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FUEL-NOZZLE—AIR-INDUCTION SYSTEMS

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OBJECT

The long range objective of the combustion research program sponsored by the Selas Corporation of America has been the development of an effective oil-fired radiant burner which can also be operated with gaseous fuels. The immediate objective stated in Progress Report No. 1, November, 1954, was to obtain an understanding of those factors which contribute to the effectiveness of an oil-fired radiant burner. That a partial understanding of these factors was obtained is evidenced by the results summarized in Progress Report No. 1, namely,

1. A radiant burner can be operated with "Bunker C" fuel oil to obtain high cup temperatures.

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 Achimedes spiral.

3. Atomization of fuel oils can be simplified 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.

4. The "petal" effect in present Selas burners, due to a cold region of no gas flow next to the cup wall, can be eliminated by aerodynamic streamlining of the cup inlet.

These results and the attainment of the immediate objective were discussed on November 30, 1954, in Philadelphia, Pennsylvania, at a meeting between Mr. A. A. Furczyk, Mr. James Henwood, and Mr. Charles Morck of the Selas Corporation and Dr. A. Weir of the University of Michigan. At this time, Mr. Furczyk stated that, considering item 2 above, only 5 percent of the total air for combustion could be used for the purpose of fuel atomization and induction of the remaining 95 percent of the air, and that this air was required to be at a pressure of less than 30 psig. In other words, it was required to design a fuel-nozzle-ejector system which would atomize the fuel using an atomizing fluid-fuel ratio of 0.7 and induce 19 pounds of air per pound of atomizing air, the downstream pressures to be those resulting from efficient stoichiometric combustion, the upstream atomizing air pressure to be less than 30 psig. The experimental work performed during the months of December, 1954, and January, 1955, in order to achieve this objective (i.e., design of a fuel-nozzle-ejector system to meet the above requirements) is presented in this report.

SUMMARY

This report covers experimental and analytical work performed during the months of December, 1954, and January, 1955. Previous work is reported in Progress Report No. 1, November, 1954. Experiments performed on a water table show the recirculation occurring within the cup with present Selas Burner cup shapes. An analysis of the requirements for the fuel-nozzle—air-induction system indicated that the momentum of the primary atomizing stream would have to be increased if the amount of air induced was to be increased to 95 percent of the total air requirements. Since increase in pressure or mass flow of the primary fluid was undesirable, the momentum could be incresed only by increasing the velocity of the primary fluid. Sonic velocity already occurred in the fuel nozzle; therefore, the velocity could be increased only by raising the temperature of the primary stream (if the geometry of the nozzle remained constant), thus increasing the speed of sound in the primary fluid. Experiments were performed using a preheated air cycle and an exhaust product cycle. With the preheated air cycle, it was possible to induce 93 percent of the air required to achieve radiant cup temperatures with No. 4 fuel oil. Experiments with the exhaust product cycle indicated that it was feasible to obtain a high-velocity high-temperature gas stream to be fed to the fuel nozzle as an atomizing fluid at a pressure of 30 psig. Calculations also indicated the feasibility of a fuel-vaporizing nozzle.

EXPERIMENTAL PROCEDURE AND RESULTS

A. RECIRCULATION PATTERN IN BURNER CUP

These experiments were performed as part of the research objective of gaining an understanding of the factors affecting radiant burner performance.

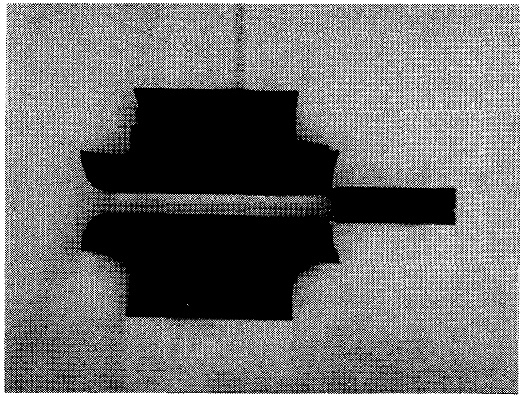
A small two-dimensional model of the precombustion chamber section and cup inlet section of a radiant burner was placed on a water table, permitting direct observation of the flow streamlines throughout the burner model. Streamlines were marked by lycopodium powder dropped in the water upstream of the model section. The resulting flow pattern was projected on a screen and photographed. Figure 1 is a photograph of the test section under no flow conditions.

When the flow rate through the nozzle was increased beyond a certain value, the flow backed up, changing upstream conditions as shown in Fig. 2. The upstream water level was increased at this point. This is analagous to sonic choking of a nozzle with a gaseous fluid. Thermal choking (Mach = 1) at the precombustion chamber exit or at the cup inlet could also be obtained if sufficient heat is released in the precombustion chamber section.

Other photographs at a lower flow rate are shown in Figs. 3, 4 and 5. These flow patterns show that the burner acts as an ejector near the cup as well as in the converging-inlet section. A sketch of the observed flow pattern is presented in Fig. 6. Water is induced toward the cup surface by the exhausting fluid. This is one reason why radiant burner surfaces are cooler when not inclosed, a continuous cooling effect being produced by the induced air from the atmosphere. Although the ejector action was not readily photographed, it was observed easily on the water table. This recirculation pattern with present cup shapes would be disadvantageous in dusty atmospheres (as in cement kilns).

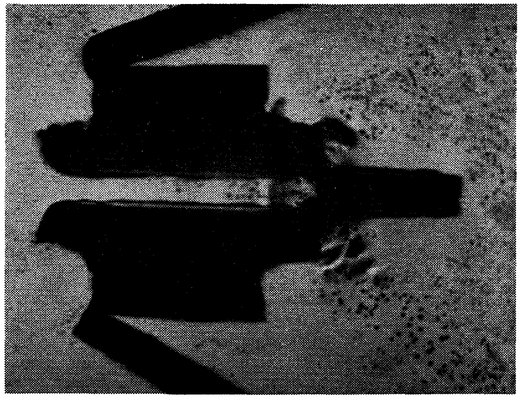
B. EFFECT OF DOWNSTREAM PRESSURE ON INDUCED AIR RATE

Progress Report No. 1 reported that when burning occurs in the main combustion chamber, the quantity of air induced is approximately one



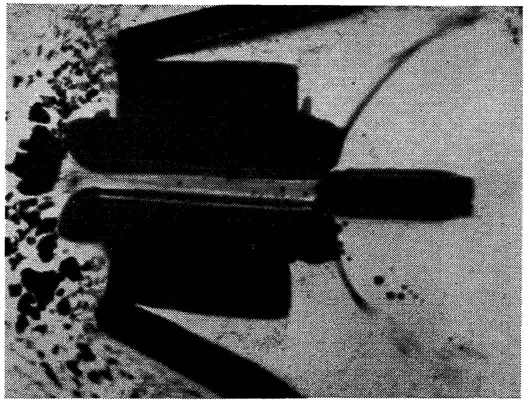
1

TEST SECTION



2

FLOW BACKED UP



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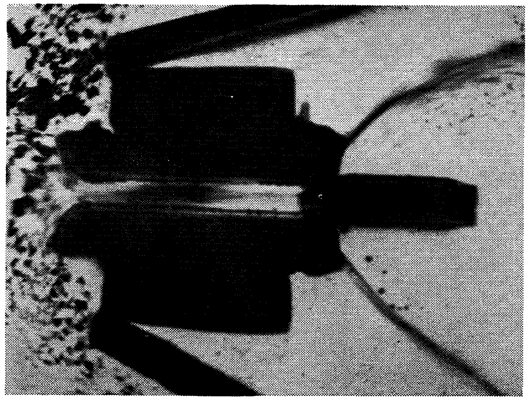

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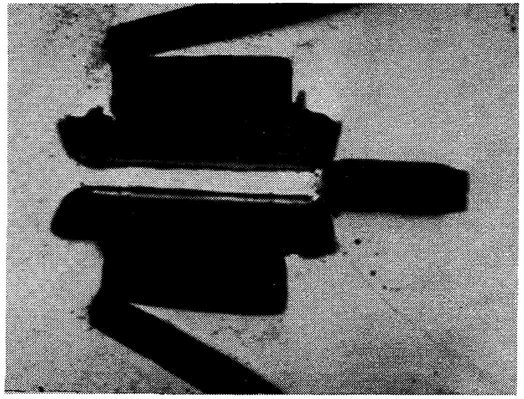

5

FLOW PATTERNS

WATER-TABLE PHOTOGRAPHS

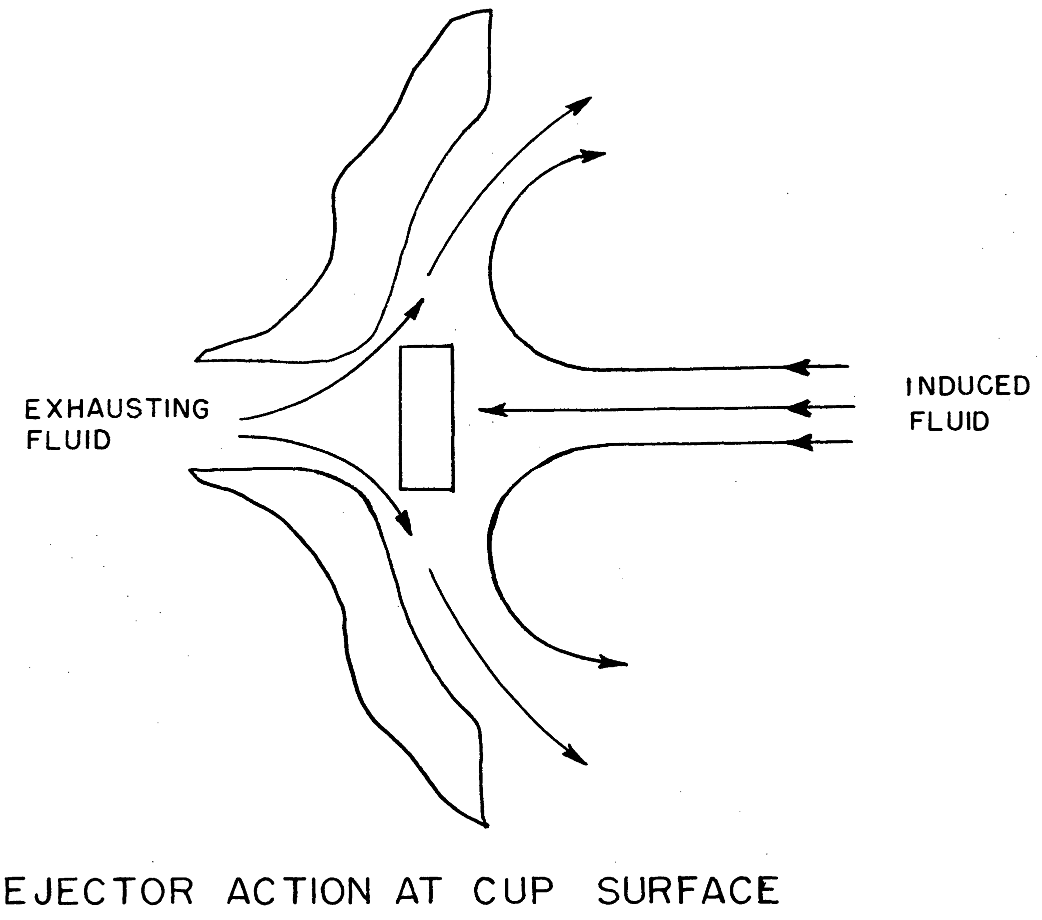

Fig. 6. Ejector action at cup surface.

third as much as that induced when no burning takes place, other conditions being the same. This reduction is caused by the pressure drop of burning, requiring higher burner-inlet pressures and hence higher converging-inlet downstream pressures. The pressures which can be anticipated due to burning were discussed in the previous report, and the results of experiments performed with a combustion chamber similar to the precombustion chamber have been published.*

Although the pressure drop of burning is known,* the effect of downstream pressure on the amount of air induced has not previously been determined for the present inlet shape and fuel-nozzle system. Therefore, an experiment was designed to determine it quantitatively.

The apparatus is shown in Fig. 7. A 36-inch length of 3-inch pipe was fitted with a converging inlet at one end (Archimedes spiral) and with a movable plate at the other end. Primary air was metered into the converging-inlet section, and the amount of air induced was measured by a static-pressure tap at the entrance to the pipe. Bernoulli's equation gave the velocity and thus the mass rate of flow, assuming a uniform velocity and density distribution. Moving the plate at the end of the tube changed the downstream pressure and the amount of air induced.

The downstream static pressure was measured with an inclined water manometer connected at a point two thirds of the length from the inlet. The primary air nozzle was a 9/64-inch-diameter orifice. The effective induced-air area at the flow-measuring pressure tap was 0.044 square foot. The air temperature was about 60°F.

The ratio of induced air to primary air versus downstream pressure for different primary-air mass flow rates is shown in Fig. 8. A cross plot of this curve yields the ratio of induced air to primary air versus primary-air flow rate with downstream pressure as a parameter, as shown in Fig. 9. As the downstream pressure is increased by a few inches of water, the ratio of induced air to primary air is materially decreased, this effect being most pronounced at lower primary-air flow rates. For downstream pressures less than 1 inch of water, there is a maximum ratio of induced to primary air at some primary-air mass flow rate. For a downstream pressure greater than 1 inch of water, there is no maximum and the ratio continuously increases with increasing primary-air mass flow rates.

Progress Report No. 1 showed that burning produces a decrease in the air ratio to one third of its value when no burning occurs. Figures 8 and 9, at a primary-air mass flow rate of about 70 pounds per hour, show that

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*Alexander Weir, Jr., Ind. Eng. Chem. 45, 1637, (1953).

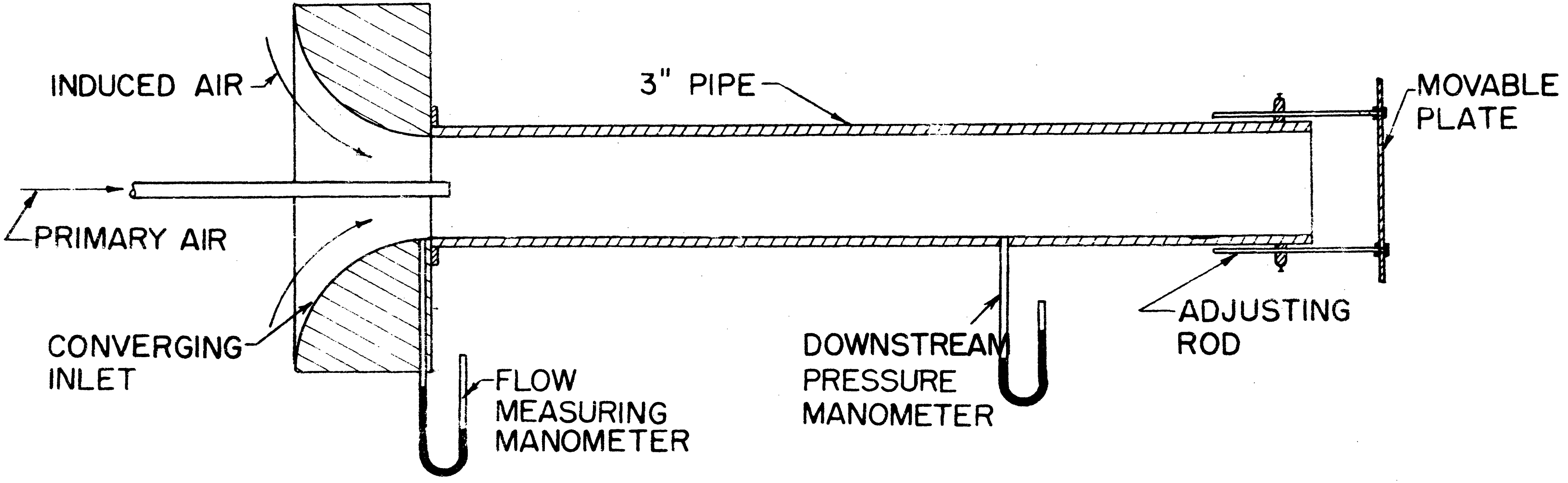
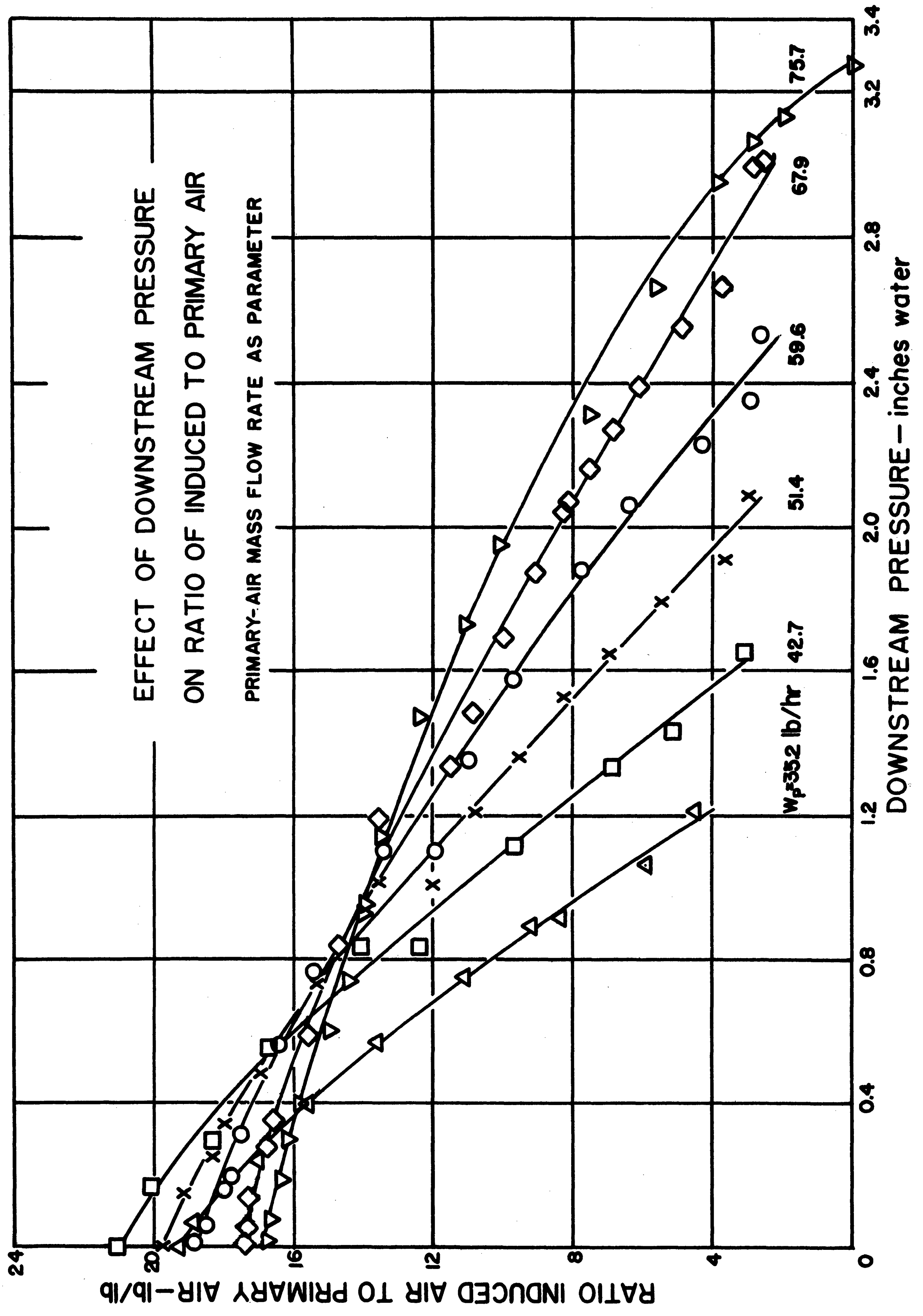


Fig. 7. Apparatus for study of effect of downstream pressure on induced air rate.



EFFECT OF DOWNSTREAM PRESSURE ON RATIO OF INDUCED TO PRIMARY AIR

PRIMARY-AIR MASS FLOW RATE AS PARAMETER

Fig. 8.

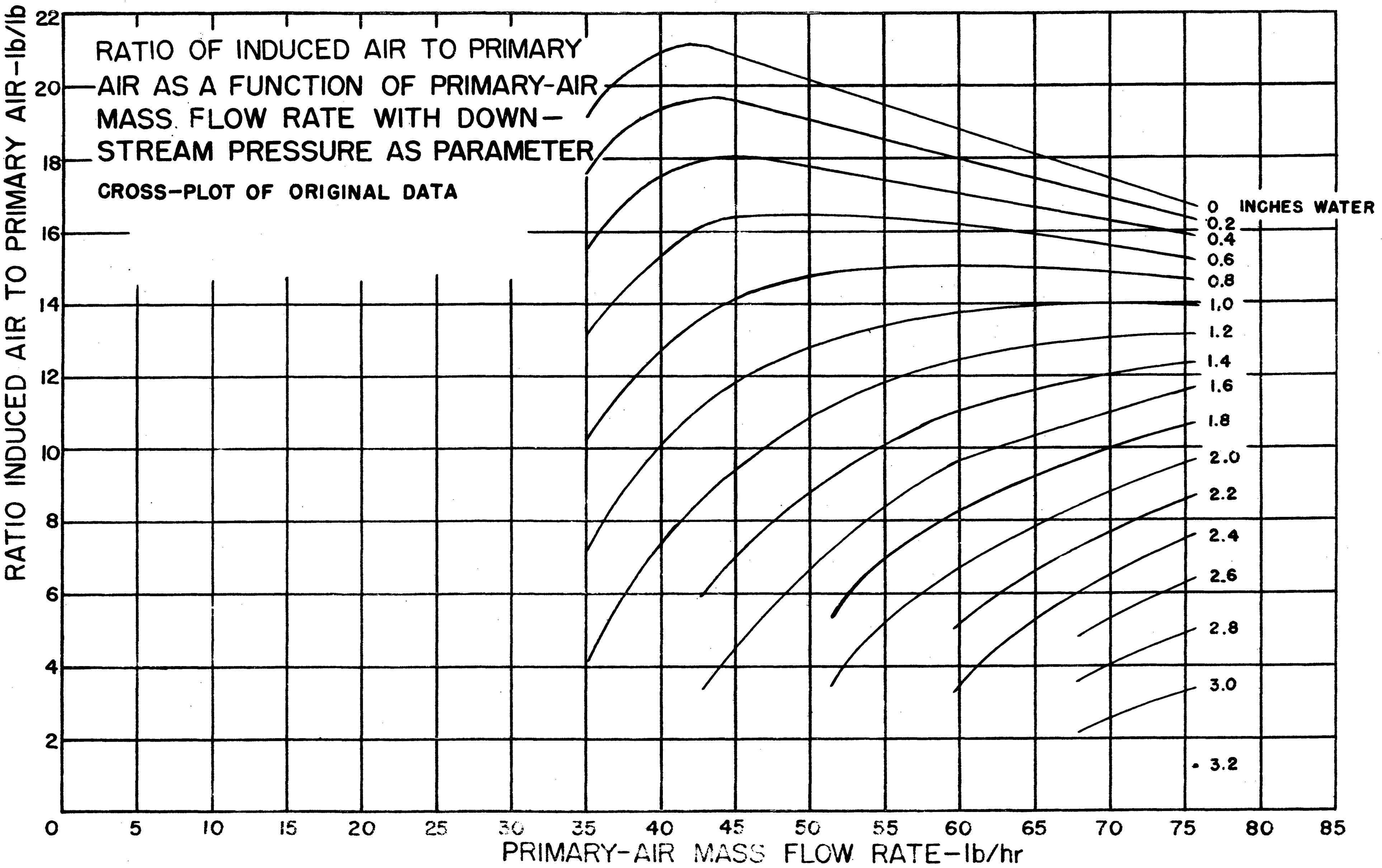

RATIO OF INDUCED AIR TO PRIMARY AIR AS A FUNCTION OF PRIMARY-AIR MASS FLOW RATE WITH DOWN-STREAM PRESSURE AS PARAMETER
CROSS-PLOT OF ORIGINAL DATA
RATIO INDUCED AIR TO PRIMARY AIR-lb/lb
22
20
18
16
14
12
10
8
6
4
2
0
5
10
15
20
25
30
35
40
45
50
55
60
65
70
75
80
85
PRIMARY-AIR MASS FLOW RATE-lb/hr
0 INCHES WATER
0.2
0.4
0.6
0.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
2.8
3.0
3.2

Fig. 9.

the burning is equivalent in blocking effect to a downstream pressure of roughly 2.5 inches of water. This small pressure must be overcome in order to triple the ratio of induced air to primary air now obtained with burning.

C. EXPERIMENTS WITH HIGH-TEMPERATURE PRIMARY STREAMS

Figure 9 shows for cold (60°F) primary air streams that in order to induce 19 pounds of air per pound of primary air (the 5-percent limitation given as an objective) the downstream pressure must be less than 0.2 inch of water for the converging-inlet sections used. Even at this low pressure value, only 37 to 52 pounds per hour of primary air may be used. (By adjusting the size of the orifice in the nozzle, any desired flow rate can be achieved with an upstream pressure less than 30 psig.) However, the preceding section showed that about 2.5 inches of water, downstream pressure, results from the downstream combustion. Hence, it is not possible to induce the required 19 pounds per pound of air using the present system.

It has been experimentally demonstrated, however, that a high-temperature primary stream will induce the required 19 pounds per pound. The quantity of air induced depends upon the momentum of the primary stream; thus, for a given mass flow, the quantity of air induced is solely dependent on the velocity of the primary stream. The primary-stream velocity is limited to the speed of sound in the minimum area, which is located in the primary-stream orifice. The velocity of the primary stream and its momentum may be increased by raising its temperature, because the speed of sound is directly proportional to the square root of the absolute temperature. This increase in momentum should result in a large increase in the air induced for a given amount of primary air at a pressure less than 30 psig.

An increase in temperature would also aid in vaporizing the main fuel stream, and the hot oil vapor might be treated as other gases are in present Selas burners. The limitations of 30-psig upstream pressures can thus be met by using higher temperature streams and increasing the momentum by increasing the velocity. The additional energy required to heat the primary stream can be obtained in several ways. Experiments were performed on two different techniques. First, high-temperature exhaust products, resulting from the combustion of 5 percent of the total air required, were used as the primary atomizing fluid. Second, a regenerative cycle heated 5 percent of the total air for combustion in a heat exchanger located in the precombustion chamber, and this heated air served as the atomizing fluid in the fuel nozzle. The experimental procedure and results of these techniques are described below.

1. Exhaust-Product Cycle.—The equipment used for the study of the exhaust-product cycle is shown in Fig. 10. Primary air at 30 psig (5 percent of the air eventually needed for combustion) is mixed stoichiometrically with propane gas and the mixture enters the primary combustion chamber through the primary-stream entrance nozzle where it is initially ignited by a spark plug. (Burning will sustain itself once the mixture is ignited.) The gases burn in the primary combustion chamber and leave through the primary-stream exit nozzle at high temperature and velocity. The hot stream strikes the main fuel oil stream, which vaporizes and ignites upon being mixed with the induced air. The high-velocity stream should induce enough air to burn the fuel oil completely, that amount being twenty times the amount of air provided to burn with propane. The fuel oil vapor then burns in the main combustion chamber, and the hot products of combustion are deflected by the baffle against the cup, making it radiant.

The primary combustion chamber described above was constructed from 5-inch steel pipe, 12 inches long. The pipe was lined with ceramic rings having an inside diameter of 3 inches. In the preliminary trial, the entrance and the exit to the combustion chamber were reduced to 1/8-inch diameter. Burning was accomplished and a very-high-temperature high-velocity stream left the 1/8-inch exit. Unfortunately, the research program was cancelled before the amount of air induced at different operating conditions could be measured.

2. Preheated Air Cycle.—The apparatus used for these experiments is shown in Fig. 11. A 1/2-inch-diameter stainless-steel tube, formed into a U shape, was inserted in the precombustion chamber. The primary air was fed through this tube and thus was preheated before being used to atomize the fuel oil. A thermocouple and pressure tap were installed just upstream of the nozzle (at point 2 in Fig. 11) to measure the temperature and pressure of the heated air stream. The data obtained in the two runs made are presented in Table I.

TABLE I

EXPERIMENTS WITH PREHEATED-AIR CYCLE

Run No.	Air Data		T_2, °F	Oil Rate, lb/hr	Primary Air Rate, lb/hr	Ratio of Induced Air to Total Air (Stoichiometric)
	P_1, psig	P_2, psig				
1	94.3	95.0	1090	15.0	14.7	0.93
2	92.7	92.0	1115	15.0	14.5	0.93

P_1 = static pressure before heat addition
P_2 = static pressure after heat addition and before fuel nozzle
T_2 = stagnation temperature after heat addition and before fuel nozzle

Table I shows that 93 percent of the air required for stoichiometric combustion was induced. This is a considerable increase in the ratio of induced to primary air over that reported in Progress Report No. 1. In that report, the ratio of induced to primary air at stoichiometric conditions was 6 to 1, and Table I indicates that this ratio was increased to 13.3 to 1 under burning conditions.

The fuel nozzle used in these experiments is shown in Fig. 12. The oil rate through this nozzle was about one third the rate required for a burner output of 10^6 Btu per hour. A larger nozzle would be required for higher flow rates, since sonic velocity is established in the 1/8-inch air inlet. Although this nozzle was satisfactory for an output of 10^6 Btu per hour when cold (60°F) air was used, the use of 1100°F air requires a much larger area for sonic flow if upstream pressures are the same. (The mass flow rate through a sonic orifice is inversely proportional to the square root of the upstream stagnation temperature.*) The velocity of the hot air stream just upstream of the nozzle was calculated to be 36 feet per second. At a temperature of 1115°F, this corresponds to a Mach Number of 0.0186.

For an air temperature of 1100°F and an upstream air pressure of 30 psig, the mass velocity of the air through the nozzle inlet would be* approximately 83 pounds per second per square foot of orifice area. For a 10^6 Btu per hour burner in which 95 percent of the total air is induced, using No. 4 fuel oil, the nozzle would be required to handle 37.5 pounds of air per hour. The diameter of the air inlet on the fuel nozzle should then be, if an effective void space radius* of 0.004 inch is used, 0.159 inch rather than the 0.125-inch-diameter inlet presently used.

3. Oil-Vaporization-Nozzle Calculations.— With the nozzle shown in Fig. 12, insuffcient time is available for heat transfer from the air to the fuel oil. If the oil stream was introduced further upstream of the nozzle, however, sufficient mixing length would be available to heat the fuel oil. Consider the nozzle shown in Fig. 13. If the mass flow rate of oil was equal to the mass flow rate of air, and constant-pressure mixing occurred, then for 60°F (No. 4) fuel oil and 1200°F air, the exit mixture temperature would be

$$0.55\,(T_f - 60) = 0.25\,(1200 - T_f)$$
$$T_f = 415\,°F$$

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* A. Weir, J. L. York, and R. B. Morrison, "Two and Three-Dimensional Flow of Air Through Square-Edged Sonic Orifices", ASME Paper No. 54-A-112 (Preprint).

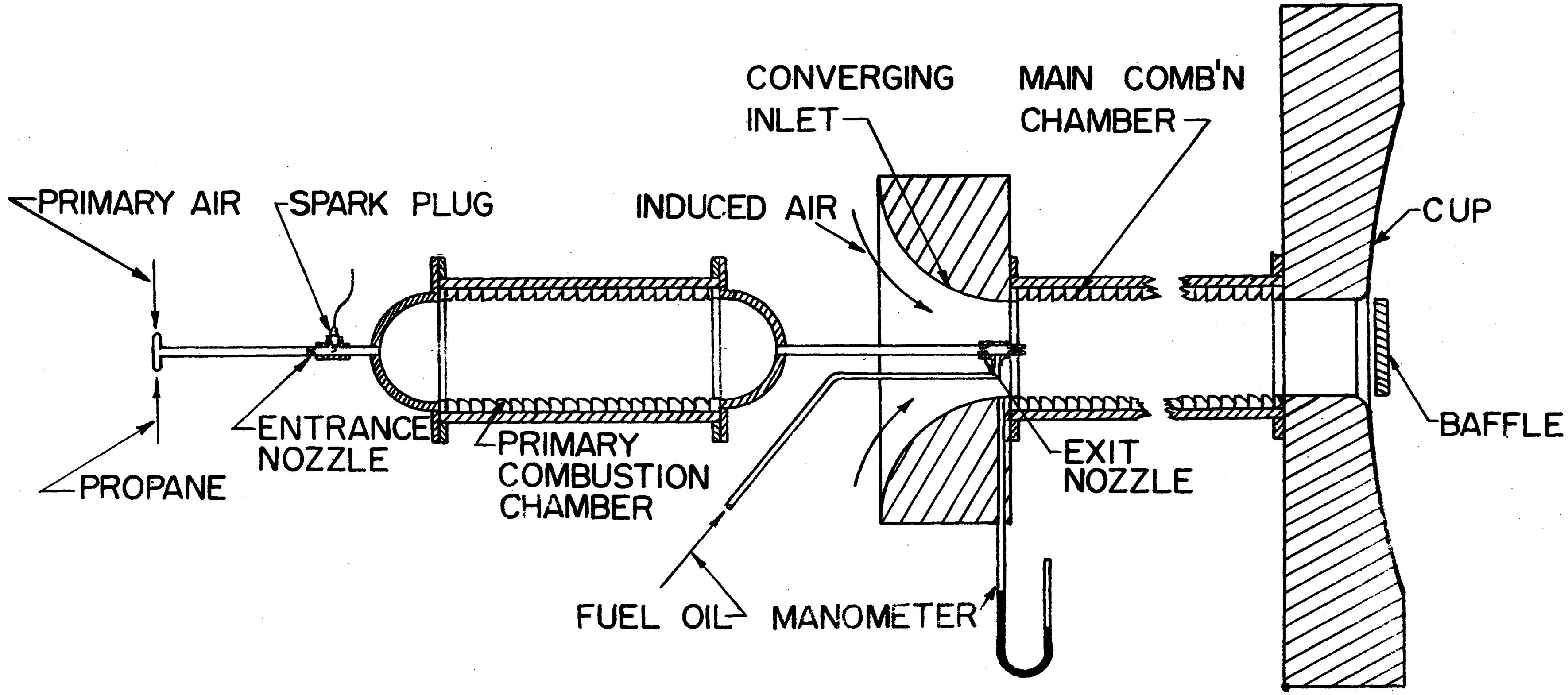

Fig. 10. Exhaust-product cycle.

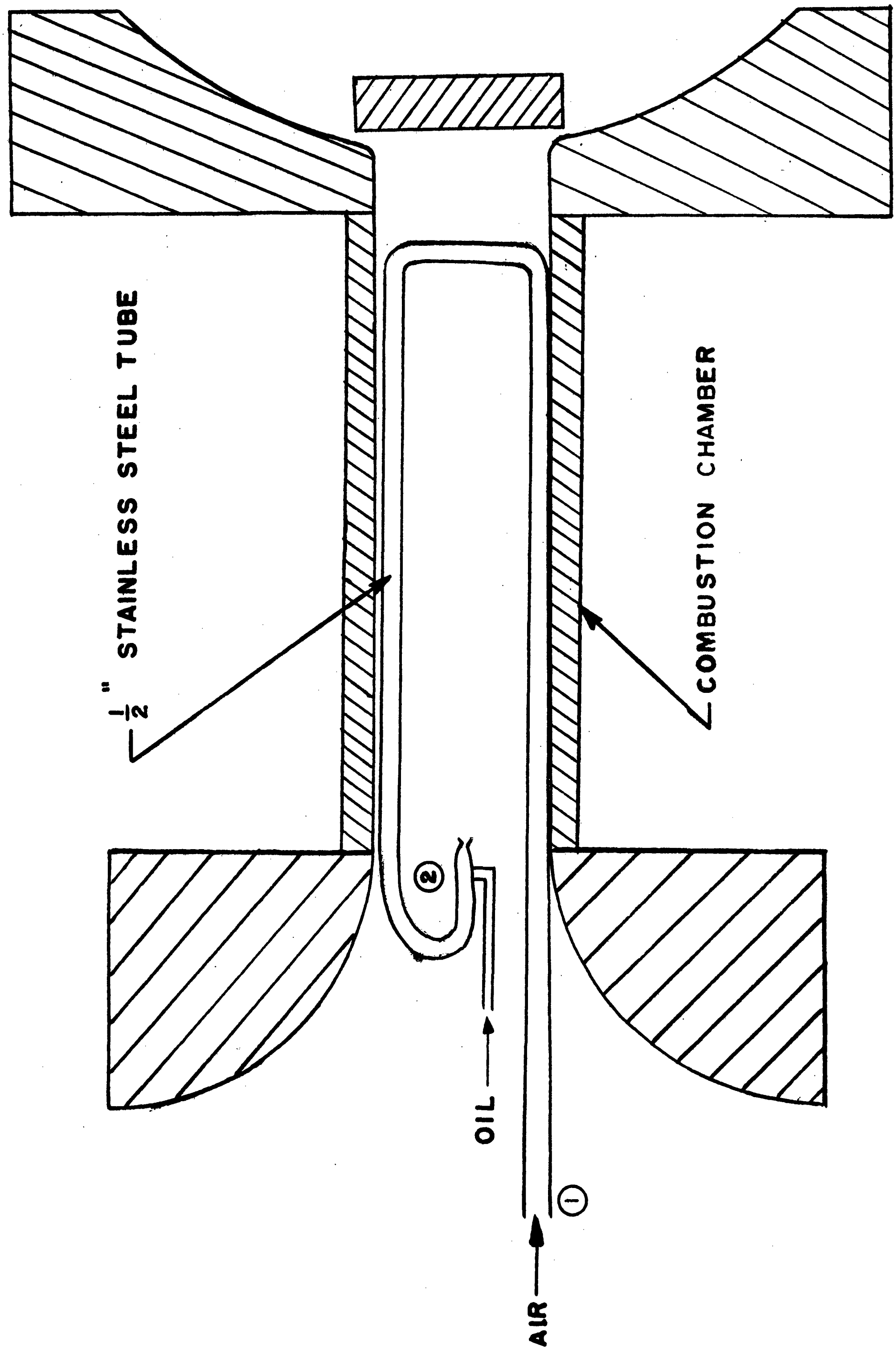

Fig. 11. Preheated-air cycle.

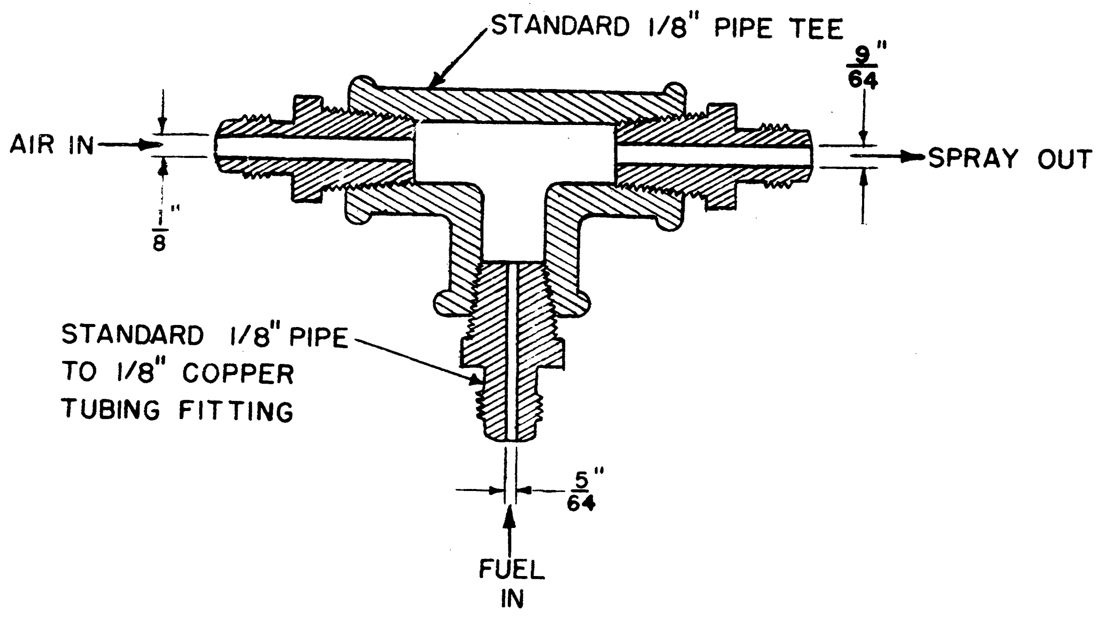

Fig. 12. Pipe-tee nozzle.

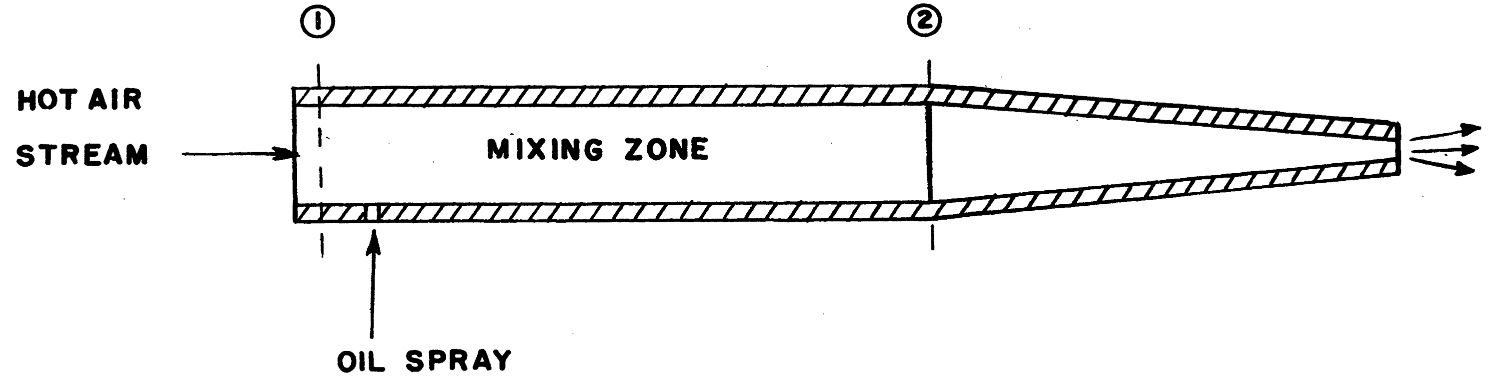

Fig. 13. Fuel-vaporization nozzle.

which is below the 500°F, 50 percent distillation point of the oil. A much hotter stream of air would be required for complete vaporization of the fuel oil, but material problems would undoubtedly exist at higher air temperatures. For 2500°F exhaust-gas-cycle nozzles (eliminating the necessity for heat exchangers inserted in the precombustion chamber) the equilibrium temperature of the fuel vapor and exhaust product gas would be 675°F. If the oil were preheated to 500°F before being fed to the nozzle, the equilibrium temperature would be raised to about 900°F.

In nonvaporizing fuel nozzles (the preheated-air cycle and the exhaust-product cycle) the stream would have a higher temperature and hence a higher sonic velocity compared to vaporizing nozzles where the temperature and sonic velocity would be lower. In the second case, however, the mass flow would be greater and quite possibly the overall momentum of the stream, and hence the amount of air induced, would also be greater. The use of completely vaporized fuel oil might offer many advantages. Unfortunately, this research program was cancelled before experiments could be performed with this type of nozzle.

CONCLUSIONS

1. Experiments performed on a water table show recirculation within the present Selas Burner cups because of the ejector action at the cup inlet.

2. Using 60°F air, it is possible to induce 86 percent of the air required for stoichiometric burning of No. 4 fuel oil in a radiant burner, and achieve radiant cup temperatures.

3. Using 1100°F air, it is possible to induce 93 percent of the air required for stoichiometric burning of No. 4 fuel oil in a radiant burner, and achieve radiant cup temperatures.

4. It is possible to operate a combustion chamber under a pressure of 30 psig and achieve a high-velocity high-temperature stream through the 1/8-inch-diameter inlet to a fuel-atomization nozzle.

5. The use of either fuel-vaporization nozzles or the exhaust-product cycle presumably would induce more air than the 93 percent of the total air induced with the preheated-air cycle. The amount of air actually induced using these processes was not measured.