APPLICATIONS OF BAFFLES AND EXHAUST ENERGY TO MOTORCYCLE CYLINDER COOLING

by

Irenee duPont, Jr.

Submitted in Partial Fulfillment of the Requirements for the Bachelor of Science Degree in Mechanical Engineering

from the

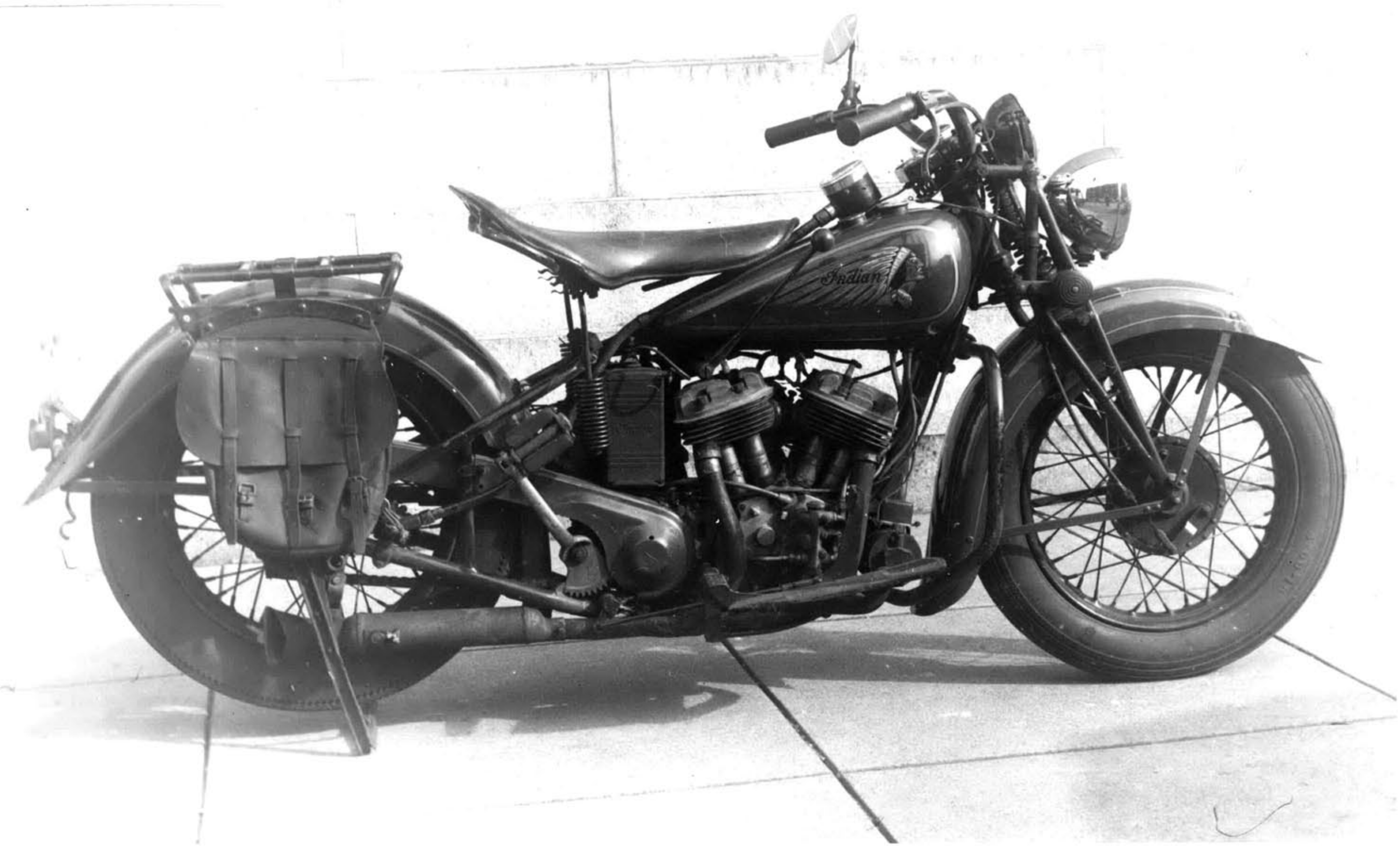
Massachusetts Institute of Technology

1 9 4 3

Acceptance:

Instructor in Charge of Thesis, ____________________

(Date) Jan 19 194[illegible]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PURPOSE OF THE INVESTIGATION

The original purpose of this investigation was to develop an improved method of cooling motorcycle cylinders and compare this method with the conventional system. The problem immediately became one of building a sheet metal baffle to be placed over the cylinders. Next a pressure difference was required to move the air beneath the baffle. Many experimenters have shown conclusively that a correctly designed baffle both increases the heat transfer and improves the temperature distribution on an aircraft cylinder.

The author has endeavored to build a device capable of maintaining a draft beneath the baffle. It also seemed important that such equipment could be easily attached to any motor cycle if possible. Hence centrifugal blowers and fans were abandoned due to problems in drive mechanism and high cost of production. An aspirator or ejector device fitted on the exhaust system seemed most desirable if it could be made to work effectively. Having no moving parts, it could be easily fabricated and would make use of energy normally wasted at the exhaust pipe exit. So as the effort progressed, emphasis fell to a greater degree on the utilization of exhaust energy to produce the necessary pressure difference beneath the cylinder baffle. Since ordinary motorcycle cylinders are most irregular in shape and form the center of a blossom of accessories, the fitting of a correctly designed baffle was found well-nigh impossible.

Therefore it was decided to build the best possible baffle under the circumstances and spend the greater effort in constructing an effective aspirator device.

REASON FOR SELECTION OF SUBJECT

The problem of cooling motorcycles has never been adequately solved. American manufacturers have favored air-cooling chiefly due to reduction of costs and secondly due to diminished weight. Motorcycles, though produced in small quantities, must undersell by a wide margin the lowest priced automobiles. Since the demand for motorcycles is almost as small as the competition between manufacturers, much has been lacking in even the fundamentals of research in their problems. For this reason motorcycling opens a vast field for amateur experimentation.

Many of the common ailments of motorcycles arise from overheating of the engine. In every motorcycle repair shop is a box full of ruptured pistons. Evidence of such failures extend from slight scuffing of skirts all the way to melting away of one side of the aluminum alloy pistons. Motorcycle owners are all too familiar with frequent refinishing of cylinder barrels, the occasional cracking of the barrel casting between the cylinder wall and exhaust port, the breakage of piston rings, the short life of lubricating oil, worn out valve guides. All such inconveniences recur with regularity that would be more than intolerable to an automobile operator.

These failures occur at both extremes of road speed operation. At the high end of the speed range such troubles are in some ways excusable since it is more often the driver than the engine parts which needs replacing. On the other hand many an interested observer has been impressed by the destruction which overcomes police motorcycles used on parade escourt duty. Here the engine does little more than idle for several hours while the road speed is but a walking pace. Nevertheless these engines have troubles comparable to the ninety-mile-an-hour specialist.

Further evidence of overheating is seen on mountain roads where driving conditions require the use of low speed. The rider first becomes aware of intense combustion knock, then excessive vapors eminating from the crankcase breather. If the machine is not permitted to cool off, the pistons will probably seize very shortly. From the foregoing it should be clear that an improved cooling system is a necessity on motorcycles of the future.

The bulk of the problem doubtlessly lies in proper design of cooling fins and location of the exhaust ports. In the present day L head cylinders there is no space for air to pass between the ports and the cylinder walls. Fins are as sparse as four to the inch and seldom are greater than an inch in depth. This is quite a contrast to recent machined fins on aircraft cylinders. Here fins are spaced nine to the inch and in some places attain a depth of over two inches. But even the best designed air-cooled cylinder requires a good blast of air. Though it is impossible for the student to design and build an improved cylinder, he can attempt to construct an air circulating system. It is this phase of the problem dealt with in this investigation.

PREVIOUS DATA INADEQUATE

Although much research has been done on aircraft cylinder and baffle design, no references could be found where an exhaust aspirator had been applied to such a problem. For this reason along with many others the investigation passed over the shaping of the baffle rather lightly and dealt chiefly with aspirator design. Several references to ejectors and aspirators were consulted, but such devices have been employed generally on high pressure equipment. Multi-stage ejectors have been used in steam power plants which are capable of producing vacuums of better than 98 per cent atmosphere, but here high pressure steam was used for the jet.

The motorcycle problem is vastly different. Nothing can be permitted to interfere with the free exhausting of the cylinders, hence the velocity of the exhaust must be taken as is. This velocity varies greatly with speed and load conditions and is of a pulsating nature. The complete lack of data on such a specialized problem has left an excellent opening for the student experimenter.

PROPOSED ROAD TEST IMPRACTICABLE

The original experiment was to be simple in character. Its essence was to take temperature readings of the cylinders while driving on the road (a) with normal engine setup and (b) with the baffle and ejector apparatus in operation. Comparing the operating temperatures over the same road course under the two conditions, a qualitative estimate of the merit of the device could be obtained. This phase of the experiment had to be abandoned because of two troubles which developed. First was the impossibility of installing a correctly designed baffle over cylinders not intended for such. Second was the unavailability during the present war effort of a portable galvanometer for measuring the thermocouple electromotive force.

The chief problem in building a suitable baffle arose in fitting exit ducts to the rear of the cylinders. A correctly formed baffle for road use should necessarily pick up the dynamic pressure of the vehicle in motion. This would require the exit duct to be placed at the rear of the cylinders. The rear of one cylinder was obstructed by the inlet manifold while the other was in contact with the frame of the motorcycle and less than half an inch from the storage battery. According to

M. J. Brevoort[1] much of the success of a baffle lies in the generous filleting of the exit duct. All things considered, a correctly designed baffle was impossible without drastic changes in the cylinders and design of the motorcycle.

Many attempts were made to take qualitative readings of the thermocouple with the meters available. All millivoltmeters gave less than one-eighth scale reading at the highest temperatures encountered. Even milliammeters were tried, but all with suitable sensitivity had such high resistances as to reduce the current markedly. No meter available gave better than one-fifth scale reading.

Measurements of exhaust velocity by a Pitot tube were found impossible due to the pulsating of the gases. At certain speeds resonance effects completely reversed the pressure on the manometer so that readings were meaningless.

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[1]Maurice J. Brevoort, "Energy Loss, Velocity Distribution, and Temperature Distribution for a Baffled Cylinder," N.A.C.A. Technical Notes No. 620, Langley Field, 1937.

PART II

Description of Experimental Work

Discussion of Results

Considerations from the Momentum Theory

Conclusions

Suggestions for Future Investigators

DESCRIPTION OF EXPERIMENTAL WORK

Equipment

The experiment was performed on an Indian Motorcycle Model 635 MY. This has an L head engine of 45 cubic inches displacement and was built in 1935. No doubt other models have been built on which such an experiment would have been performed with greater ease and success. However, the choice of machine was dictated entirely by that which happened to be in the author's possession.

A sheet steel baffle was formed and welded up to fit closely over the right side and top of the cylinders. Care was taken that all air entering the baffle should pass along at least five inches of cylinder surface before leaving through the exit duct. The exit duct was located centrally between the two cylinders. Holes were cut in the baffle to accommodate the protruding spark plugs. A four-inch stovepipe connected the exit duct with the aspirator device at the rear of the motorcycle.

The aspirator was of the same welded steel construction as the baffle and permitted the union of the exhaust pipe and exit duct as shown in the diagram of setup. (Figure 1) The Venturi in the aspirator was formed first of Plasticene modeling clay for tests in the laboratory and then replaced by more durable plaster of Paris for tests on the motorcycle.

A Pitot tube reading dynamic pressure in the exhaust pipe was connected to a manometer conveniently located on top of the gasoline tank. A similar Pitot tube and manometer read the dynamic pressure in the exit duct. This Pitot tube was movable across the diameter of the duct in order to pick up the velocity distribution. A temperature gage of the pressure bulb type was located on the gas tank with its bulb in the exit duct.

Temperature measurements were taken on the cylinder by means of a copper-constantan thermocouple formed on a spark plug washer. The cold junction of the thermocouple was placed in a Thermos bottle of melting ice attached to the motorcycle frame beside the engine.

Test on Aspirator

On making up the aspirator many questions arose in its design. The object was to draw the maximum weight of air through the exit duct with a given velocity of exhaust gases in the exhaust pipe. It was therefore necessary to experiment with Venturi sizes and distances of exhaust pipe terminus from the Venturi throat. To facilitate this, the aspirator was set up in the laboratory. The laboratory supercharger system was connected to the exhaust pipe jet, and standard orifices were placed in the exit duct to simulate the conditions of a baffled cylinder. The air jet was adjusted in its location with respect to the Venturi. Venturi sizes and forms were molded in Plasticene. For each Venturi size and for various rates of duct flow readings of vacuum were taken as the distance between jet and Venturi throat was varied.

Rate of Temperature Rise

The motorcycle was permitted to stand with engine idling. Starting with the engine cold, the temperature of the spark plug gasket was taken every thirty seconds. This was done twice, once with the baffle and aspirator set up and once with no more provision for cooling than the conventional unbaffled cylinder. By plotting a curve of temperature versus time, the rate of temperature rise would be given by the slope of the curve. The curve with the least slope would then signify the better cooling condition. This test had two destinct advantages: (1) speed and load conditions were easily held constant by throttle setting alone; and (2) good thermocouple readings were made possible by the use of a Leeds and Northrup potentiometer. Such an instrument could not be used on the road for obvious reasons.

Copper - Constantan Thermocouple

A simple and significant location for the thermocouple used in measuring cylinder temperature was the spark plug gasket. This region was well under the baffle and should show the cooling effect of the apparatus as well as any spot on the cylinder structure. Here the couple was easily accessible and was in plain view in case a poor connection should develop.

The thermocouple circuit was as follows: One copper lead was silver soldered to the solid copper spark plug gasket. A constantan wire was also silver soldered to the gasket and ended in a soldered joint with the other copper lead to form the cold reference junction. This junction was kept in a bath of melting ice. Both copper leads were connected to a Leeds and Northrup potentiometer.

The thermocouple and potentiometer were then calibrated against a standard mercury in glass thermometer. Using the thermometer as a stirring rod, a water bath containing the hot junction was slowly heated to boiling and allowed to cool off again. Potentiometer and thermometer readings were taken simultaneously about every ten degrees during both heating and cooling processes. A similar procedure was carried out using an oil bath to cover temperatures between 212° and 305° F. A transfer chart was made of electromotive force plotted against temperature. (Figure)

DISCUSSION OF RESULTS

Duct Vacuum vs. Distance from Jet to Venturi Throat

In one test on the aspirator the vacuum in the exit duct was measured for varying distances of the exhaust pipe end and the Venturi throat. (Figure 2) The flow in the duct was blocked and the velocity of the jet was held constant. For the four sizes of Venturies tested a surprising result was obtained. The vacuum increased as the distance between jet and throat was lengthened up to five inches. Then suddenly the vacuum fell off as an apparently critical distance was reached. This point of maximum vacuum was found to be independent of jet velocity and Venturi size.

The cause of vacuum increasing with distance between jet and throat is reasonable. One would expect better transfer of momentum of the high velocity gases to the gases from the exit duct as this distance increased because of the greater opportunity for the two streams to mingle.

However, the reason for the sudden reverse of this effect is not so evident. It is natural to expect some sort of limit on the optimum distance between jet and throat, otherwise there would be no virtue in having a Venturi at all. As increasing the distance brings benefits due to better mixing of the two streams, it also introduces greater friction losses due to the lengthening of the pipe. Lengthening the pipe, however, could not produce so great an effect as to completely reverse the slope of the curve by the addition of a couple of inches. The exceeding sharpness of this optimum jet location can only arouse a suspicion that some other factor is creeping in. It may be the result of an abrupt curve in the exit duct where it meets the exhaust pipe jet. When the jet was withdrawn to the critical position, a pressure wave may have been started which rebounded from the oblique wall of the exit duct and broke up the flow. The effect of such irregular factors could be eliminated by a slight change in design of the aspirator as noted in Figure 2 under suggestions for further investigation.

Vacuum vs. Airflow

A test was run showing variations of duct vacuum with air flow for different Venturi sizes. Jet to throat distance was held at five inches in order to give maximum vacuum readings. Jet velocity was kept at 126 feet per second, a convenient standard of the order of magnitude of actual exhaust conditions. It was found for the three Venturi sizes, 2.00, 2.30, and 2.75 inches, that the vacuum fell off with air flow at about the same rate, the curves (Figure 3) being nearly parallel. For the 3.00 inch Venturi a definite decrease in the slope of the curve was prominent. Evidently the Venturi throat was a point of pressure drop in the system. As the Venturi size was increased, a point was reached where the friction loss in the restriction became small compared to that at the thin plate measuring orifice. Although the 3.00 inch Venturi was not as efficient in producing a vacuum for no-flow conditions, it upheld the vacuum better in the large flows by causing less frictional disturbance.

Vacuum vs. Jet Velocity

Variations of exit duct vacuum (no-flow conditions) with the dynamic pressure of the exhaust jet as measured by a Pitot tube were found and plotted. Three Venturi sizes, 2.00, 2.30, and 2.75 inches were tried and the resulting curves were nearly straight lines. Curves for the larger Venturies had the steeper slope. This confirmed the results of other tests that the 2.75 inch Venturi could draw the greatest vacuum.

This linear relation of vacuum with jet dynamic pressure also illustrated that the aspirator operated on the principle of conservation of momentum.

(1) $$M_E V_E = (M_E + M_H) V_H$$

where

$M_E = \frac{\text{mass}}{\text{second}}$ exhaust

$M_H = \frac{\text{mass}}{\text{second}}$ cooling air and exhaust

V_H = velocity of cooling air

V_E = velocity of exhaust

For the flow of air through an orifice or pipe

(2) $$M = K_1 \sqrt{\Delta p}$$

where

M = mass rate of flow

Δp = pressure difference

K_1 = a constant

For flow of ideal incompressible fluids through a pipe of uniform cross section

(3) $$Q = AV \qquad M = \rho VA$$

(4) $$M = K_2 V$$

where

Q = volume rate of flow

ρ = mass density

V = velocity of flow

A = area of pipe cross section

M = mass rate of flow

K_2 = a constant

For a Pitot tube

(5) $$V = K_3\sqrt{h}$$

where

h = dynamic pressure of flowing fluid

From equations (4) and (5)

$$M_E \backsim \sqrt{h}$$

From equation (2)

$$M_H \backsim \sqrt{\Delta p}$$

From the experiment exit duct vacuum is proportional to jet dynamic pressure

$$\Delta p \backsim h$$

$$M_E \backsim M_H$$

From equation (4)

$$M_E \backsim V_E \qquad (M_E + M_H) \sim V_H$$

therefore

$$M_E V_E \backsim (M_H + M_E) V_H$$

The Pitot tube readings of exhaust dynamic pressure were transferred to velocities and another set of curves were drawn showing variation of vacuum versus jet velocity. This illustrated the parabolic relationship of vacuum varying with the square of the velocity.

Rate of Temperature Rise

The conventional engine setup was started cold and permitted to idle at 1000 revolutions per minute as the temperature of the spark plug gasket was taken every 30 seconds. The same process was repeated a second time with the aspirator and baffle in place. When the transient temperature was plotted against time for the two "warm-ups", the slope of the curve for the unbaffled cylinder was found to be the steeper. This indicated that for idling conditions where no draft was produced by the forward motion of the machine the device actually was a benefit. Computations have shown that the device could not give satisfactory cooling for conditions of maximum power output without assistance from road air velocity. The experiment confirmed this conclusion for idling conditions since the temperature was still rising when the test was stopped. There was no evidence to indicate that an equilibrium temperature would have been reached before the engine had become badly overheated.

CONSIDERATIONS FROM THE MOMENTUM THEORY

Assume the engine is operating at full output. (20 brake horsepower) The momentum of the exhaust gases leaving the jet must equal the momentum of the exhaust and cooling air leaving the aspirator. The momentum equation is :

$$M_E V_E = (M_E + M_H) V_H$$

Computations in Appendix (A) show that

M_E	=	mass rate of flow of exhaust	$= .00155 \ \frac{slugs}{sec.}$
M_H	=	mass rate of flow of cooling air	$= .031 \ \frac{slugs}{sec}$
V_E		velocity of exhaust	$= 248 \ \frac{ft}{sec}$

The maximum velocity that the exhaust gases can geve to the cooling air is:

$$V_H = \frac{M_E V_E}{M_E + M_H} = \frac{(.00155)(248)}{.00155 + .031} = 12.2 \ \frac{ft}{sec}$$

To get the required mass rate of flow of cooling air at this velocity would require a venturi whose throat area is:

$$A = \frac{Q_E + Q_H}{V_H} = \frac{2 + 15}{12.2} = 1.4 \ ft^2$$

where

Q_E = volumetric flow of exhaust

Q_H = volumetric flow of cooling air

values found in Appendix (A)

The throat diameter of such a venturi would be 16 in.

The space occupied by this apparatus and the impossibility of getting momentum transfer in such an aspirator make these dimensions ridiculous.

Although the above calculation is based on many assumptions, the results, even if in error by 300% show that the velocity of the exhaust gases normally issuing from the pipe is insufficient to cool the engine.

CONCLUSIONS

1. An exhaust aspirator with straight pipe nozzle is impracticable for passing cooling air over motorcycle cylinders under running conditions.

2. The aspirator and baffle were of some benefit under idling conditions.

3. Conventional motorcycle cylinders are unsuitable for baffled cooling

4. An aspirator built on a 1.25 inch exhaust pipe jet should have a venturi throat diameter of 2.75 inches. The throat should be located 5 inches from the exhaust pipe terminus.

5. Thermal energy of the exhaust gases should be used to increase the velocity of the jet.

SUGGESTIONS FOR FUTURE INVESTIGATORS

1. Experiment with more effective nozzles using the thermal energy of the exhaust gases. Nozzle should be built in cylinder port using the valve seat as its throat. The sonic velocity during "blow down" might even be augmented by suitable diverging nozzle.

2. Experiment with optimum length of aspirator casing on down-stream side of venturi.

3. Attempt no road tests.

4. Set motorcycle up on dynamometer and run all tests made here using actual exhaust jet. The pulsating flow does unpredictable things in resonance.

PART III

APPENDIX

A. Computations

B. Figures, Data, Graphs

C. Bibliobraphy

D. Photographs of Equipment

COMPUTATIONS

Mass rate of flow of cooling air required

Engine output 20 horsepower (brake)

Assume heat rejection through cylinder walls and head equal to brake horsepower output. Heat rejection:

$$H = 20 \text{ h.p.} = 14.2 \text{ B.T.U./sec.}$$

Assume $\Delta t = 60^{\circ}$ F. temperature rise of air passing through baffle

Spacific heat of air at constant pressure C_p = BTU/#°F.

$$H = M C_p \Delta t$$

$$M = \frac{H}{C_p \Delta t} = \frac{14.2}{(.24)(60)} = .98 \text{ \#/sec.}$$

Volumetric rate of flow of cooling air required

Assume mass density ρ of air at 130°F = .002 slugs/cu.ft.

weight density = .064 #/cu. ft.

$$Q = \frac{M}{\rho g} = \frac{.98}{.064} = 15.5 \ \frac{\text{cu.ft.}}{\text{sec}}$$

Required velocity through 4 inch exit duct

Area of 4 inch pipe = .0872 square feet

$$V = \frac{Q}{A} = \frac{15.5}{.0872} = 178 \ \frac{\text{ft.}}{\text{sec.}}$$

Air capacity at 20 h.p. out-put

Assume brake spacific fuel consumption of .6 pounds per brake horsepower hour.

Assume fuel air ratio of .07

$$\text{air capacity } M = \frac{.6}{.07} = 8.58 \quad \#\text{air/B.H.P. Hr.}$$

$$M = (20)(8.58) = 172 \quad \#\text{air/Hr.}$$

$$M = .0477 \ \#\text{air/sec.}$$

By conservation of matter weight of fuel per second plus weight of air per second must equal weight of exhaust per second.

Mass rate of flow of exhaust

#air per second	.0477
#fuel per second	.0033
# exhaust per sec.	.051

Spacific volume of exhaust gases

Assume mols exhaust = 1.06 mols inlet gas

assume exhaust temperature 1200°F

$$\text{molecular weight exhaust} = \frac{28.96}{1.06} = 27.3$$

$$\text{R exhaust} \quad \frac{1544}{27.8} = 57.6$$

$$v = \frac{R\ T}{P}$$

spacific volume $v = \dfrac{(56.6)(1660)}{(144)(14.7)} = 44.4 \ \dfrac{ft^3}{\#}$

Volumetric flow of exhaust

$$Q = Mv = (.05)(44) = 2.2 \ \frac{ft^3}{sec}$$

Velocity of exhaust gas

Area of 1.25 inch pipe $=$ .0085 square feet

$$V = \frac{Q}{A} = \frac{2.2}{.0085} = 248 \ ft/sec$$

SCHEMATIC DIAGRAM OF SET UP

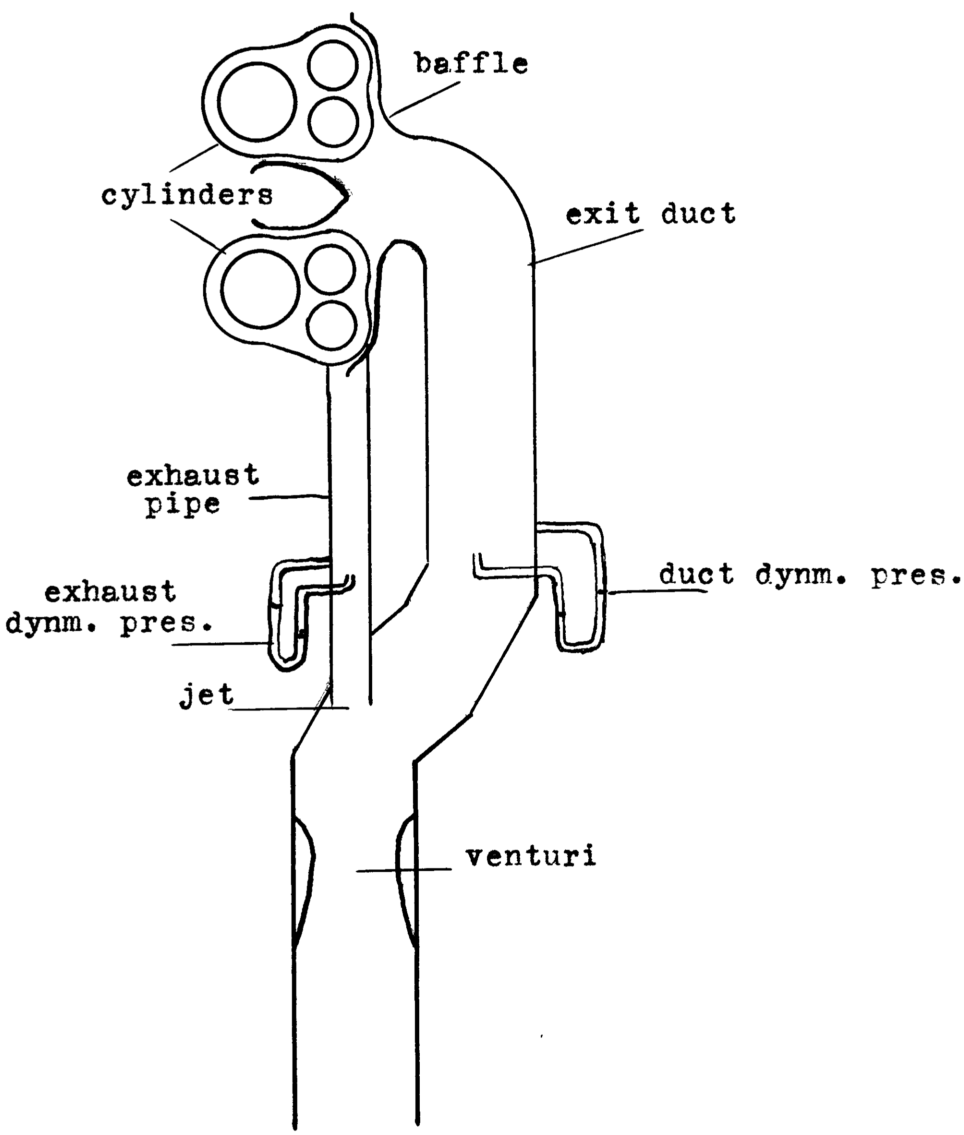


Fig 1

VACUUM vs. DISTANCE JET TO THROAT

for 4 throat diameters

Jet velocity 126 feet per sec.

distance "	2.00"	2.30"	2.75"	3.00"
1	.27	.38	.30	.30
2	.28	.42	.37	.33
3	.35	.48	.45	.36
4	.40	.50	.52	.44
5	.49	.53	.58	.40
6	.23	.45	.55	

VACUUM vs. DISTANCE
JET TO THROAT

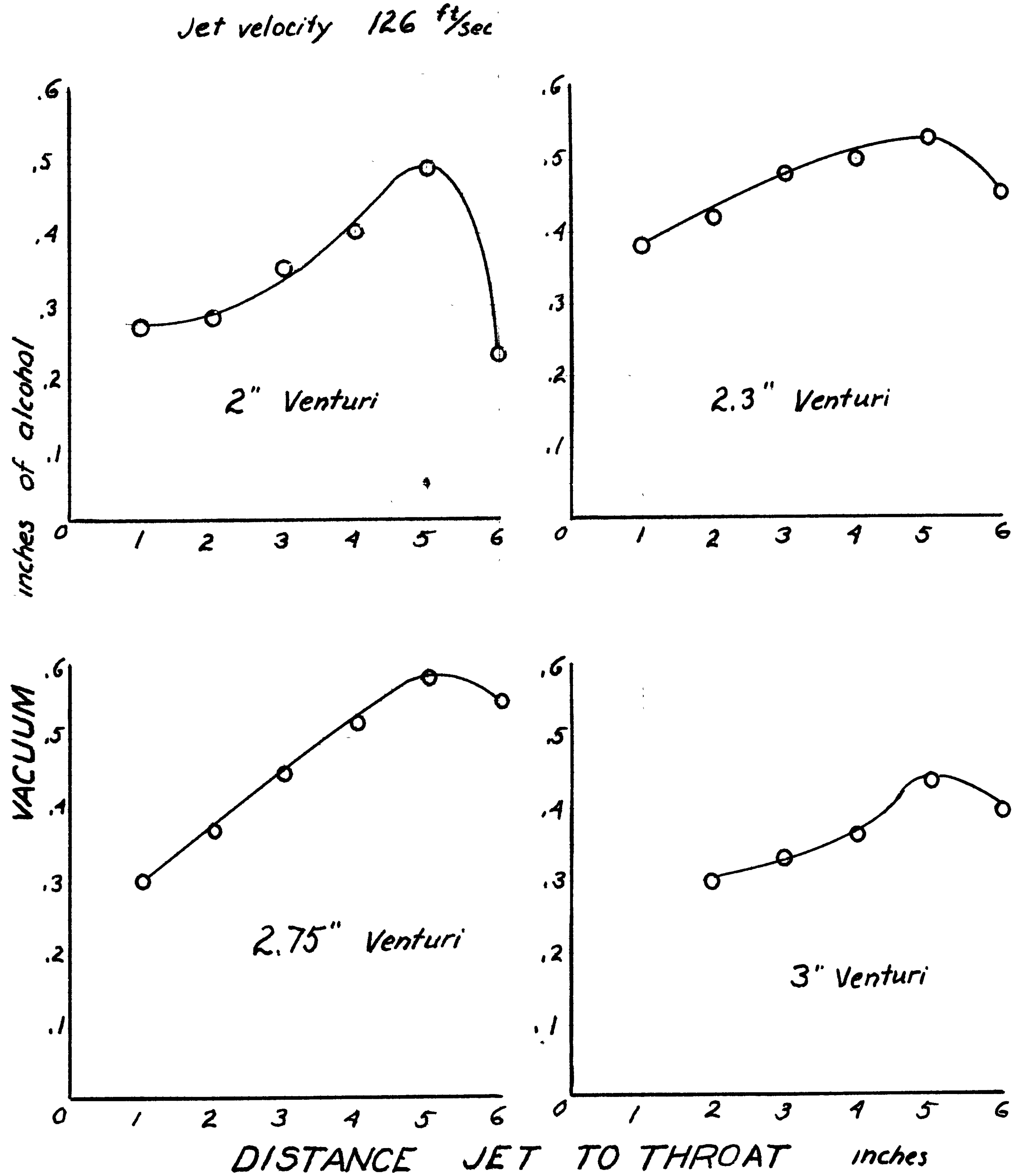


Fig 2

VACUUM VS. DUCT AIR FLOW

Jet Velocity 126 ft/sec. 5" Jet to Throat

$$W = .0308\, A \sqrt{h}$$

W = LBS. air per second; A = area of orifice

h = vacuum inches of alcohol

orafice diameter	vacuum	air flow	orafice diameter	vacuum	air flow
2.00 in. throat			2.30 in throat		
.5	.38	.0037	.5	.52	.0043
1.0	.31	.013	1.0	.44	.016
1.5	.22	.025	1.5	.32	.031
2.0	.18	.041	2.0	.23	.046
2.5	.12	.052	2.5	.14	.057
2.75 in. throat			3.00 in. throat		
.5	.56	.0045	.5	.4.4	.004
1.0	.48	.017	1.0	.38	.015
1.5	.40	.034	1.5	.33	.031
2.0	.29	.052	2.0	.25	.048
2.5	.15	.059	2.5	.20	.068

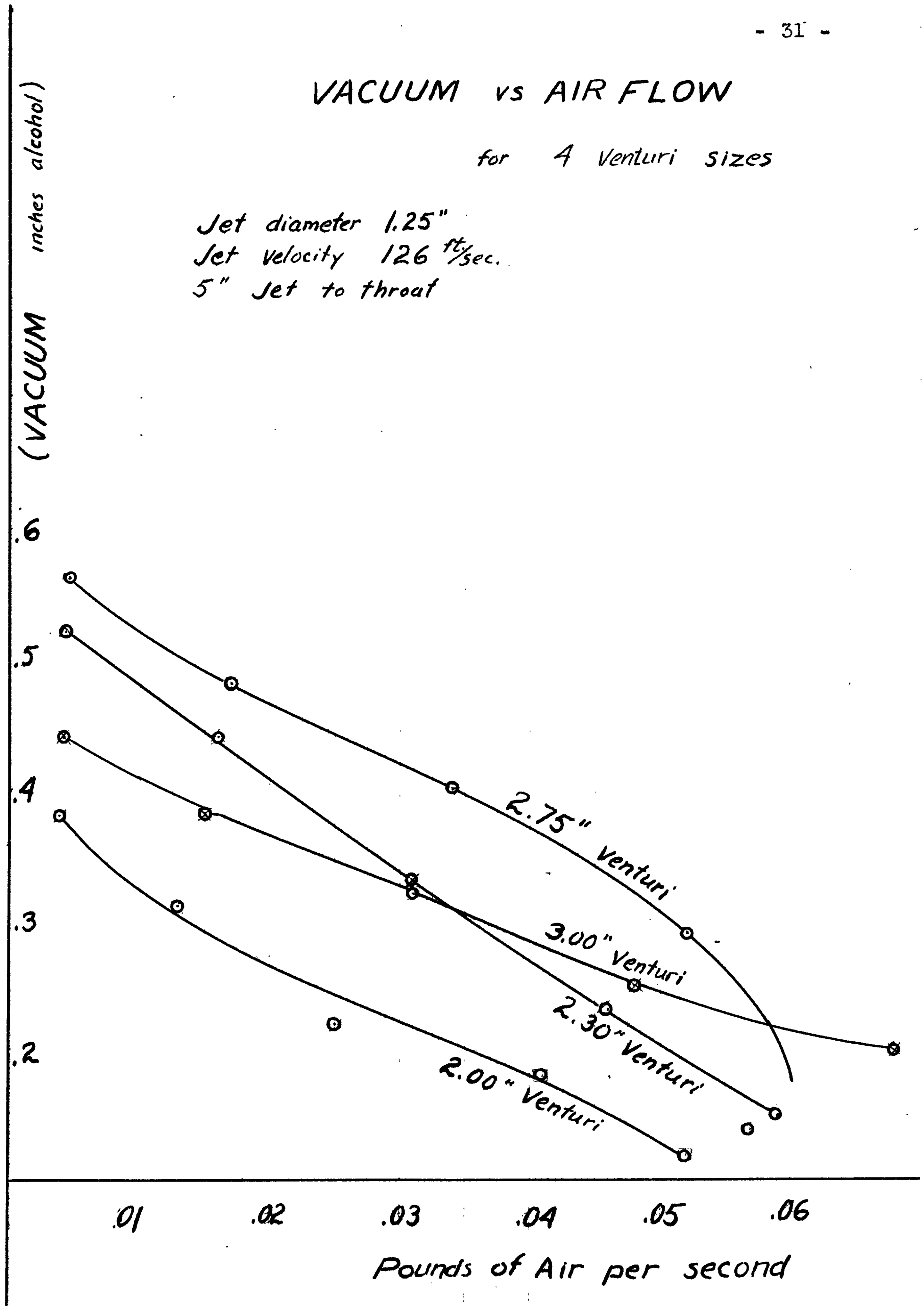

VACUUM vs AIR FLOW
for 4 Venturi sizes
Jet diameter 1.25"
Jet Velocity 126 ft/sec.
5" Jet to throat
(VACUUM inches alcohol)
.6
.5
.4
.3
.2
2.75" Venturi
3.00" Venturi
2.30" Venturi
2.00" Venturi
.01
.02
.03
.04
.05
.06
Pounds of Air per second

Fig 3

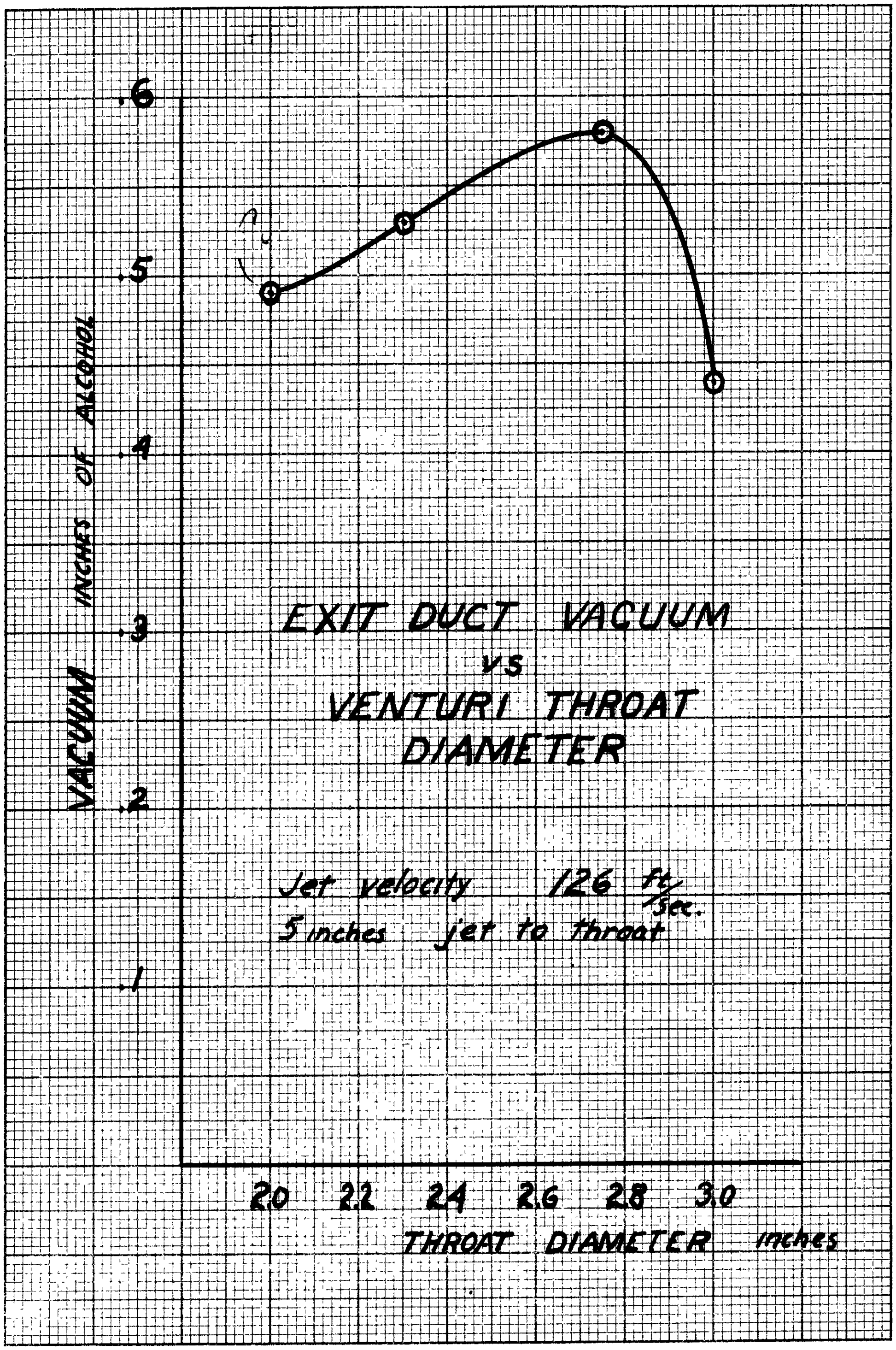
VACUUM vs. VENTURI DIAMETER

Jet diameter 1.25 inches; Jet velocity 126 feet per second

5 inches jet to throat

Exit duct air flow blocked

vacuum " alcohol	venturi diameter "
.49	2.00
.53	2.30
.58	2.75
.44	3.00



EXIT DUCT VACUUM
vs
VENTURI THROAT
DIAMETER
Jet velocity 126 ft/sec.
5 inches jet to throat
VACUUM inches of ALCOHOL
.6
.5
.4
.3
.2
.1
2.0
2.2
2.4
2.6
2.8
3.0
THROAT DIAMETER inches

VACUUM vs. JET VELOCITY

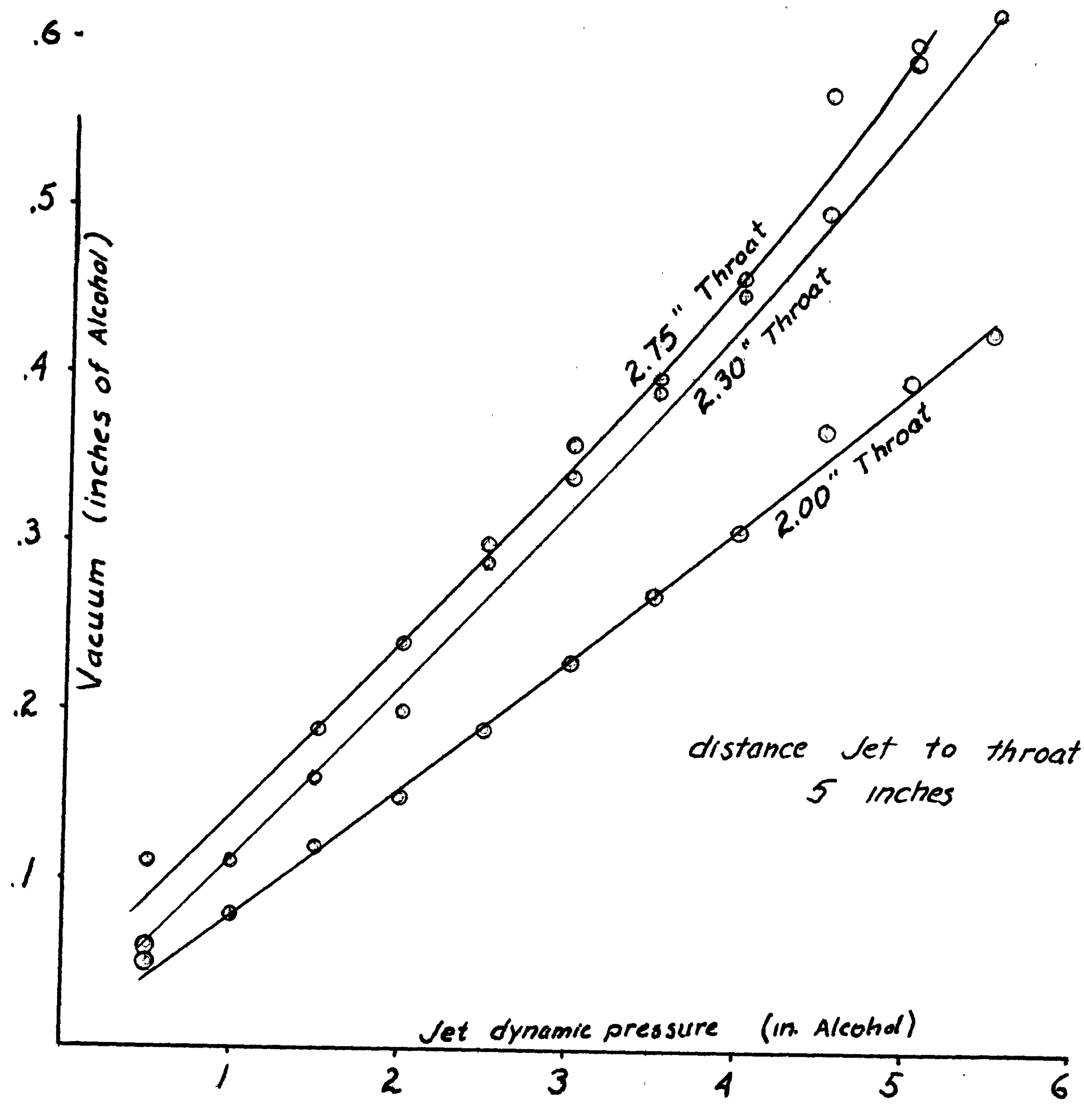
Exit Duct Flow Blocked.

Distance Jet to Throat
5 inches

Jet		Vacuum (inches Alcohol)		
dynamic pressure " Alco'l	velocity feet per second	2.00 " venturi	2.30 " venturi	2.75 " venturi
1/2	41.9	.05		.06
1	59.4	.08	.11	.11
1 1/2	72.6	.12	.16	.19
2	84.0	.15	.20	.24
2 1/2	94.0	.19	.29	.30
3	103	.23	.34	.36
3 1/2	111	.27	.39	.40
4	119	.31	.45	.46
4 1/2	126	.37	.50	.57
5	133	.40	.59	.60
5 1/2	139	.43		.62

Exit duct vacuum
vs.
Exhaust jet dynamic pressure

for three venturi sizes



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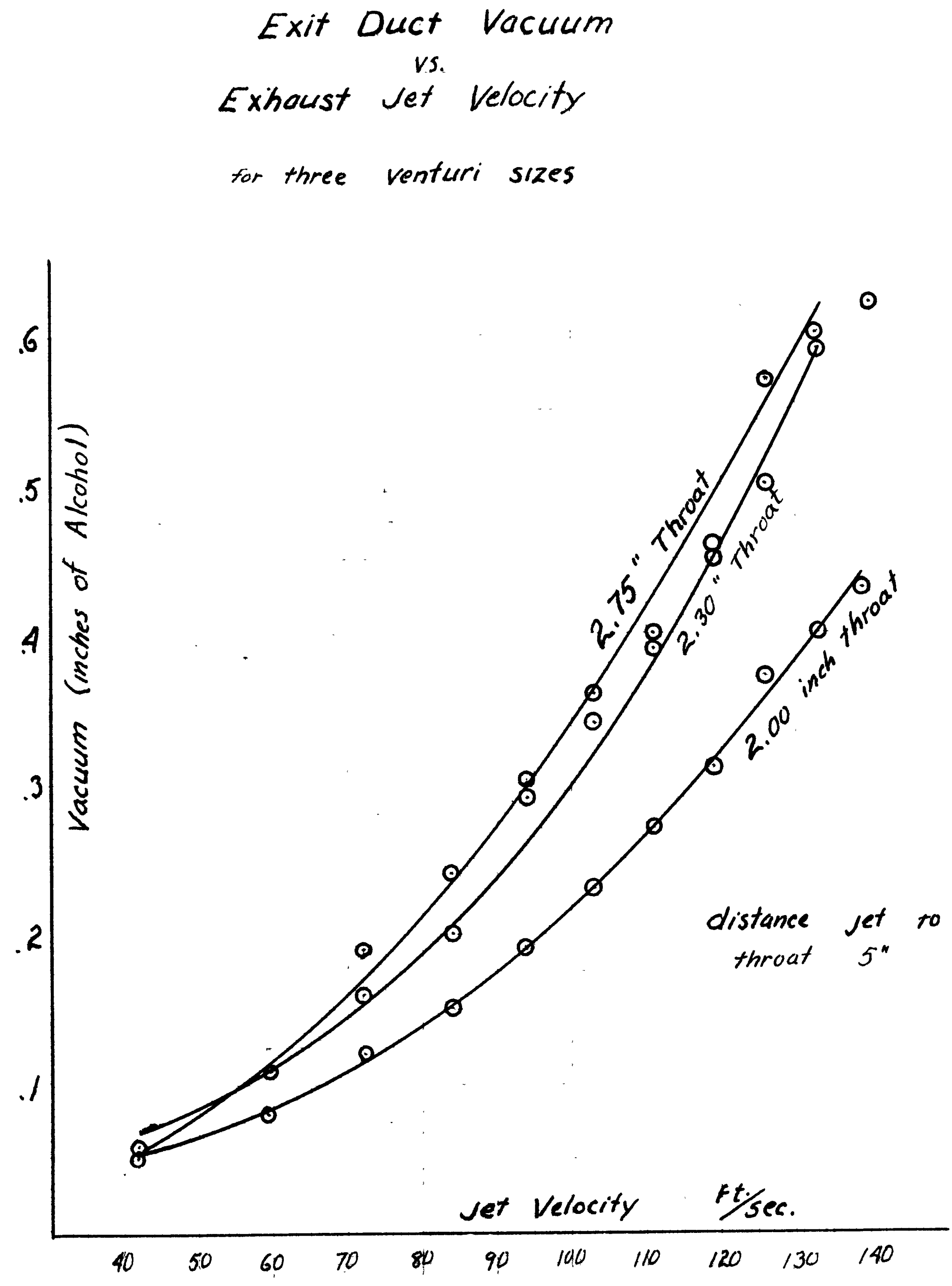


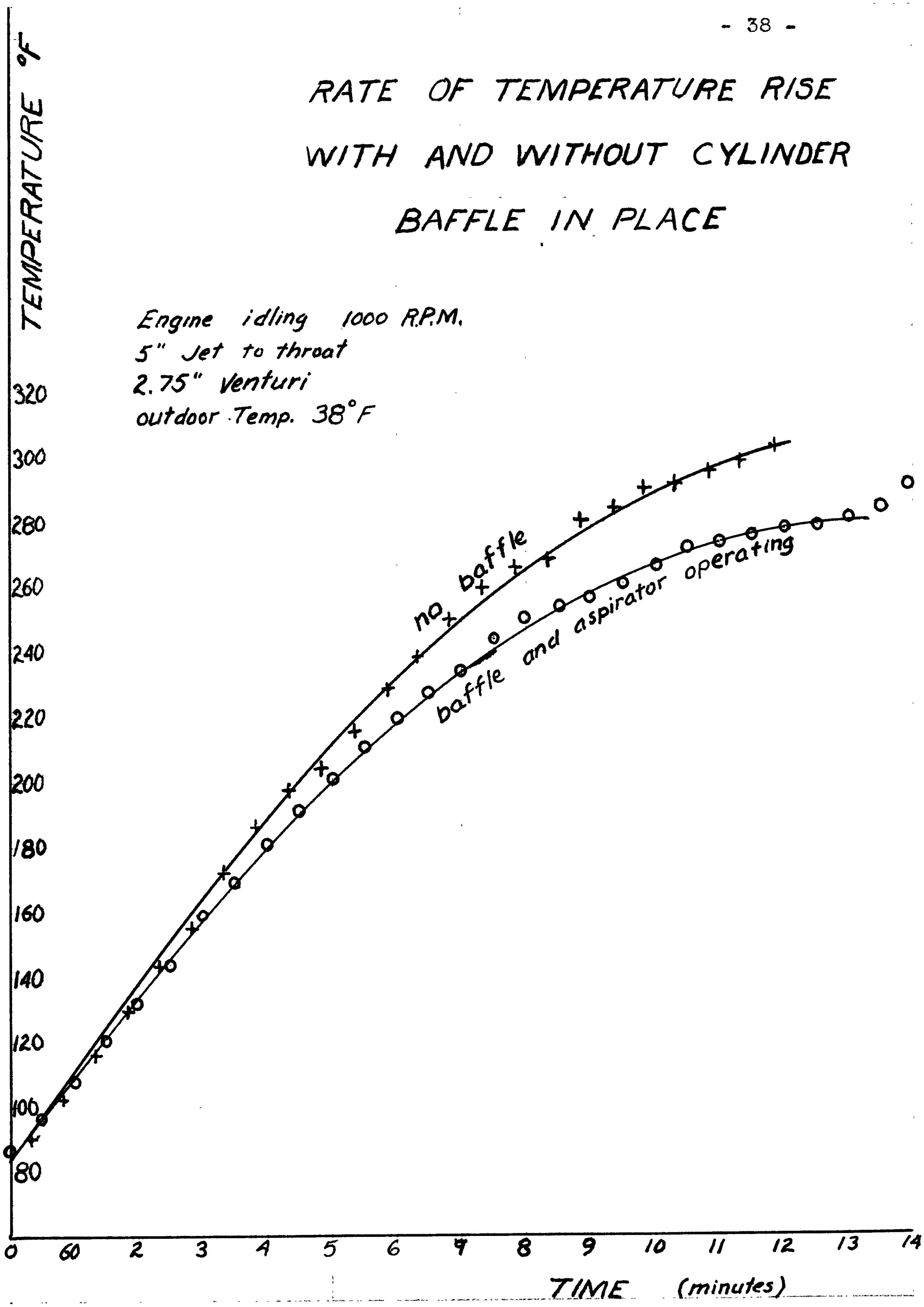
Exit Duct Vacuum
vs.
Exhaust Jet Velocity
for three venturi sizes
.6
.5
.4
.3
.2
.1
Vacuum (inches of Alcohol)
2.75" Throat
2.30" Throat
2.00 inch throat
distance jet to throat 5"
Jet Velocity Ft/sec.
40
50
60
70
80
90
100
110
120
130
140

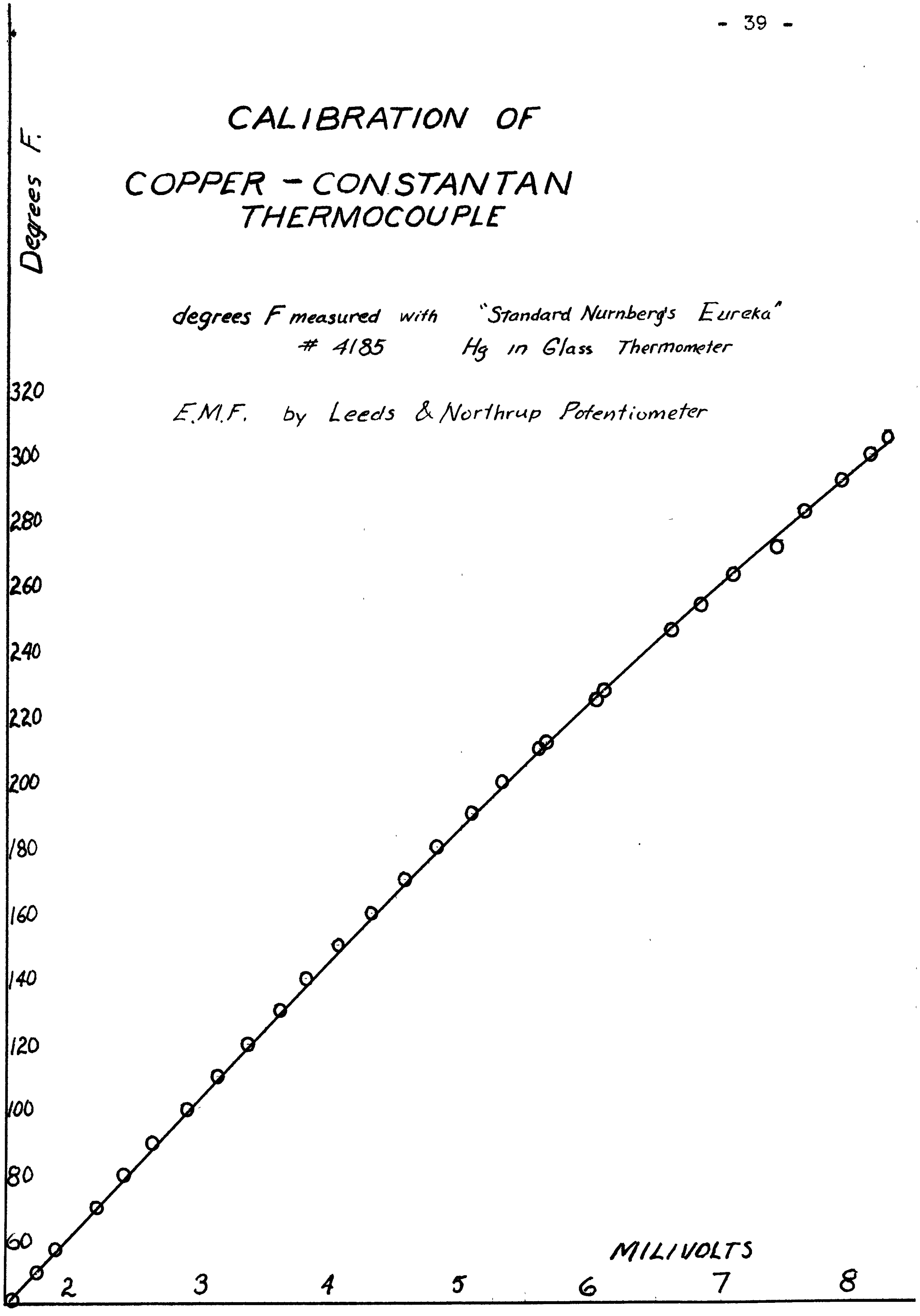
Dec 19 1942

RATE OF TEMPERATURE RISE

Engine idling 1000 R.P.M.; 5" jet to throat; 2.75" venturi; outdoor temperature 38°F.

BAFFLED CYLINDER				UNBAFFLED CYLINDER			
TIME	TEMP.	TIME	TEMP.	TIME	TEMP.	TIME	TEMP.
0	86	8½	254	0	57	8½	260
½	96	9	257	½	66	9	267
1	108	9½	261	1	85	9½	269
1½	120	10	266	1½	91	10	281
2	132	10½	272	2	103	10½	285
2½	144	11	274	2½	117	11	291
3	159	11½	276	3	130	11½	292
3½	169	12	278	3½	144	12	296
4	181	12½	279	4	160	12½	299
4½	191	13	282	4½	173	13	304
5	201	13½	285	5	187	13½	
5½	211	14	292	5½	198	14	
6	220	14½	295	6	205	14½	
6½	228	15	292	6½	216	15	
7	235	15½	296	7	229	15½	
7½	242	16	296	7½	239	16	
8	250	16½	299	8	250	16½	


RATE OF TEMPERATURE RISE
WITH AND WITHOUT CYLINDER
BAFFLE IN PLACE
Engine idling 1000 R.P.M.
5" Jet to throat
2.75" Venturi
outdoor Temp. 38°F
TEMPERATURE °F
320
300
280
260
240
220
200
180
160
140
120
100
80
no baffle
baffle and aspirator operating
0
60
2
3
4
5
6
7
8
9
10
11
12
13
14
TIME (minutes)


CALIBRATION OF
COPPER - CONSTANTAN
THERMOCOUPLE
degrees F measured with "Standard Nurnberg's Eureka"
4185 Hg in Glass Thermometer
E.M.F. by Leeds & Northrup Potentiometer
Degrees F.
320
300
280
260
240
220
200
180
160
140
120
100
80
60
MILIVOLTS
2
3
4
5
6
7
8

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Indian


Indian