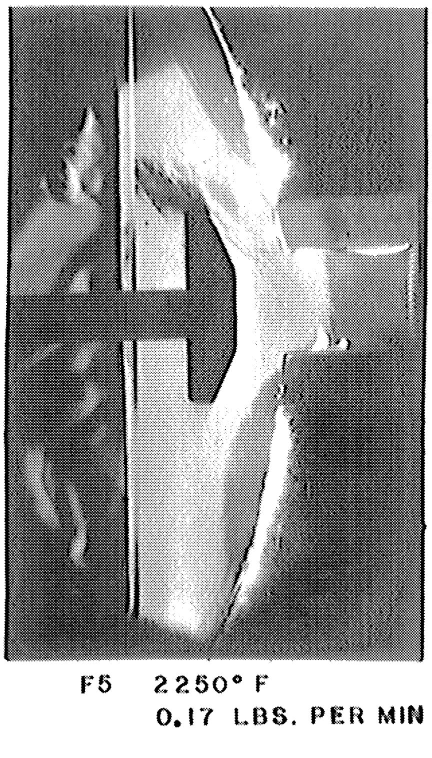
F2 2370°F
0.44 LBS. PER MIN



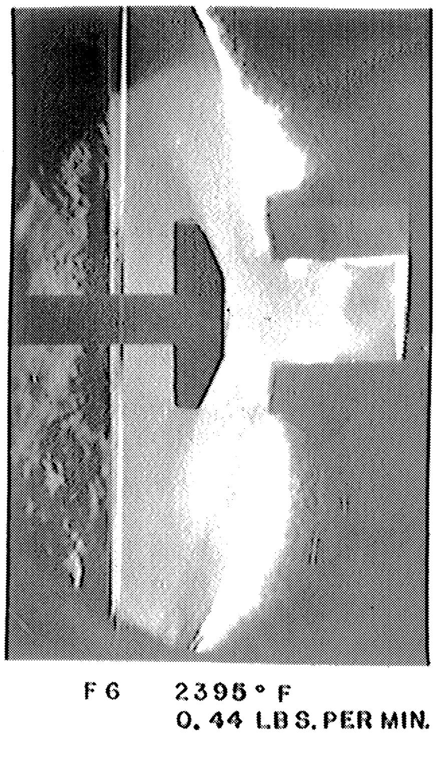
F3 2300°F
0.17 LBS. PER MIN.



F4 2440°F
0.44 LBS. PER MIN.



F5 2250°F
0.17 LBS. PER MIN



F6 2395°F
0.44 LBS. PER MIN.

SHARP CORNER, BEVELED BAFFLE

FIGURE 32

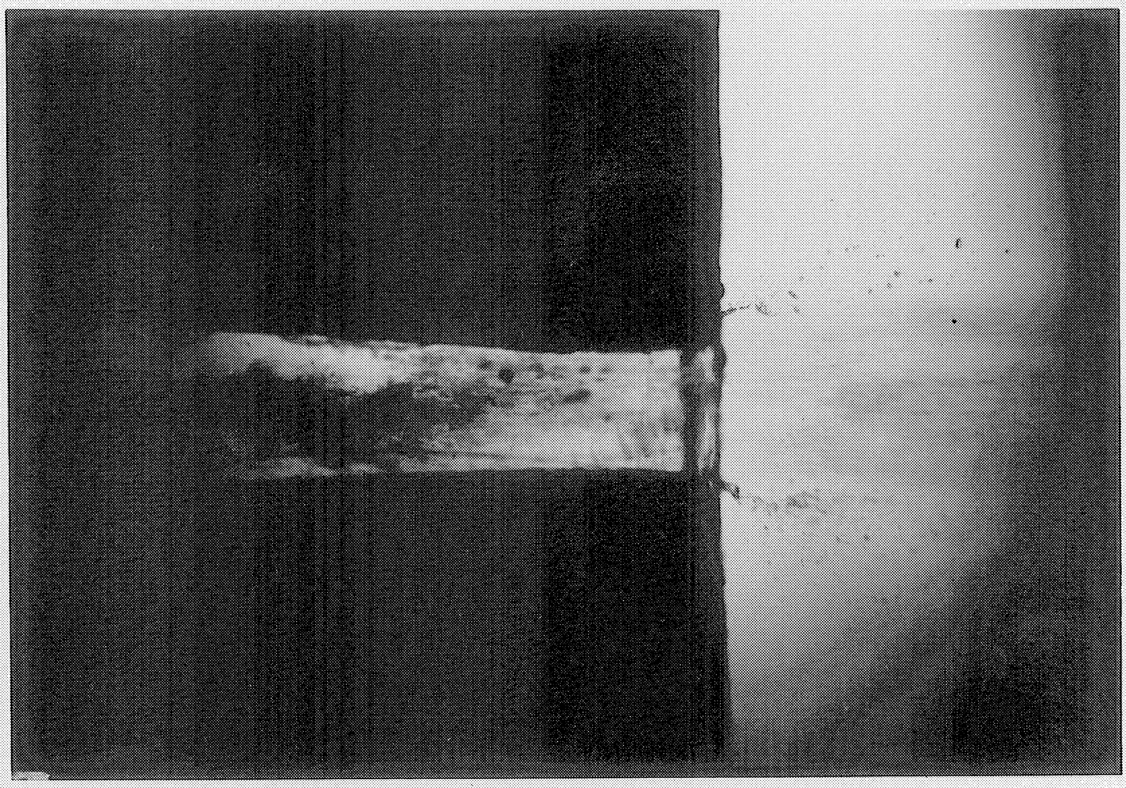


FIGURE 15

PHOTOGRAPH OF TWO-DIMENSIONAL, TWO-PHASE SPRAY NOZZLE. (AIR AND WATER)

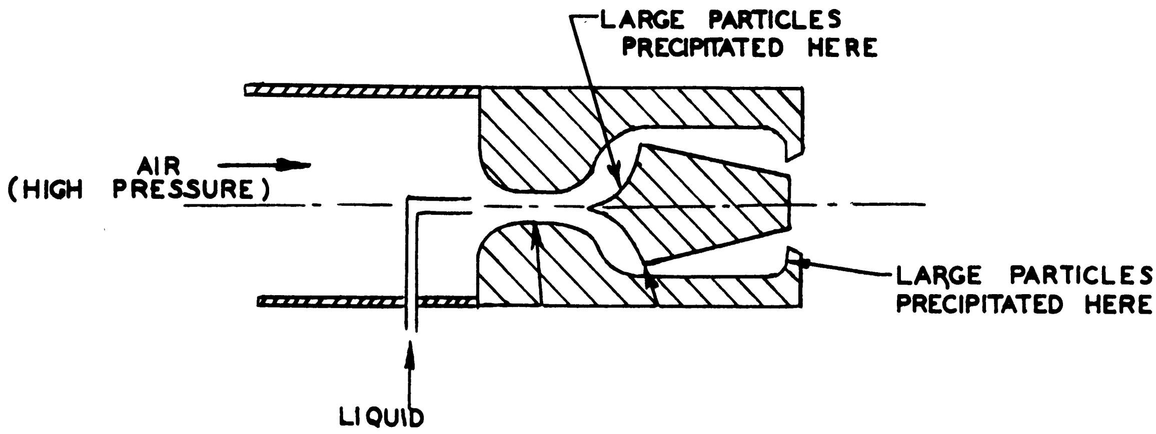


FIGURE 16 – TWO DIMENSIONAL PRECIPITATION NOZZLE

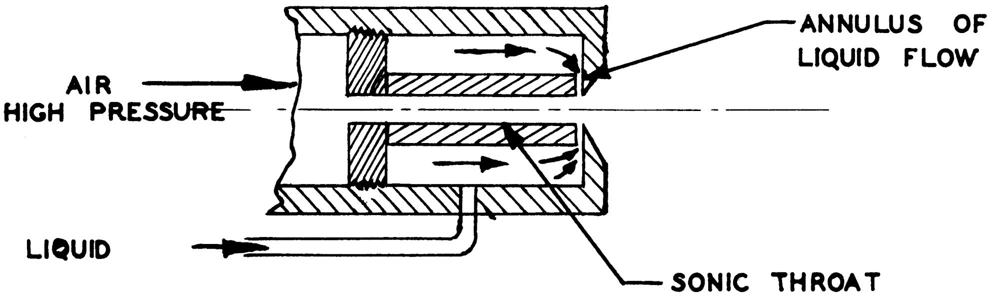


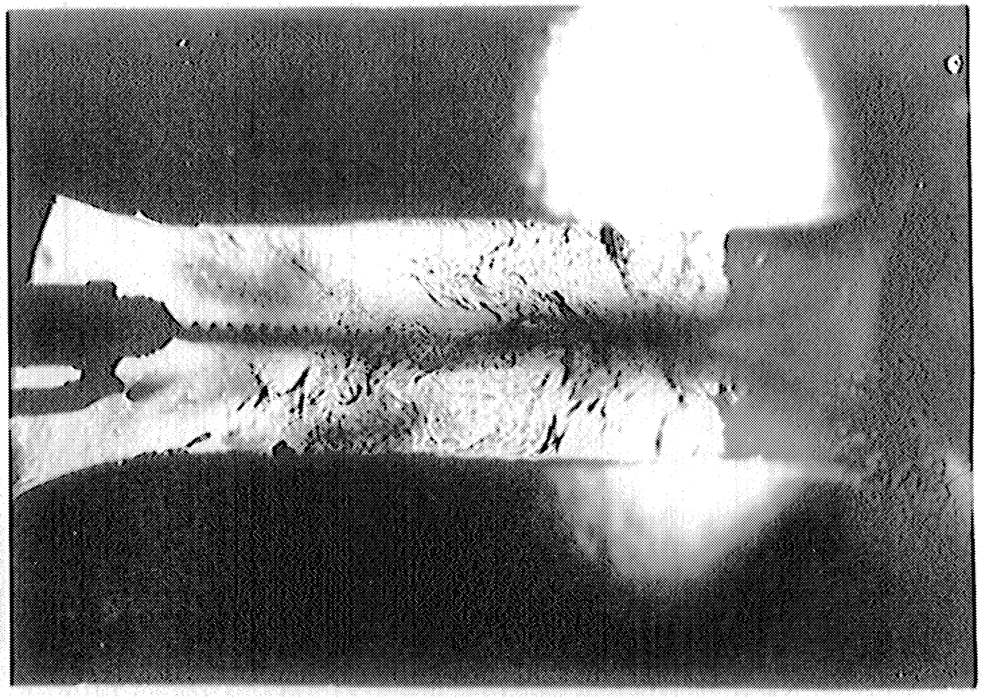
FIGURE 17 – ANNULAR ATOMIZING NOZZLE

chambers did not result in satisfactory operation of the burner, possibly due to a mismatch of nozzle capabilities and burner fuel requirements. Further experiments are to be performed with this type of nozzle.

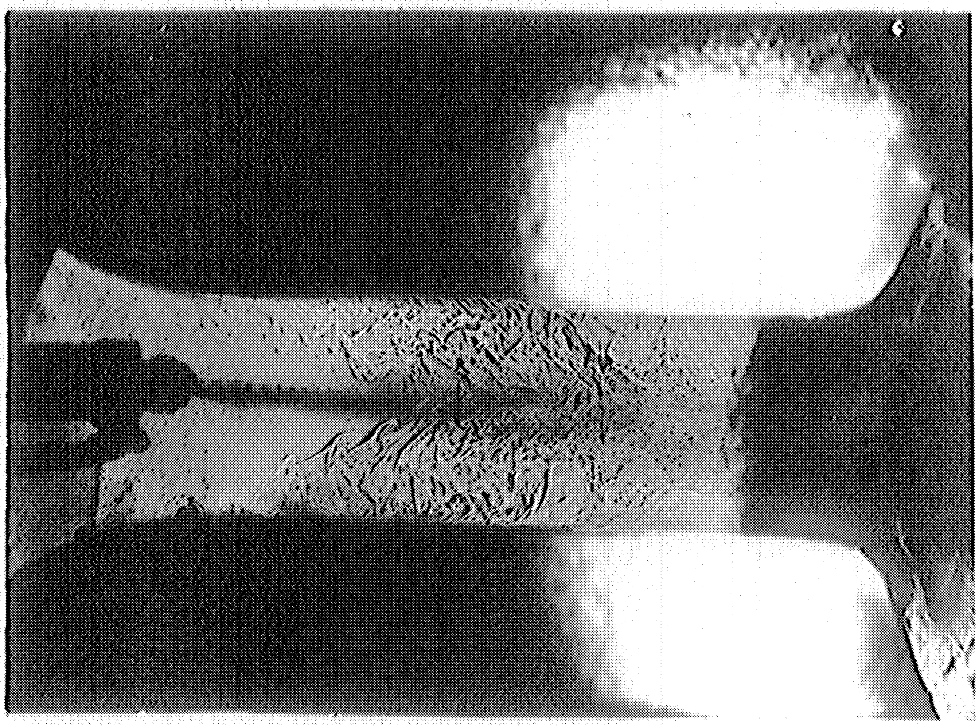
A simple type of nozzle was found to perform satisfactorily with radiant burners in which a ceramic precombustion chamber was used. This consists essentially of a small tubing "tee" with fuel oil being fed into the perpendicular outlet, where it meets a supersonic stream of air. Another variation is that the fuel oil meets a subsonic stream of air, the throat where sonic velocity occurs being at the combined fuel and air exit of the tee. An illustration of the former type in operation may be seen by referring to Fig. 18, which contains schlieren photographs of one of these nozzles using No. 4 fuel oil in a two-dimensional burner with ceramic precombustion chamber. The dimensions of this nozzle are shown in Fig. 19. Complete atomization is not obtained with his nozzle, even though shock and expansion waves exist in the jet. Mach diamonds may be seen in the top half of the photographs, while the solid liquid stream is evident in the lower portion of the jet.

That nozzle geometry and the subsequent droplet distribution, has an important effect on radiant burner performance is indicated in Fig. 20. These data were obtained with a full-scale radiant burner with a ceramic precombustion chamber and an Archimedes spiral-converging inlet. The interrelationship of the amount of air induced, the fuel nozzle geometry, and burner performance is illustrated by these graphs, since at large nozzle-exit diameters, the overall fuel-air ratio is on the fuel-rich side, due to the inability of the nozzle to induce enough air to maintain the overall fuel-air ratio at a stoichiometric value. On the other hand, at lower values of the nozzle exit diameter, the overall fuel-air ratio is on the fuel-lean side. The maximum cup temperature is obtained at a 9/64-inch nozzle-exit diameter, where the overall fuel-air ratio is at a stoichiiometric value. As previously indicated in Fig. 12, as the mass rate of driving air increases, the ratio of induced to primary air decreases for any given fuel nozzle and converging inlet combination. Thus, increasing the amount of primary air in the fuel nozzle will not necessarily increase the overall fuel-air ratio. Instead, the ratio of atomizing air to fuel in a nozzle would have to be increased to maintain the same overall fuel-air ratio in a given system as the fuel rate is increased. The dimensions of the nozzles used for the experiments in Fig. 20 are indicated in Fig. 21.

As well as the nozzles described above, experiments were also performed with the "Bloom" nozzle furnished by the Selas Corporation. For spraying Nos. 4 and 6 oil, it is believed at the present time that the "tee" nozzle or an "annular-atomizing" nozzle will perform as well or better than the "Bloom" nozzle.

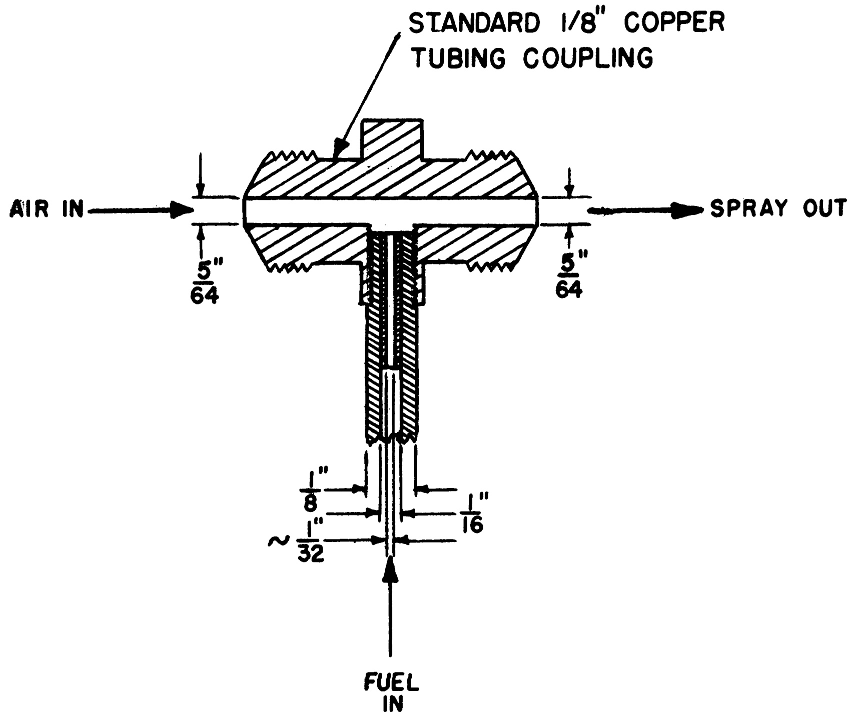


NO. 4 OIL, NOZZLE PRESSURE = 43.4 PSIA



SCHLIEREN PHOTOGRAPHS OF TWO-DIMENSIONAL BURNER

FIGURE 18



TEE NOZZLE USED FOR TWO-DIMENSIONAL BURNER

FIGURE 19

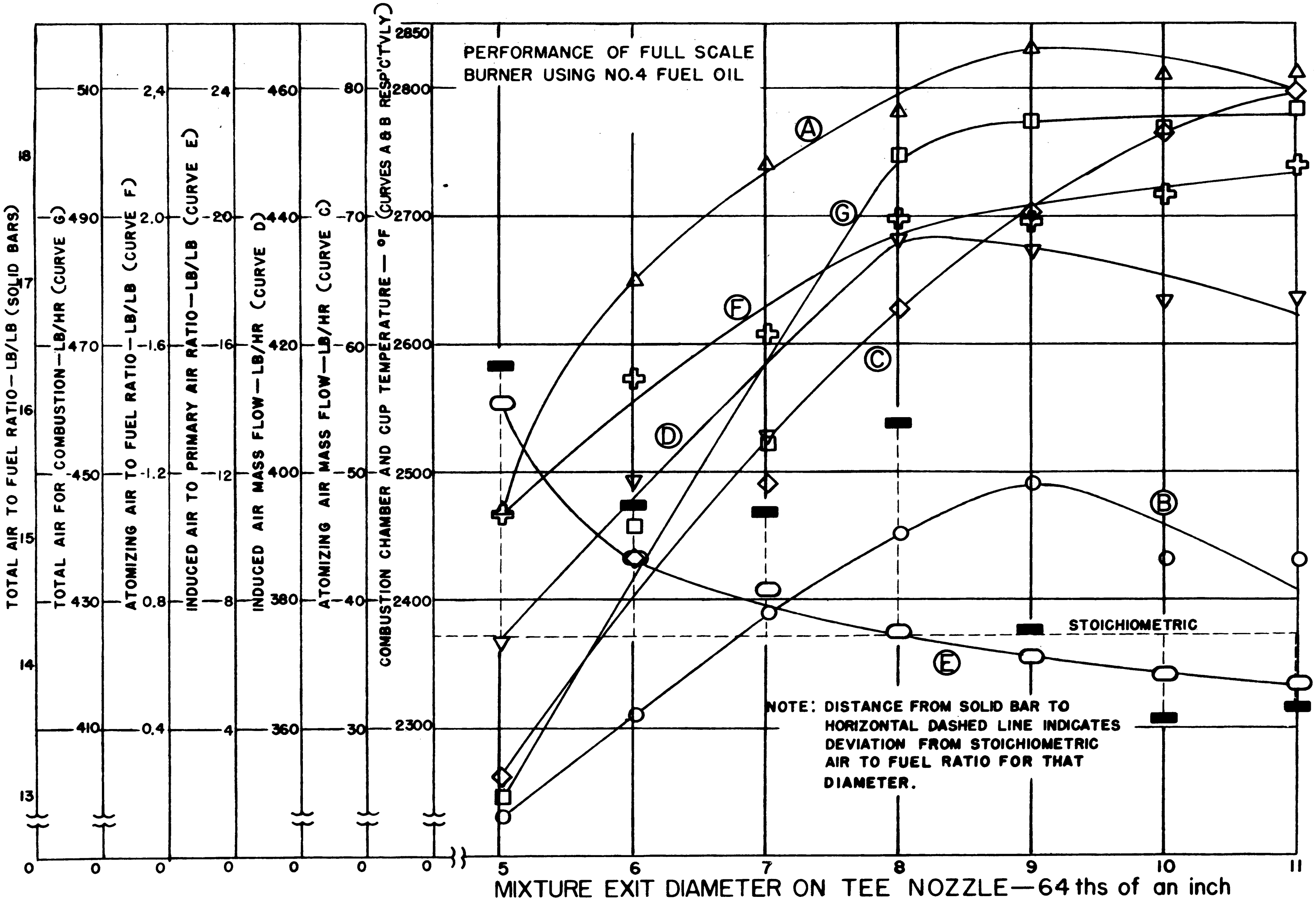

PERFORMANCE OF FULL SCALE
BURNER USING NO.4 FUEL OIL
TOTAL AIR TO FUEL RATIO—LB/LB (SOLID BARS)
TOTAL AIR FOR COMBUSTION—LB/HR (CURVE G)
ATOMIZING AIR TO FUEL RATIO—LB/LB (CURVE F)
INDUCED AIR TO PRIMARY AIR RATIO—LB/LB (CURVE E)
INDUCED AIR MASS FLOW—LB/HR (CURVE D)
ATOMIZING AIR MASS FLOW—LB/HR (CURVE C)
COMBUSTION CHAMBER AND CUP TEMPERATURE — °F (CURVES A & B RESP'C'T'VLY)
STOICHIOMETRIC
NOTE: DISTANCE FROM SOLID BAR TO HORIZONTAL DASHED LINE INDICATES DEVIATION FROM STOICHIOMETRIC AIR TO FUEL RATIO FOR THAT DIAMETER.
MIXTURE EXIT DIAMETER ON TEE NOZZLE—64 ths of an inch

FIGURE 20

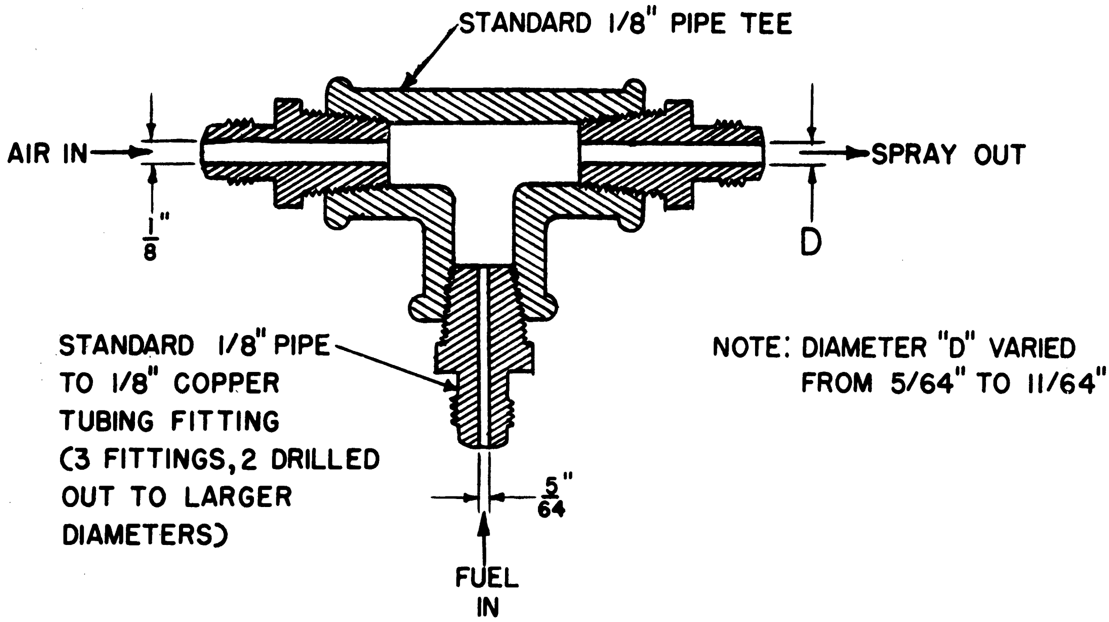


FIGURE 21 – DIMENSIONS OF TEE NOZZLES USED IN 12 INCH BURNER.

2. Fuel Properties. Experiments have been performed only with Nos. 2, 4, and 6 fuel oil. Since the properties are more or less established, changes in physical properties result only when the state of the fluid is changed. For example, preheating of No. 6 "Bunker C" oil results in lowering the viscosity and presumably improving the characteristics of the spray. Experiments are currently in progress on the effects of fuel preheating on burner performance. It was found easier to use a precombustion chamber with No. 4 fuel oil than to preheat the oil.

Experiments with No. 6 fuel oil, Bunker "C", have to date been performed only in small-scale radiant burners (up to 9-inch diameter cups). A photograph of No. 6 oil burning in one of the radiant burners may be seen in Fig. 22. Observations made during these experiments included the following:

1. Unless the No. 6 oil is preheated the spray is composed of large drops.
2. Yellow flames occur in the precombustion chamber.
3. Deposits build up in the precombustion chamber.
4. Considerable spattering occurs due to the high viscosity of the No. 6 fuel oil.
5. Burning of No. 6 fuel oil is accomplished most readily with a relatively long precombustion chamber and an incandescent baffle.

3. Air-Fuel Ratio and Absolute Quantities of Air and Fuel in Nozzle. Since in an induced air burner, the induced air to primary air ratio decreases as the absolute quantity of primary air and fuel is increased, the atomizing air to fuel ratio for a given nozzle must change as the fuel rate is increased. In a forced air burner this is not the case and a constant atomizing air to fuel ratio could be maintained. Specific data on the effect of atomizing air to fuel ratio on the droplet size and distribution have not been obtained on this project. However, considerable information exists on this subject which is applicable to the present problem. The data of York[3] indicate that good droplet distributions occur at all values of the air to fuel ratio grater than 1, hence, it appears that the air to fuel ratios required of the nozzle in an induction burner will be satisfactory.

For a given nozzle design, it would be desirable to know the mass flow of air through the nozzle at any given air pressure. Since at air pressures greater than 28.5 psia, sonic flow occurs in the nozzle, the mass flow and pressure are not independent of each other, there being only one

[3] York, J. L., "Photographic Analysis of Sprays", Ph. D. Thesis, University of Michigan, 1949.

In the pictures shown in Figs. 30 and 31, a flat plate was used, in contrast to the previous beveled shape, in order to make the exhaust port entirely convergent, thus giving the smallest area and hence the greatest velocities at the extreme outlet (Fig. 26). The original design had a diverging exhaust port and therefore the velocities were less at the end of the exhaust channel than at the front (Fig. 36).

Data taken with this design (Figs. 30 and 31) indicate that the flow pattern was much better along the ceramic and that the temperatures were higher as compared to the flow pattern and temperatures for the original design (Fig. 32). In each set of pictures the mass flow is given at two different values for each distance across the exhaust port. The temperature is seen to increase with this distance up to a certain point and then fall off. Also, the temperature increase with increasing mass flow as would be expected.

Experiments were then performed with 12-inch radiant burners. A schematic sketch of the original ceramic burner tip is shown in Fig. 33. The modifications made on this burner tip are illustrated by Fig. 34.

The burner head was modified as shown in Fig. 34 in order to cause more intimate mixing of air and propane, and because it was felt that if the burner head walls could be heated to incandescence by preburning, the fuel oil would, upon striking the hot walls, burn immediately. Thus, only the clean, hot exhaust gases would leave the burner tip. A flame holder was inserted at the entrance to the burner tip to help cause preburning to take place.

Both burner heads performed well when the fuel used was propane. Temperatures attained by the ceramic radiation reflector reached about 2400°F. With No. 2 fuel oil and the cermic burner tip shown in Fig. 33, this temperature could not be maintained. A yellow smoky flame resulted and the reflector cooled from incandescence.

Better results were obtained using the modified burner head shown in Fig. 34. The oil flame was still slightly yellow but the reflector temperature was maintained at about 2400°F (after it was heated to that temperature with propane).

The operating conditions used for both propane and No. 2 fuel oil with this modified burner head are shown in Table III

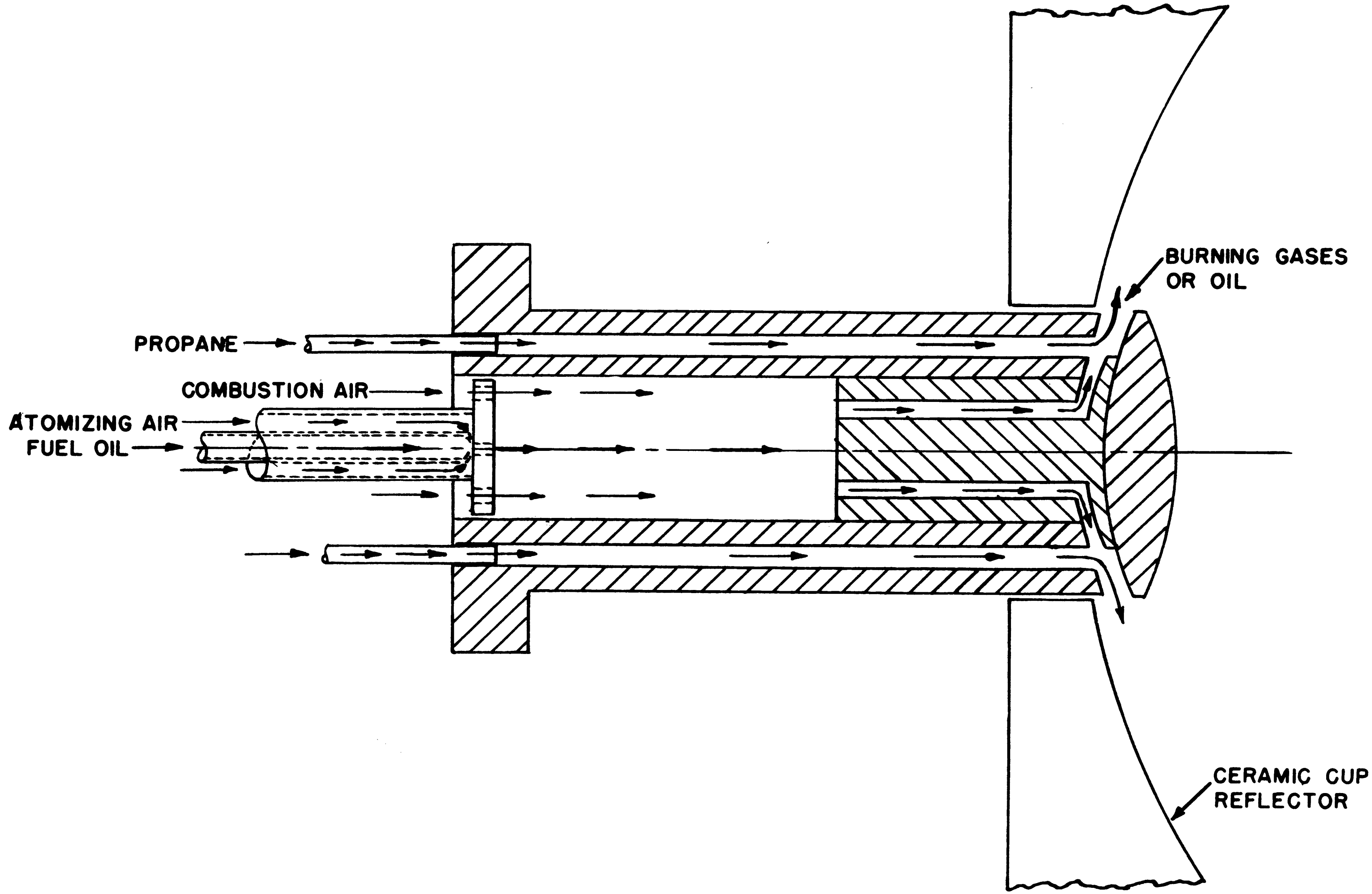


FIGURE 33- ORIGINAL CERAMIC BURNER TIP

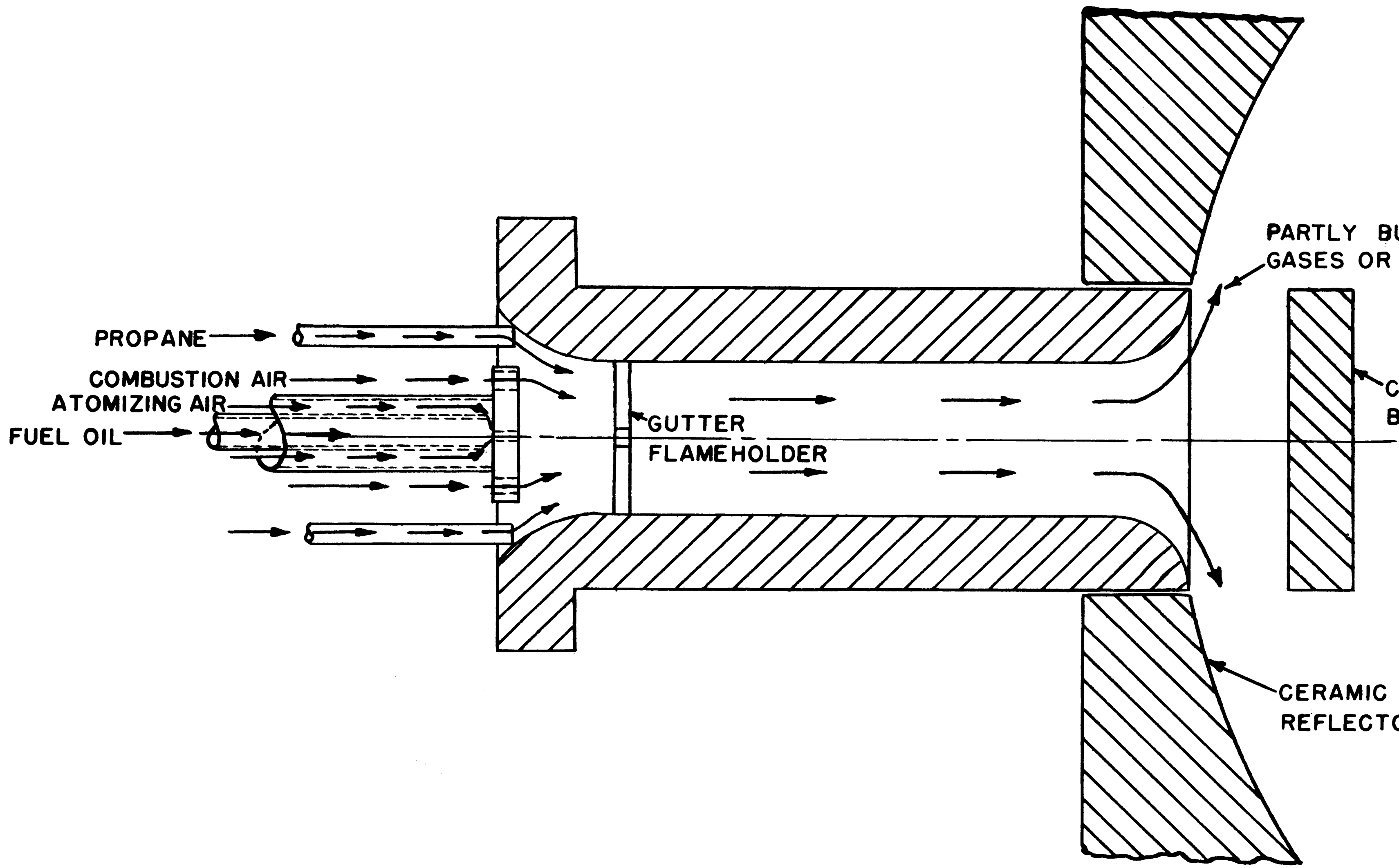


FIGURE 34 - MODIFIED CERAMIC BURNER TIP

TABLE III

OPERATION OF 12-INCH RADIANT BURNER WITH MODIFIED CERAMIC TIP

Material	Combustion Air	Atomizing Air	No. 2 Fuel Oil	Propane
Pressure	20 in. H_2O	100 psig	90 psig	30 psig
Temperature	92°F	101°F	85°F	95°F
Rotameter Reading	4700	74	20	150
Standard Volume Flow	4700 ft^3/hr	260 ft^3/hr	-	150 ft^3/hr
Density	0.075 lb/ft^3	0.556 lb/ft^3	-	0.326 lb/ft^3
Actual Volume Flow	4650 ft^3/hr	94.4 ft^3/hr	-	89 ft^3/hr
Mass Flow	349 lb/hr	52.5 lb/hr	28 lb/hr	29 lb/hr
Stoichiometric Fuel Rate	-	-	30.5 lb/hr	28.8 lb.
Theoretical Heat Output	-	-	580,000 Btu/hr	620,000Btu/hr

The temperature attained by the reflector is very dependent upon the distance of the baffle from the burner head. The distance was adjusted until optimum burning took place. The heat output and temperature attained were limited by the supply of combustion air available.

2. Shape and Location of Baffle

As previously indicated, a flat baffle was found to give the highest cup temperatures with the present parabolic cup shape. Experiments were performed with a full-scale induced air burner using No. 4 fuel oil to determine the optimum location of the baffle. The results are given in Table IV.

TABLE IV

EFFECT OF BAFFLE LOCATION IN 12-INCH RADIANT BURNER

Distance of Baffle From Tip, in.	Combustion Chamber Temperature, °F	Reflector Wall Temperature, °F
1/2	2740	2250
3/4	2660	2270
1	2630	2400
1-1/4	2670	2410
1-1/2	2700	2420
1-3/4	2700	2430
2	2710	2430
2-1/2	2680	2370

These results show that if the distance is greater than 3/4 inch, the distance has not much effect on reflector wall temperature. However, the distance has an effect on temperature uniformity acros the surface of the reflector. This is shown pictorially in Fig. 35 and described below with reference to Fig. 35 . Dark areas indicate regions of lower temperature.

A - Cold ring around tip to start, closed in after burning for short period, uniform heat,
B - Similar to A,
C - Slightly cooler toward rim,
D - Similar to C
E - Slightly cooler aroung tip, almost completely uniform after period' of operation
F - Cool ring has widened and begun to move outward, almost uniform after short period of operation,
G - Similar to F
H - Hot at rim and around tip, cold ring has widened to cover most of surface, hot at rim and around tip

For uniformity of temperature across the surface, the smaller distances are superior. However, higher temperatures are attained at greater distances. Further experiments will be run at about 1-1/2 inches to improve the temperature uniformity.

3. Shape and Surface Roughness of Cups. Experiments on this phase have not yet been performed.

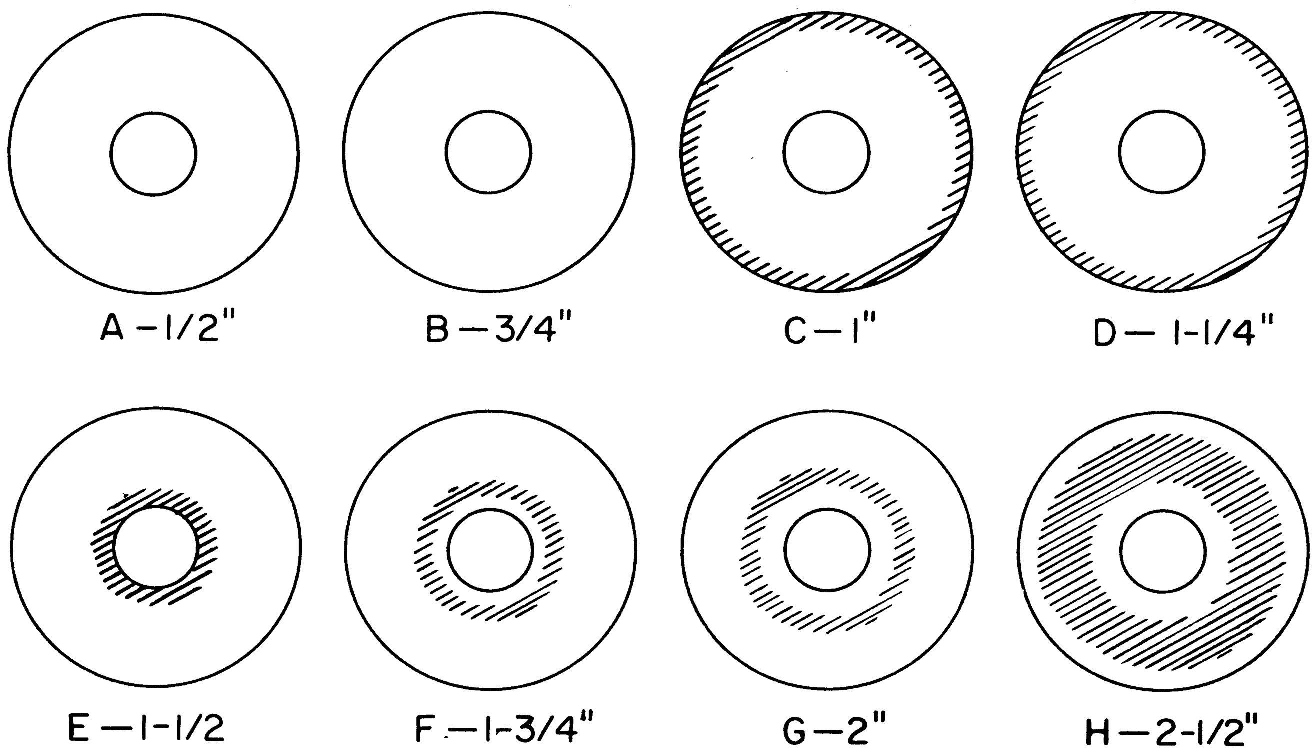


FIGURE 35- EFFECT OF BAFFLE LOCATION ON TEMPERATURE UNIFORMITY

4. Baffle and Cup Materials. To date, porous ceramic firebrick has performed satisfactorily in 12-inch open radiant burners. Material difficulties are anticipated in closed furnaces, however. With smaller (9-inch diameter cup) radiant burners constructed from K-30 firebrick, experiments were made using different baffle materials, the baffle being cemented to the cup. The results of these experiments are shown in Table V. As indicated, baffles constructed from K-30 saturated with zirconia and from Vycor glass offer the most promise. A photograph of a 9-inch diameter cup radiant burner with a Vycor baffle using No. 4 fuel oil may be seen in Fig. 36. As this photograph indicates, the use of a transparent Vycor baffle eliminates the dark area in the center of the cup normally caused by a solid baffle on burner tip; it also allows an approach to a uniform cup temperature.

TABLE V

RESULTS OF BAFFLE MATERIAL TESTS

Baffle Materials	Quality of Performance
1. H-W-26 Insulating firebrick	Melted
2. K-30 Insulating firebrick	Softened
3. K-30 Painted with "Selas" 206 ceramic cement	Softened
4. K-30 Painted with "Sairset" bonding mortar	Softened
5. Cast ["Selas" 206 + ground K-30]	Eroded
6. Cast ["Sairset" + ground k-30]	Cracked and was hard shape.
7. K-30 saturated with "Sairset"	Better than painted but still softened at highest temperatures.
8. K-30 saturated with "zirconia"	Good except it cracks on cooling.
9. Vycor (siO_2) glass	Softens slightly at highest temperatures and cracks easily from small tensile forces.

II. INVESTIGATION OF OIL-FIRED RADIANT BURNER, TIP ATOMIZER TYPE

A type of oil burner was built which would atomize the fuel in the burner cup instead of in a chamber behind the burner. This was intended to prevent the deposition of fuel on the baffle or burner tip which results in cracking of the fuel oil, blocking the exit ports. The two types of burner-head atomizerstested are shown in Figs. 37 and 38.

The air atomizer was designed to distribute fuel uniformly around the axis of the nozzle and in the plane perpendicular to the axis. This nozzle operated moderately well with 10 to 20 psig combustion air but would not operate on an air pressure of 40 tO 50 inches of water as desired. It was difficult to obtain a uniform distribution of the spray because of inability to maintain accurate tolerances at the fuel and atomizing air exits.

The pressure atomizer operate more satisfactorily at 10 to 20 psig combustion-air pressure than the air atomizer because of its even distribution of fuel aroundthe burner cup.

An interesting feature of the pressure-atomizer burner head is that the circulation forces the spray back against the burner wall. That feature alone makes the burner become radiant, but it has the disadvantage of periodically flattening the cone against the burner head, preventing atomization.

Figure 39 shows the pressure atomizer operated in three different diameter burner cups, 3, 5, and 10 inches, and a description of its characteristics in each case.

III. INVESTIGATION OF RADIANT BURNERS, BURNER TYPES WHICH DO NOT USE BAFFLES OR BURNER TIPS.

It would be desirable to design a radiant burner which would function without the present type of baffle. The purpose of the baffle is two-fold. First, it serves as a type of flameholder, and second, it diverts the hot gases along the radiant walls. Without this baffle, blowout might occur.

The first experimental model is shown in Fig. 40. The corner served as a flameholder as well as channeling the gases along the radiating surface.

Results with this type of burner were encouraging, but the gases tended to separate from the surface, and the flame was not too stable when starting. Also, the lower end of the radiating surface did not get very hot.

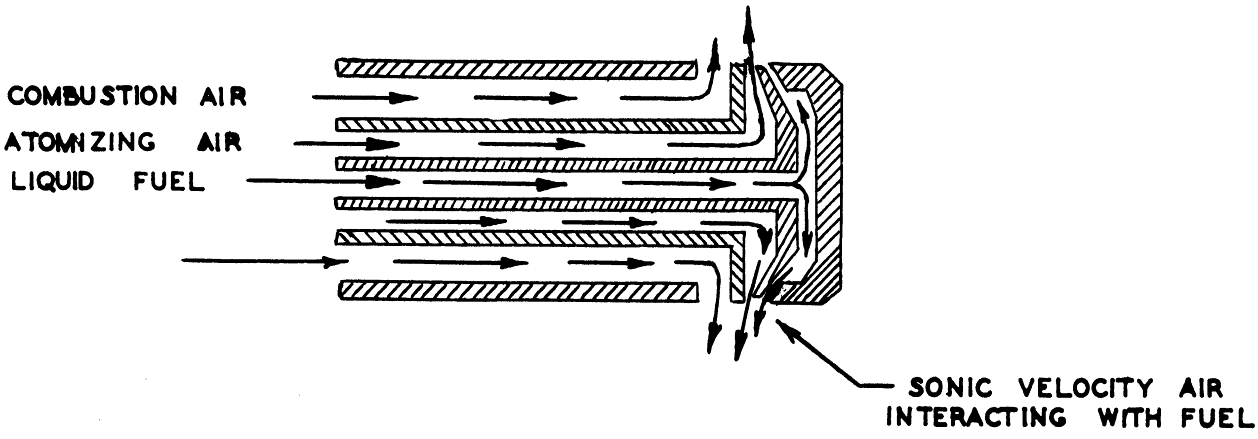


FIGURE 37 - AIR ACTIVATED BURNER-HEAD ATOMIZER

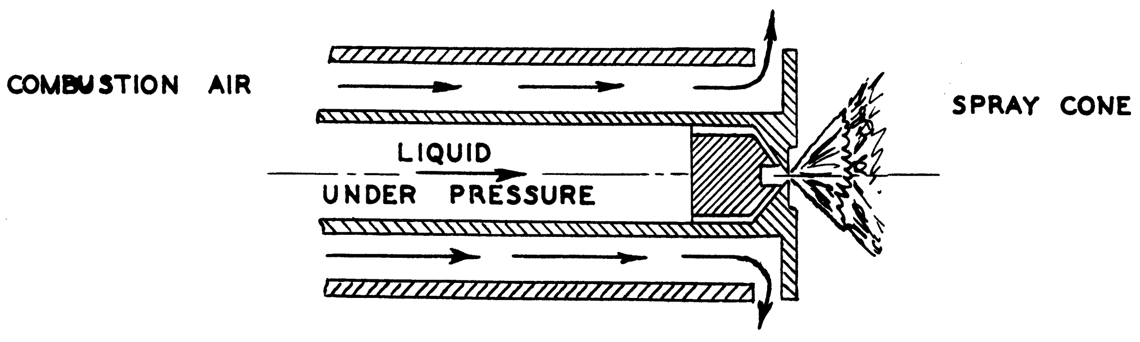
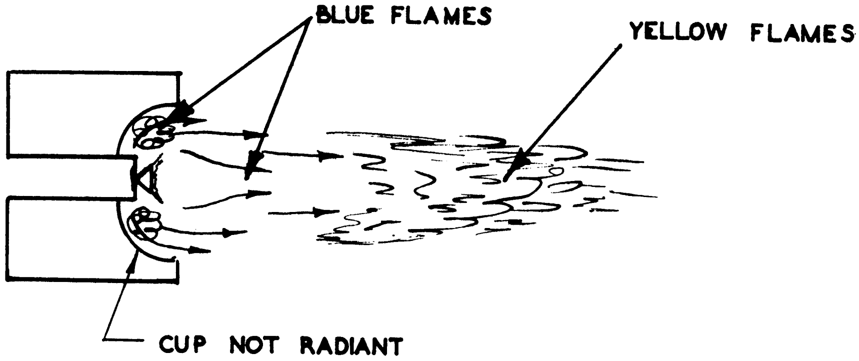
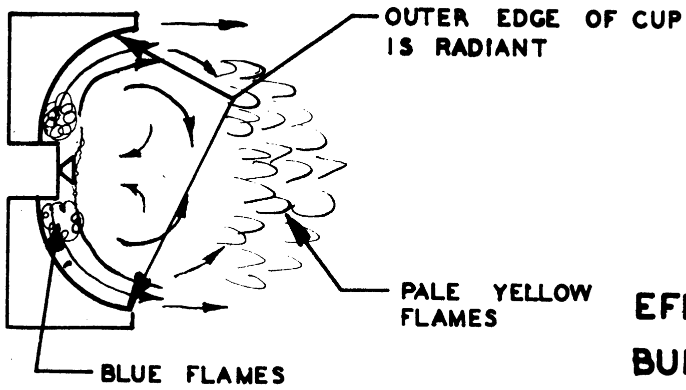


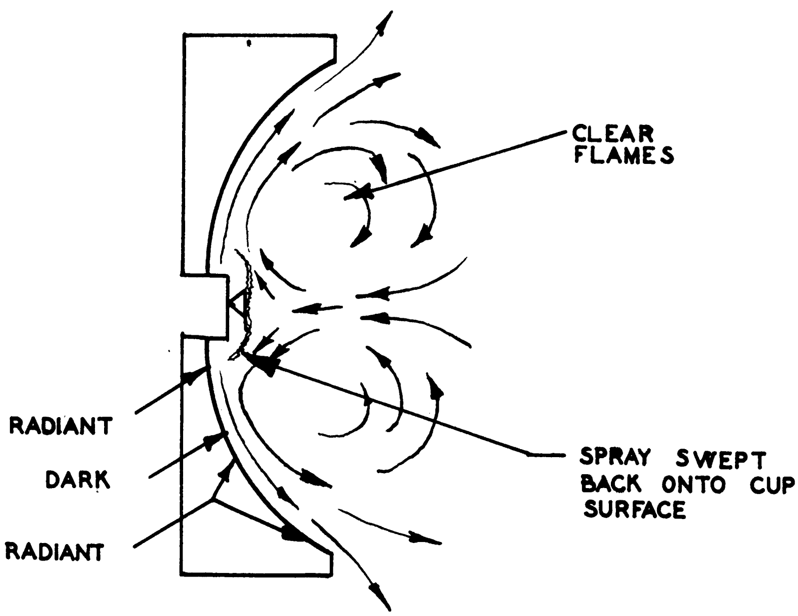
FIGURE 38 - PRESSURE ACTIVATED BURNER-HEAD ATOMIZER



A. 3 INCH BURNER



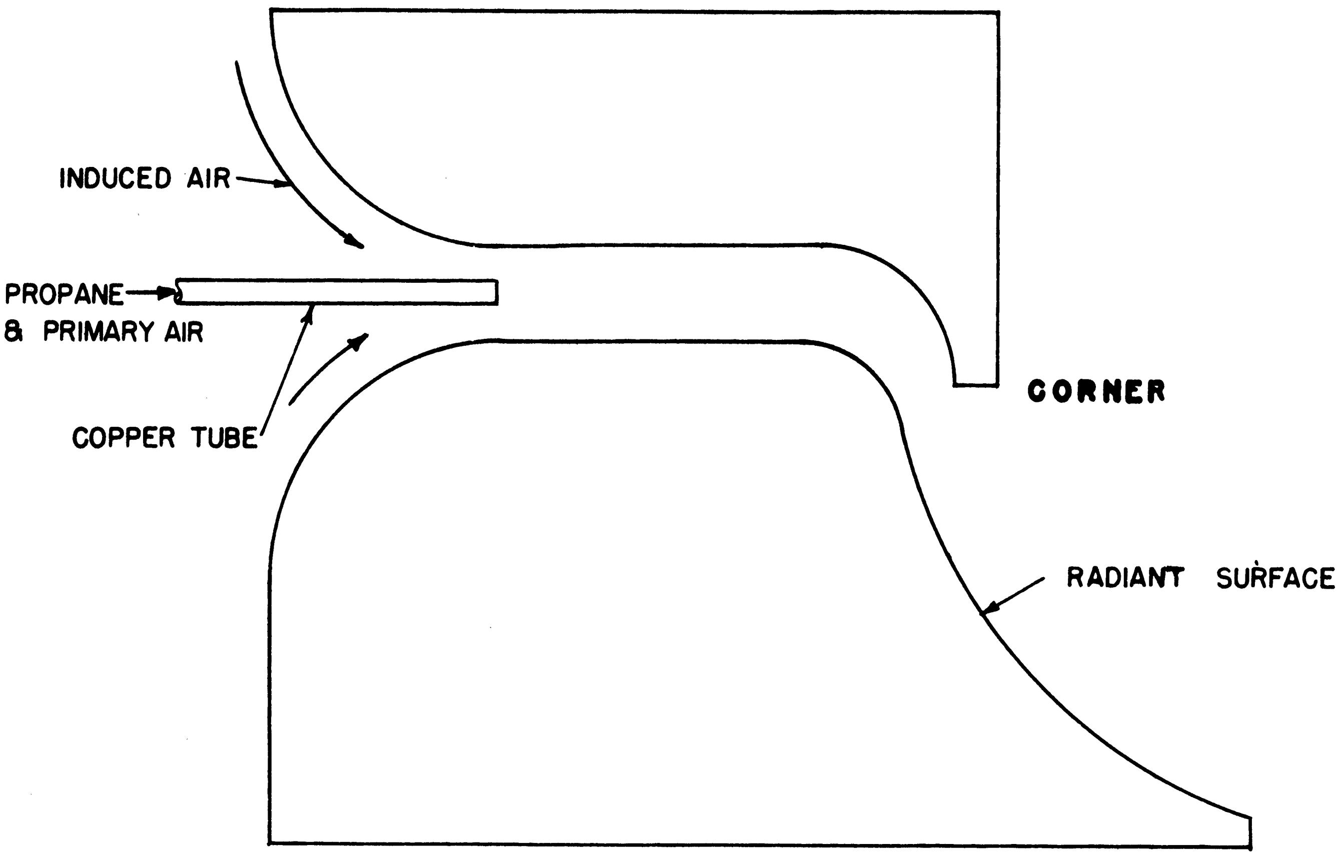
B. 5 INCH BURNER



C. 10 INCH BURNER

FIGURE 39

EFFECT OF CHANGING BURNER CUP SIZE ON RADIANCY OF CUP AND FLAME CIRCULATION, (FOR PRESSURE ACTIVATED BURNER-HEAD ATOMIZER.)

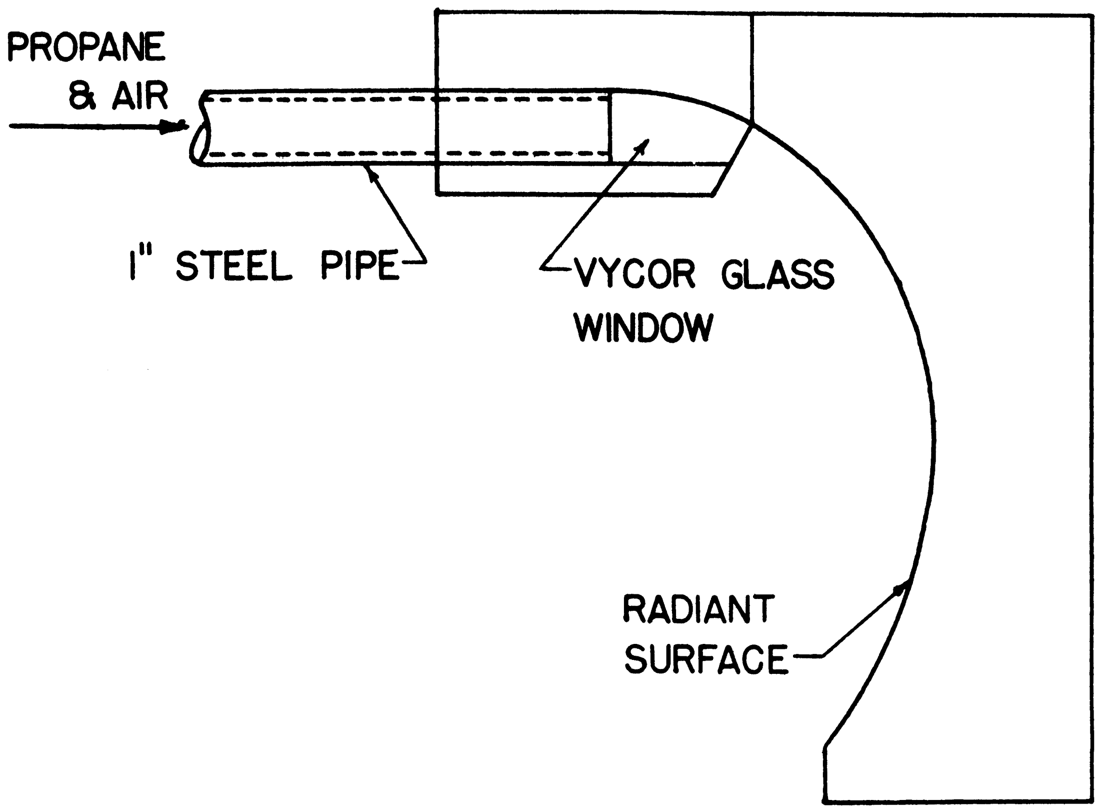


TWO DIMENSIONAL BAFFLE-LESS BURNER
NEGATIVE CUP CURVATURE

FIGURE 40

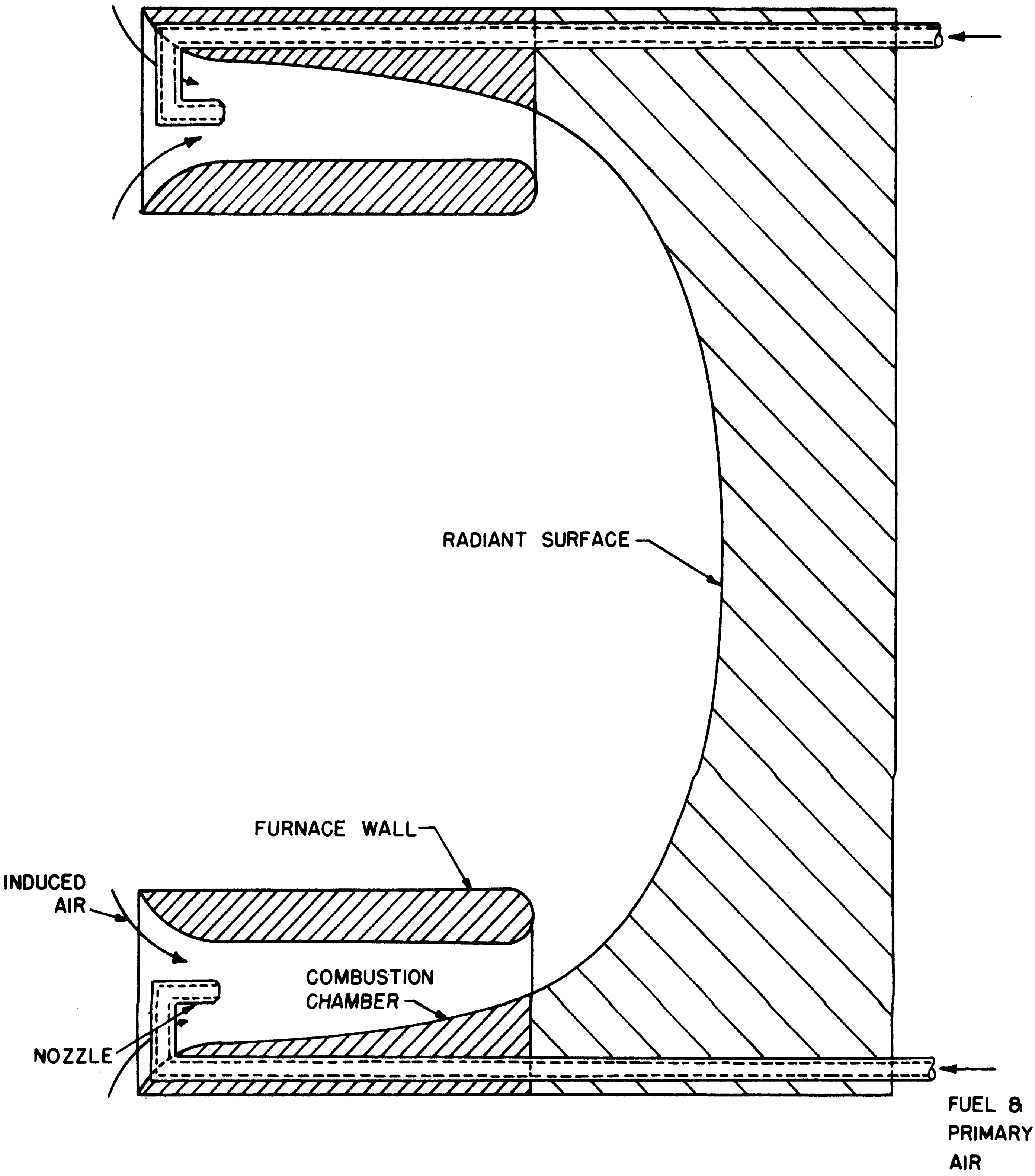
A second burner was designed to force the gases along the radiating wall (Fig. 41). At low flow rates the radiating surface would get hot only about halfway down.

A third type of burner could be made two dimensionally entirely out of ceramic as shown in Fig. 42. Experiments are to be performed with this design in the immediate future.



TWO DIMENSIONAL BAFFLE-LESS BURNER
POSITIVE CUP CURVATURE

FIGURE 41



TWO DIMENSIONAL BAFFLE-LESS BURNER
POSITIVE CURVATURE—DOUBLE INLET

FIGURE 42