A PRELIMINARY INTAKE MANIFOLD INVESTIGATION

Thesis for the Degree of M. S.

MICHIGAN STATE COLLEGE

Sydney Alvin Olsen

1951

This is to certify that the

thesis entitled

A PRELIMINARY INTAKE MANIFOLD INVESTIGATION

presented by

SYDNEY A. OLSEN

has been accepted towards fulfillment of the requirements for

M.S. **degree in** MECHANICAL ENGINEERING

[signature]
Major professor

Date June 25, 1951

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A PRELIMINARY INTAKE MANIFOLD INVESTIGATION

By

Sydney Alvin Olsen

A THESIS

Submitted to the School of Graduate Studies of Michigan State College of Agriculture and Applied Science in partial fulfillment of the requirements for the degree of

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ACKNOWLEDGEMENT

The author wishes to express his sincere appreciation to Professor G. W. Hobbs for his invaluable contributions to this investigation, and to Mr. Fredric J. Warrell for his assistance in many phases of the experimental work.

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INTRODUCTION

The experimental work upon which this paper is based is, as the title indicates, merely of a preliminary nature. The investigation is concerned with the measuring of the pressure variations along the branches of the automotive intake manifold. An emperical approach was made to the problem in an effort to discover the effect these pressure variances have upon engine performance.

After considerable primary research, it was presupposed that the liquid fuel state present in the manifold will settle out of the fluid flow in the region of highest pressure. Should the high pressure points be removed, then the liquid ends would tend to stay in suspension and distribute themselves to the cylinders more evenly.

Ethyl Research Laboratories in Detroit have a clear plastic intake manifold on one of their experimental automotive engines. Upon observation of the mixture flow, it was noted that the liquid portions settled out at the bends in the branches and collected on the inner, or low pressure sections. Most of our modern in-line engines utilize siamese intake ports. Therefore, the liquid that settles out tends to enrichen either one or the other of the cylinders, promoting bad fuel distribution and engine roughness.

The primary research that was mentioned, brought out the fact that manifolding and fuel distribution are one of the big problems in multicylinder engine design. Many tests have been devised to determine the extent of the poor distribution, and many hours spent in experimenting with manifold sizes, shapes, and with the bends in the branches. A summary of these findings from various books, journals, and observations is included in the appendix as background material.

As for the particular investigation that this paper is based upon, not much published material could be found as a foundation. All of the work included here is original as far as is known - speaking with reference to the pressure-variance angle of approach. The apparatus used was simple in design and limited in range, but it was extremely effective in pointing out the suspected results. It is felt that this is strictly a preliminary investigation. There is much more work to be done over a wider speed range and with greater variables. The very idea of having a three-phase fluid flow (air, vapor, liquid) existing in the manifold demands further investigations upon its behavior at a multitude of conditions.

The results herein are taken from just two conditions, that is, using only two manifolds of different design. The first intake manifold was of a standard production design, unaltered in any fashion. Manifold Type I, as it is referred to in the text, is a design as far radical from standard as could be accomodated on the engine. The test engine, or engines, to be more correct as two were used, were of a standard six-cylinder, overhead-valve design in mass production today. The main experimenting engine was new and all dynamometer runs were made with it. The secondary engine was worn by 35,000 miles of normal passenger car use. It was used only as a check against the main one during one test phase to make sure that the results were not peculiar to one engine, as is often the case in automotive work. The engine type was not chosen for any particular reason of performance, but was used because its design is indicative of current trends, i.e., over-head valves, and because it was convenient and simple to work with.

OBJECT

The object of this experiment is to investigate the effect of pressure variances within the intake manifold upon automotive engine performance.

PROCEDURE

Before any test runs were made, the new engine was completely disassembled, checked and measured, and carefully reassembled to the manufacturer's specifications. This was done because the engine had previously been used as a tear-down sample for the Automotive Engineering classes at Michigan State College.

The standard intake manifold was drilled and tapped at predetermined positions to accomodate the pressure take-off lines used in the set-up. Figure 1 shows the manifold with plugs in the positions where the pressure taps were made. The nine separate points were connected to a common header through shut-off valves. The valves and single collector were used to simplify the hook-up to the inclined draft gauge used to measure the pressure differentials. In all cases the pressures along the manifold branches are recorded as differentials from a common point just below the carburetor in the intake riser.

The engine was run and pressure variances measured in a series of four steps - full, three-quarter, half, and one-quarter throttle settings. Turbulence of flow within the manifold disturbed the draft gauge above 2000 engine rpm, so this speed was established as the upper limit of the pressure tests. At the conclusion of the pressure-check runs, a standard full-throttle performance test was made on the engine as equipped with its original intake manifold.

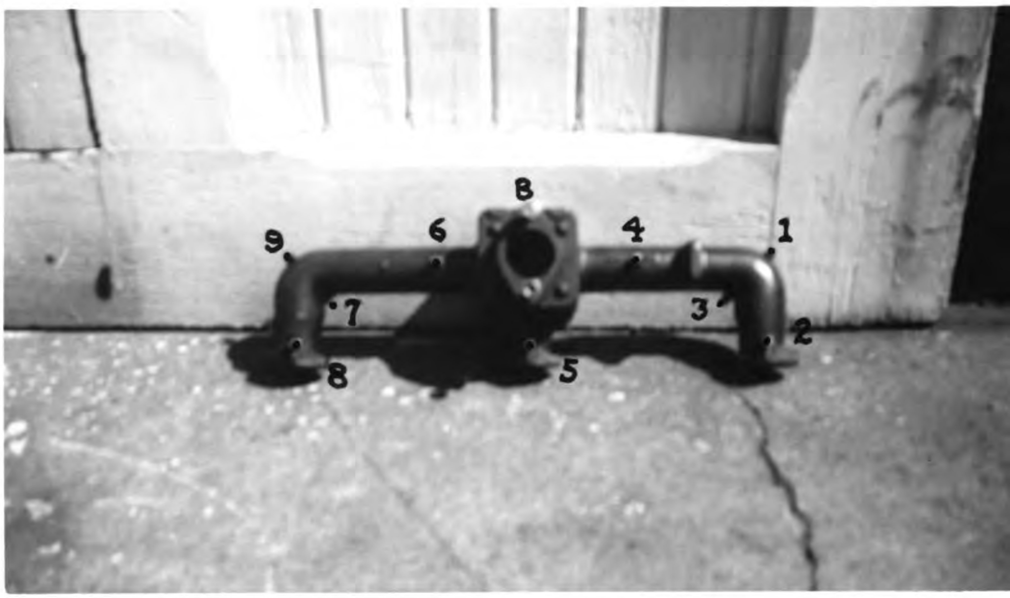


Figure 1. Standard Intake Manifold

Figure 2. Manifold Comparison

There was no particular basis for the design of the Type I intake manifold. It was conceived merely as a design that would, it was believed, most nearly smooth out the pressure variances that were found along the branches of the standard type. A picture of the two manifolds is shown in Figure 2. Figure 3 illustrates the Type I without a heat riser jacket, pressure tapped in positions similar to the original, and in position on the test engine. The new design manifold was constructed of 1.375 inch, 16 gauge seamless steel tubing with welded joints in the first sample, brazed in the final version. Test runs were made on this manifold as before, with one exception. Before the full-throttle performance test was made, a new Type I manifold was constructed, including this time a heat riser. The heat riser was incorporated to improve fuel vaporization, the incompletion of which was evidenced in experimenting with the cold version.

Volumetric efficiency and friction horsepower tests of the engine as equipped with each of the manifolds completed the first phase of the investigation. Comparative road tests were next made, using both test engines mounted in the same standard chassis. These road tests were concerned with acceleration, economy, and overall performance of the automobile. Figure 4 shows the new manifold installed in road operating condition.

Figure 3. Type I Dynamometer Installation

Figure 4. Type I Road Installation

OBSERVATIONS

The data sheets and curves plotted from the test results are all included in the appendix. Figures 5 through 18 are concerned with the initial pressure tests of the standard manifold, as are Tables 2 through 5. Figures 19 through 32 and Tables 6 and 7 are for the pressure tests of the Type I manifold. Tables 8 through 11 and Figures 33 through 36 are combined results of the remainder of the runs.

Figures 5 through 9 are plots of the differential pressures in the standard intake manifold from idle to wide-open throttle settings. The pressures are plotted vs. the take-off point number for each rpm. The points are connected not as an indication of the pressure existing between the take-off ports, but merely to illustrate the wide variances found. It can be seen that the higher the engine rpm at any one throttle setting, the greater are the pressure variances; and as the throttle opening is increased, the pressure differentials rise with a slight increase in variances.

At 1/4 throttle, the pressures vary a maximum of 2.33 inches of water at 1660 rpm, and a minimum of .8 inches at 920 rpm. The 1/2 throttle maximum is 3.43 inches at 2170 rpm, with the minimum of 1.00 inches at 940 rpm; 3/4 throttle maximum 3.43 inches at 1940 rpm, minimum 1.13 inches at 1080 rpm; and the full throttle maximum is 3.7 inches at

2090 rpm, with the minimum .95 inches at 940 rpm.

Figures 10 through 13 are plots of the differential pressures at a point vs. the engine speed at the separate throttle openings. In all cases the differential pressure rose, or the absolute pressure dropped, as the rpm increased, but in no definite or determinable manner. The point differentials varied from .5 to 2.05 inches of water at 1/4 throttle, to from 1.25 to 4.0 inches at full throttle.

The pressure-check runs of the Type I manifold were seriously limited in speed range. Above 1700 rpm the manifold flow was too turbulent to record accurately with the draft gauge. Under 1100 rpm the engine was too rough running to gather accurate results because the manifold was unheated, inhibiting poor fuel vaporization.

At all throttle openings the curves of differential pressures vs. the take-off points were displaced upwards and flattened out by using the smooth curved manifold -- Figures 20 through 23. Maximum and minimum pressure variances at the throttle openings are noted as follows: 1/4 throttle, maximum .9 inches of water at 1620 rpm, minimum .6 inches at 1180; 1/2 throttle, .85 at 1180 rpm, .6 at 1590; 3/4 throttle, 1.0 at 1515 rpm, .6 at 1690; full throttle, 1.5 at 1700 rpm, .8 at 1270.

Figures 24 through 27 illustrate the fact that in the new manifold the individual point differential pressures rose sharply with engine speed and in a more predictable manner than was the case before. The point differentials varied uniformly with speed, averaging about 2.0 inches of water at all throttle settings.

Both manifold types were tested for bhp, torque, and specific fuel consumption at the separate throttle settings. Table 1 is a comparison of these part throttle runs, giving bhp, ft-lb of torque, the fuel consumption in lb/bhp-hr, and the manifold vacuum in "Hg, all at 1600 rpm.

	Standard				Type I			
throttle	1/4	1/2	3/4	full	1/4	1/2	3/4	full
bhp	22	42.5	48	52	26	42	50	52.5
torque	72	139	158	170	86	139	163	172
s.f.c.	.595	.505	.491	.522	.565	.53	.495	.525
vacuum	12.1	2.9	.9	.5	11.3	2.4	.65	.4

Table 1 - Performance Comparisons at 1600 rpm

In spite of the fact that the Type I manifold was operating in the strictly unheated version, it equaled or bettered the standard manifold in practically every phase of performance while operating on a lesser vacuum. The fuel consumption, however, was not as good because of the poor vaporization.

The second manifold that was made up along the pressure-relief line included a heat riser jacket. Upon comparing the full throttle performance records of the new and standard types, it was found that the new type was far superior in all phases, especially at high speeds. The one place where

the Type I was inferior was in the 600-1100 rpm range -- referring to Figure 33. This can be accounted for by the fact that since the riser of the new manifold is set out farther from the block, there is less chance for the exhaust gases to cross the narrow communicating passageway into the jacket at low speeds. This absence of proper heat reduces efficient fuel vaporization which affects horsepower and torque as well as fuel consumption.

The motoring tests of the engine as equipped with the two different manifolds are recorded in Table 9 and plotted in Figures 33 and 34. Relieving the pressure variances showed a marked increase in volumetric efficiency with a resulting slight improvement toward lessening the friction horsepower.

The final testing was done on the road, simulating average driving conditions. During this phase of the investigation the two different engines were used in the same standard chassis. The manifolds were checked for their affect on acceleration from 0 - 60 mph through gears and on the fuel consumption at various speeds. In all cases, without exception, the Type I was superior in both acceleration and delivery of economical operation. The resulting data is recorded in Tables 10 and 11, and the comparisons plotted in Figures 35 and 36. It was also noted that the engine, when equipped with the new manifold, was easier starting than usual and showed excellent overall performance,

regardless of speed or operating conditions.

Unfortunately, through lack of time and proper equipment, no tests were made on the operating air-fuel ratios of the individual cylinders. However, after careful notation of spark-plug and valve deposit coloring, and observing the experimental data, it is hard to believe that there is any serious maldistribution of the fuel mixture. If anything, there was a slight improvement over the standard set-up. This is merely a supposition, however, and not to be misconstrued as fact.

CONCLUSIONS

Any conclusions that can be drawn from the experimental work presented herein must be tempered by the fact that it was all done on one type and make of automotive engine.

It was brought out that there are wide, unpredictable pressure variances existent in this one particular standard intake manifold type. The variances are great along any one branch and differ between branches. They increase with speed and result in a highly turbulent mixture flow above 2000 rpm.

By accommodating a radical, smooth-curved-branch manifold on this engine, several factors were brought out:

1. Magnitude of the pressure variances decreased.
2. Differential pressures increased with engine speed but in a more predictable manner.
3. Absolute pressures were lower -- curves displaced upwards.
4. Turbulence of flow was evidenced at a lower speed -- 1700 rpm.
5. Part throttle performance improvements in all speed ranges.
6. Marked brake horsepower and torque improvements at full throttle, especially in the high speed range.
7. A gain was made in specific fuel consumption at full throttle.
8. Improved volumetric efficiency with a slight decrease

of friction-horsepower.

9. Astonishing improvement in road economy and performance.

10. Apparent good fuel mixture distribution to all cylinders.

It appears that a balanced riser, as used on the Type I manifold, is an aid to proper distribution. The two out-board branches are extremely long as compared to the center one but the equiangular take-offs from the riser seem to counteract the ill-effects of the different lengths. It is possible that distribution could be further improved by using flat-bottomed channels -- increasing the surface area for fuel vaporization -- instead of the round ones that were used.

This preliminary investigation presented the realization that there lies in the field of pressure analysis a possible insight to improved intake manifold design for the automotive engine. The results set forth here are not conclusive because of their short range limitations and application to one engine. However, the way has been pointed out for further investigations over a wider speed range with greater variables, and with several different engine types. It is entirely feasible, as has been stated before, that pressure analysis could be the design key.

APPENDIX

EXPERIMENTAL DATA

MECHANICAL ENGINEERING LABORATORY
MICHIGAN STATE COLLEGE

Running Log of Test of Standard Intake Manifold

Observers: Olsen, S.A.

Room Temp. 75°F

Barometer 29.4 "Hg. Date 12-29, 1950

No.	RPM	Beam load lb.	Differential pressure from pt.B "H_2O 1.	2.	3.	4.	5.	6.	7.	8.	9.	pt. B "Hg.
1	Idle											
2	370	0	2.72	2.4	2.8	2.77	2.62	2.7	2.6	2.65	2.78	19.6
3												
4	1/4 throttle											
5	1660	38	.90	1.95	3.18	1.85	2.15	1.90	3.23	2.4	1.30	12.6
6	1400	46	.58	1.60	2.13	1.0	1.88	1.18	2.05	1.67	.83	10.2
7	1220	52.5	.52	1.27	1.60	.82	1.25	.85	1.60	1.33	.65	8.8
8	1050	57	.50	1.02	1.33	.65	1.0	.67	1.3	1.1	.50	7.4
9	920	60	.40	.87	1.20	.60	.82	.63	1.18	1.0	.43	6.2
10												
11	1/2 throttle											
12	2170	66	1.72	4.2	5.0	3.0	3.0	2.6	4.95	3.37	1.57	4.7
13	1840	73	1.3	3.5	4.45	2.45	2.3	1.85	4.1	3.25	1.2	3.6
14	1650	76	1.08	3.05	4.0	2.05	2.05	1.18	3.65	2.9	1.1	2.9
15	1400	78	.53	1.85	2.5	1.13	1.52	.98	2.33	1.78	.48	2.1
16	1130	80.5	.43	1.37	1.3	.81	1.26	.68	1.72	1.30	.45	1.7
17	940	79.5	.27	.85	1.25	.50	.72	.43	1.1	.95	.25	1.1
18												
19												
20												
21												
22												
23												
24												
25												

Remarks:

MECHANICAL ENGINEERING LABORATORY
MICHIGAN STATE COLLEGE

Running Log of Test of Standard Intake Manifold

Observers ______ ______ **Date** ______, 19 ___

No.	RPM	Beam Load lb.	Differential pressure from pt. B "H_2O 1.	2.	3.	4.	5.	6.	7.	8.	9.	pt. B "Hg.
1	3/4 throttle											
2	1940	86	1.1	3.67	4.45	2.23	2.75	1.35	4.22	3.15	1.02	1.2
3	1700	87.5	.92	3.03	3.93	1.73	2.1	1.55	3.55	2.73	.38	1.0
4	1410	87	.28	1.75	2.25	1.02	1.47	.33	2.15	1.55	.25	.7
5	1190	86	.23	1.27	1.72	.73	1.23	.55	1.62	1.2	.25	.5
6	1080	86	.17	.95	1.28	.49	.93	.42	1.03	.73	.15	.4
7												
8	full throttle											
9	2090	93	1.3	4.4	5.0	2.3	3.93	2.23	4.95	3.7	1.4	.9
10	1300	94	.99	3.48	4.55	2.31	2.85	1.38	4.1	3.15	1.12	.7
11	1620	95	.45	2.75	3.75	1.8	2.45	1.3	3.5	1.93	.35	.6
12	1370	93.5	.35	1.7	2.33	1.13	1.78	.93	2.27	1.5	.30	.4
13	1140	93	.28	1.31	1.82	.78	1.6	.63	1.62	1.23	.28	.3
14	940	92	.05	.63	1.0	.44	.99	.33	.87	.59	.06	.1
15												
16												
17												
18												
19												
20												
21												
22												
23												
24												
25												

Remarks: Above 2000 rpm the flow was to turbulent to record accurately.

[illegible]

Figure 5

Differential Pressures at Take-off Points

Idle

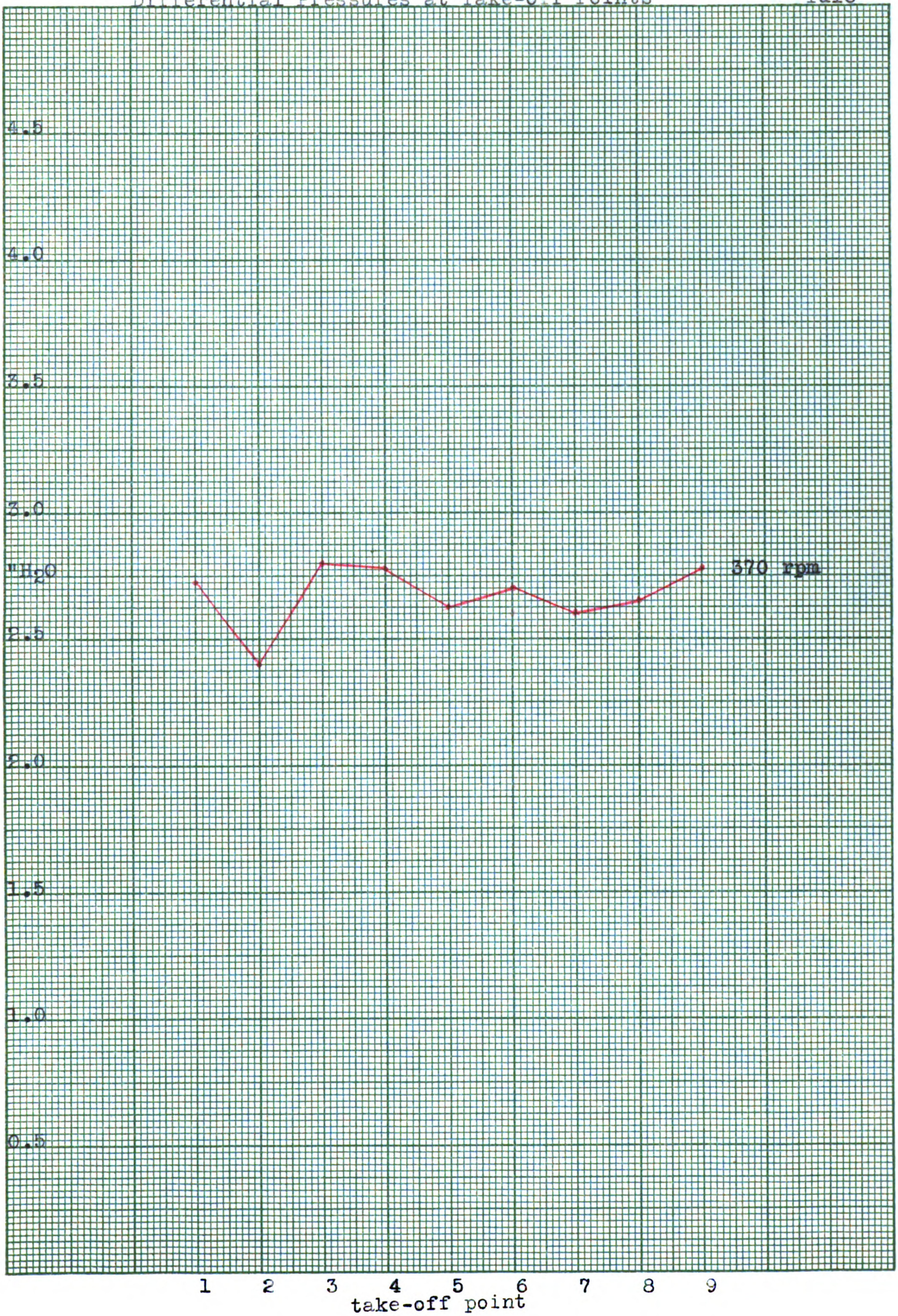


Figure 8

Differential Pressures at Take-off Points 1/4 throttle

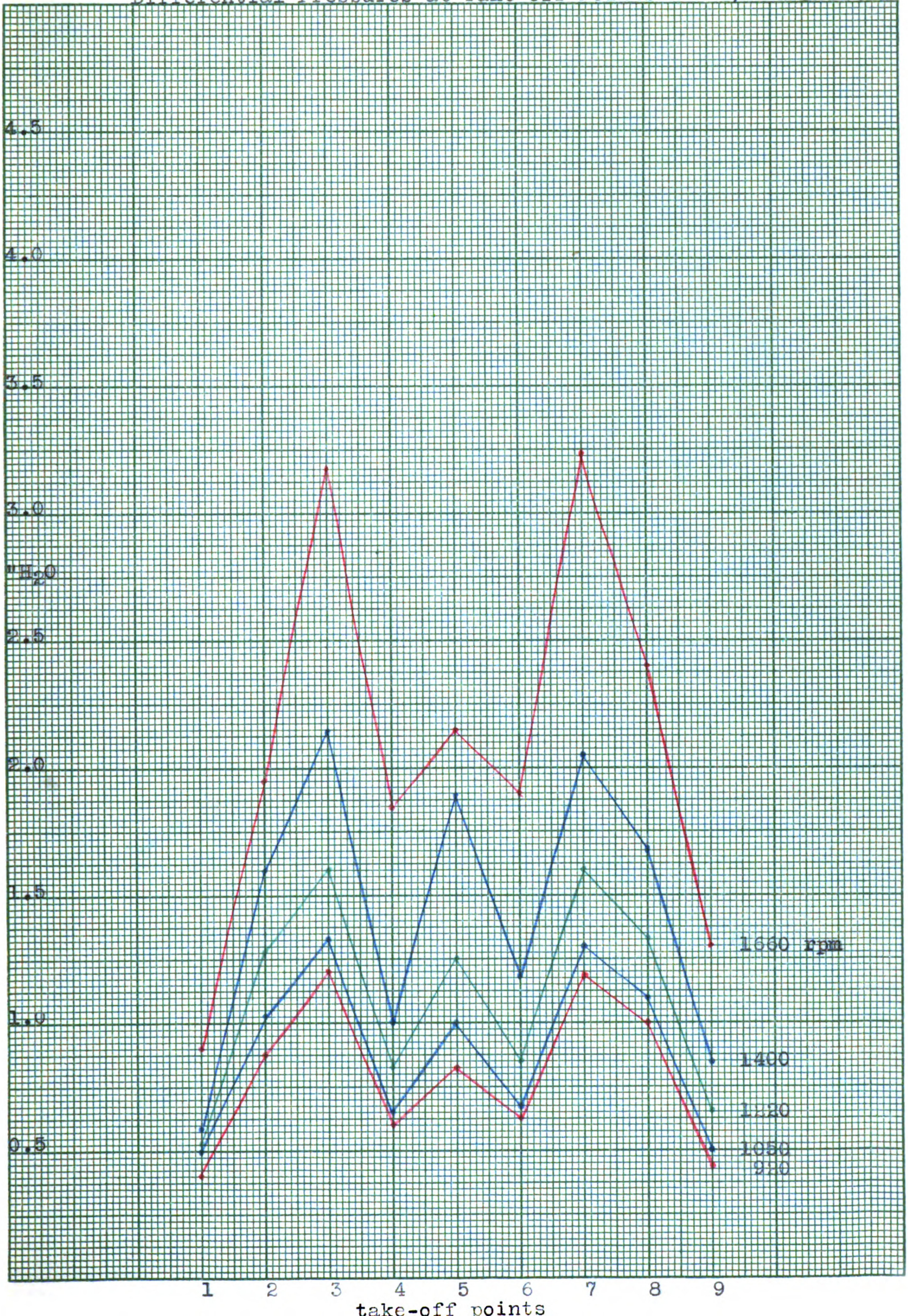


Figure 7

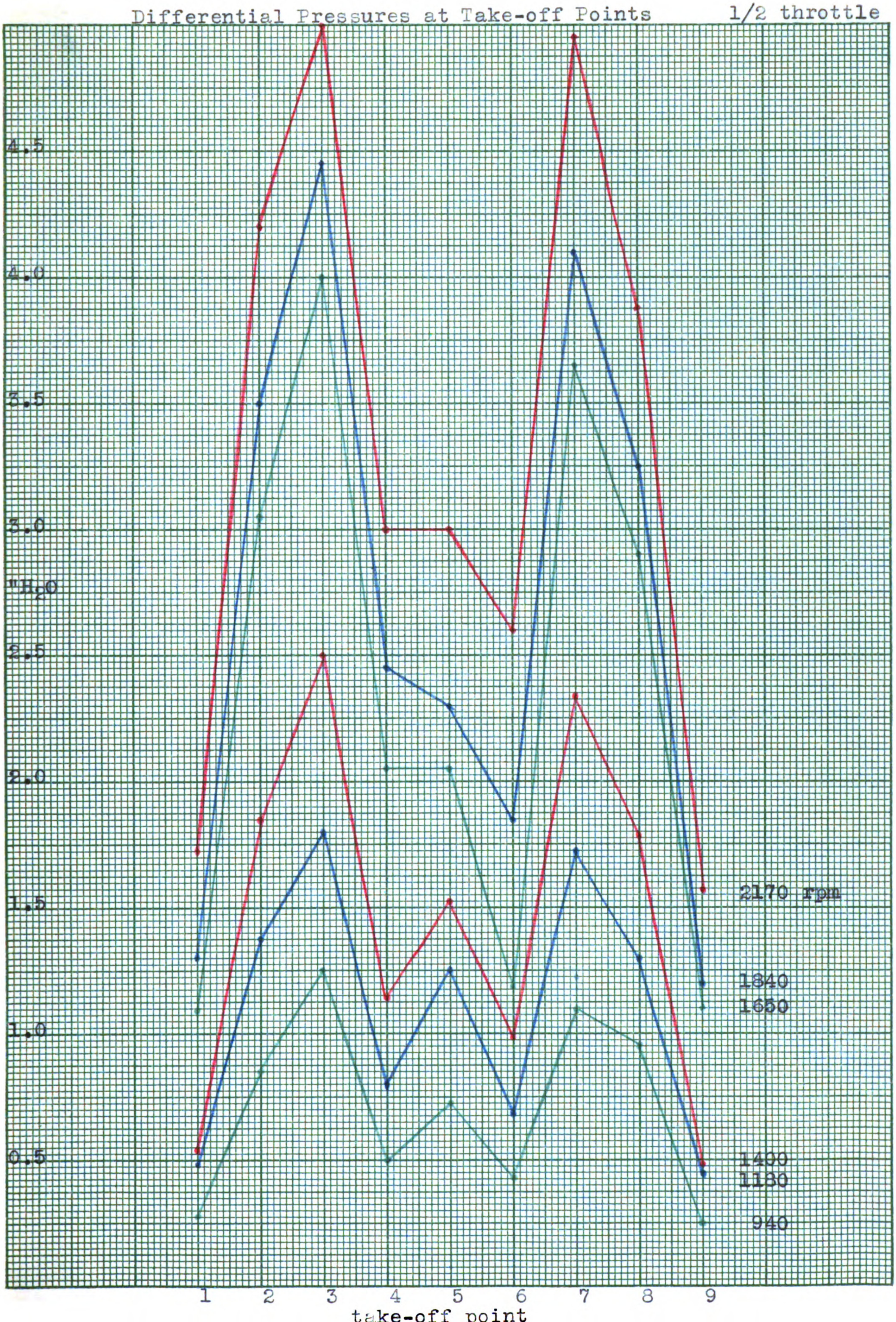

Differential Pressures at Take-off Points
1/2 throttle
4.5
4.0
3.5
3.0
"H_2O
2.5
2.0
1.5
1.0
0.5
2170 rpm
1840
1650
1400
1180
940
1
2
3
4
5
6
7
8
9
take-off point

Figure 8

Differential Pressures at Take-off Points 3/4 throttle

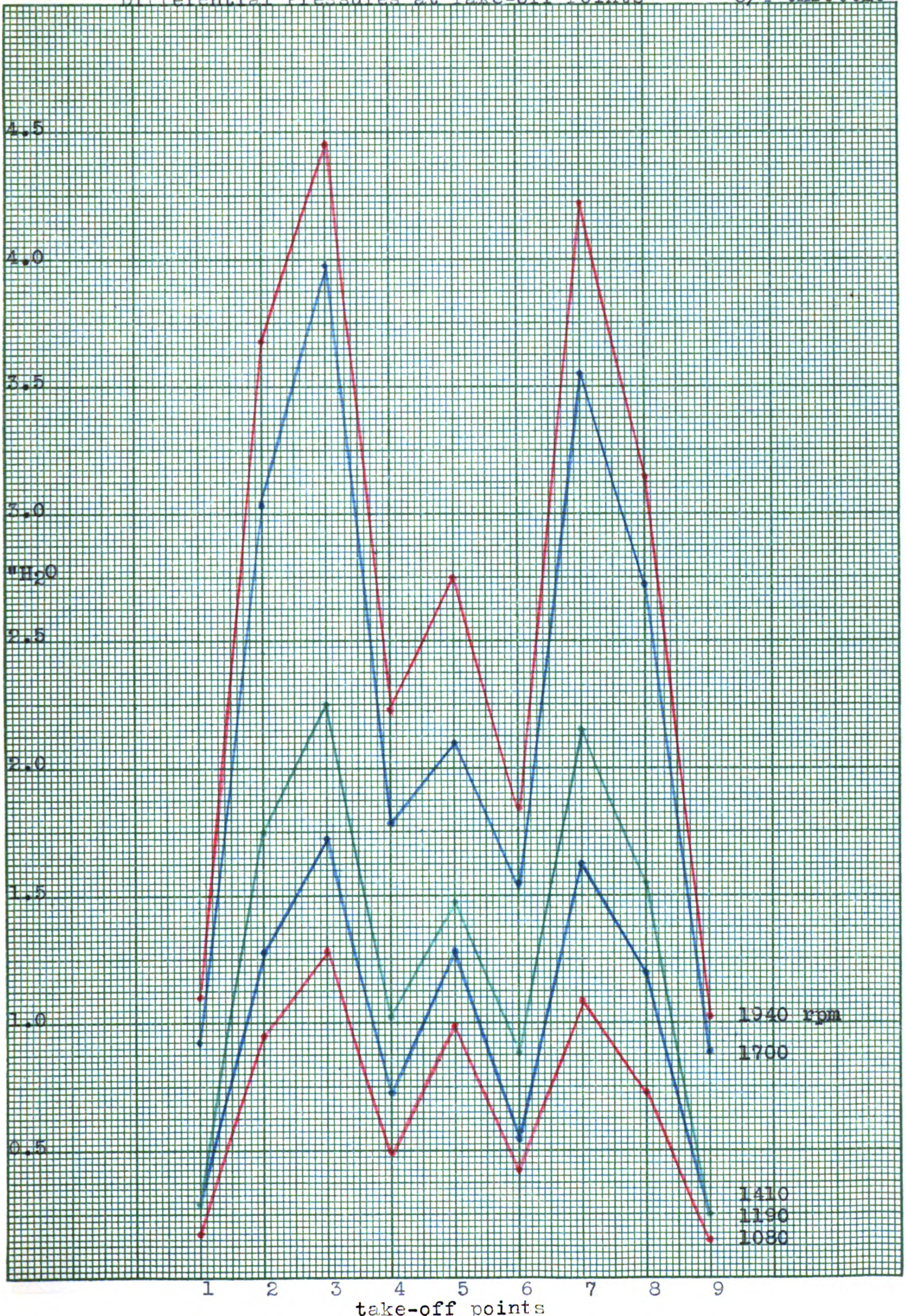


Figure 9

Differential Pressures at Take-off Points, full throttle

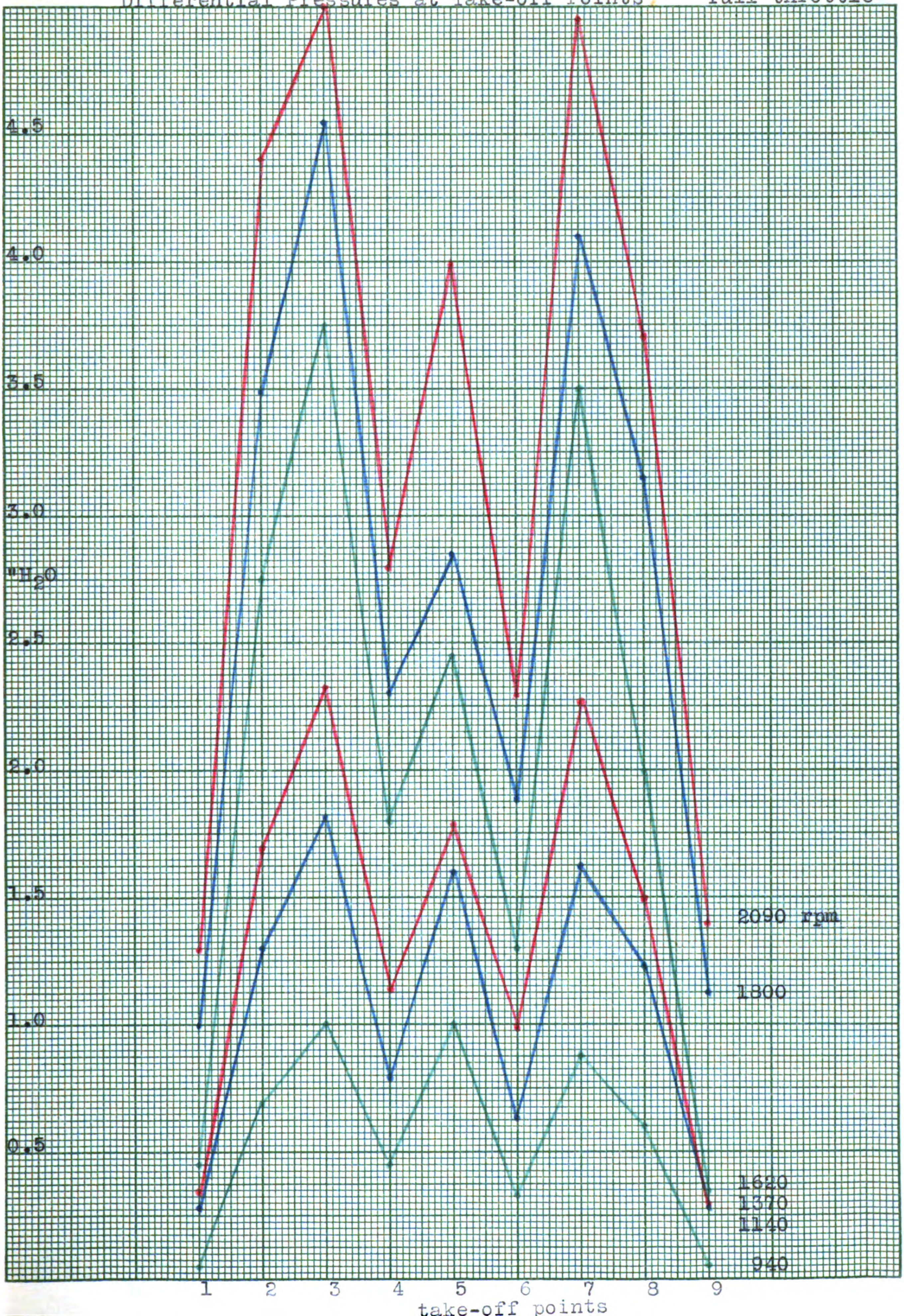


Figure 10

Differential Pressures vs Engine Speed 1/4 throttle

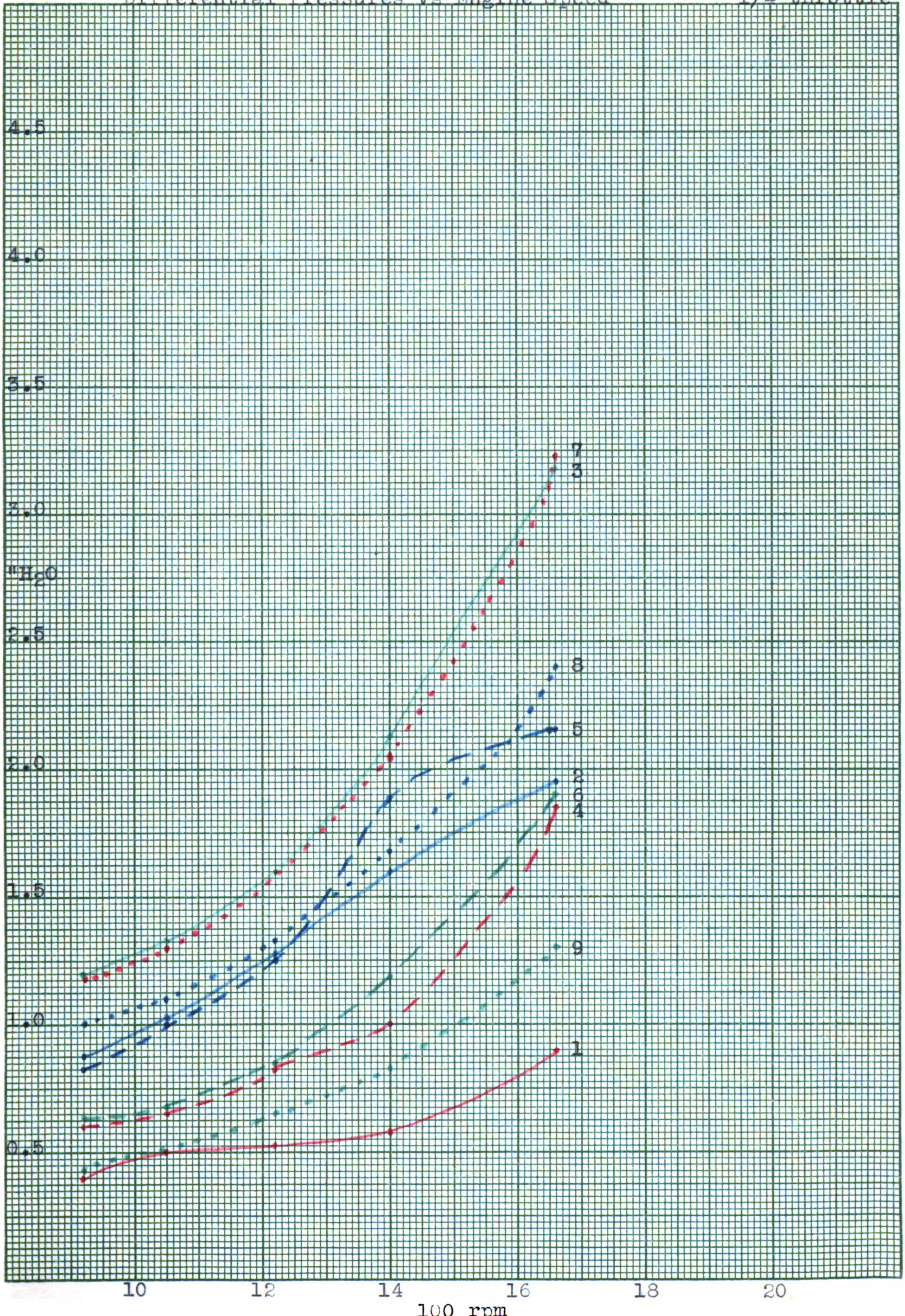
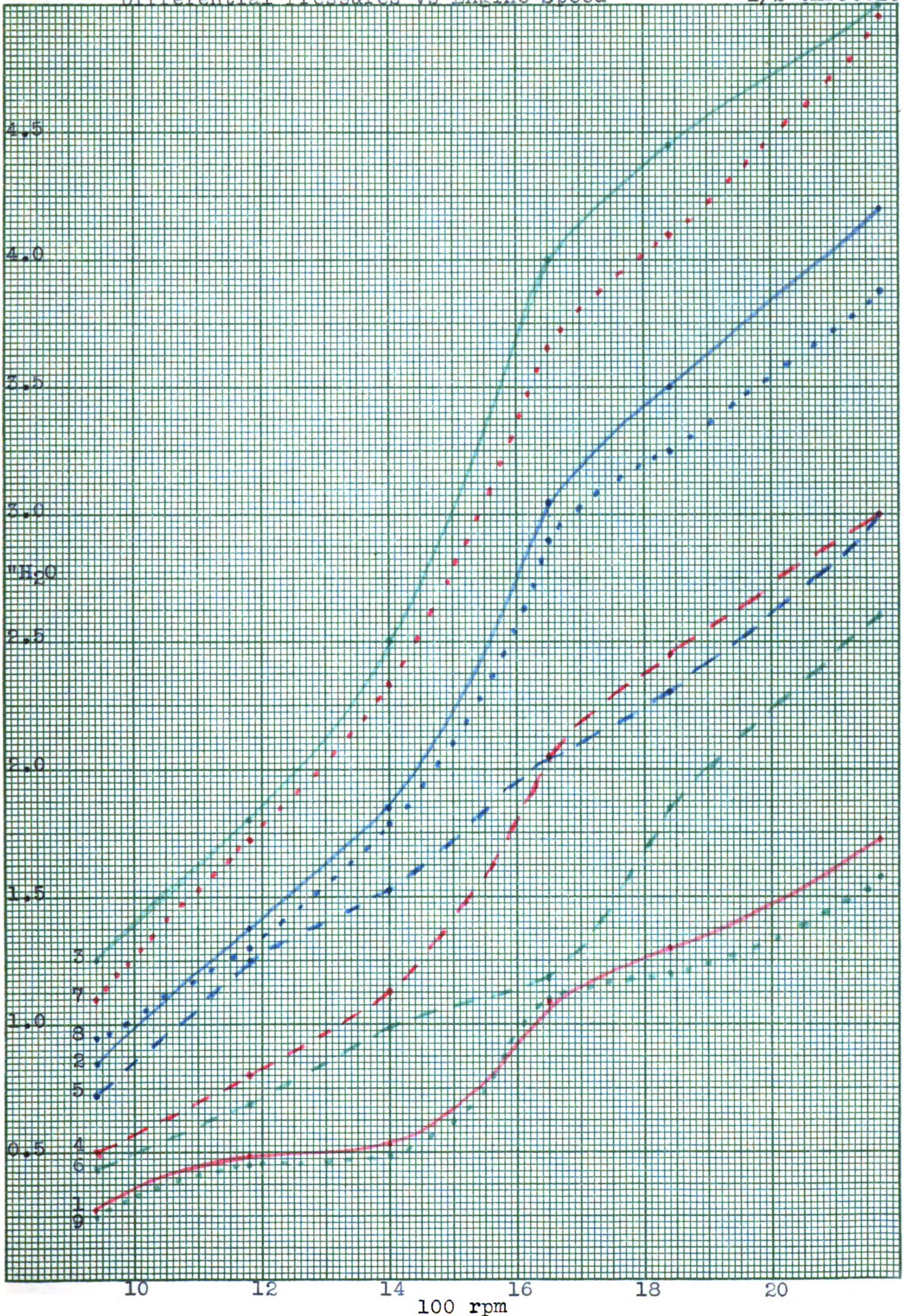


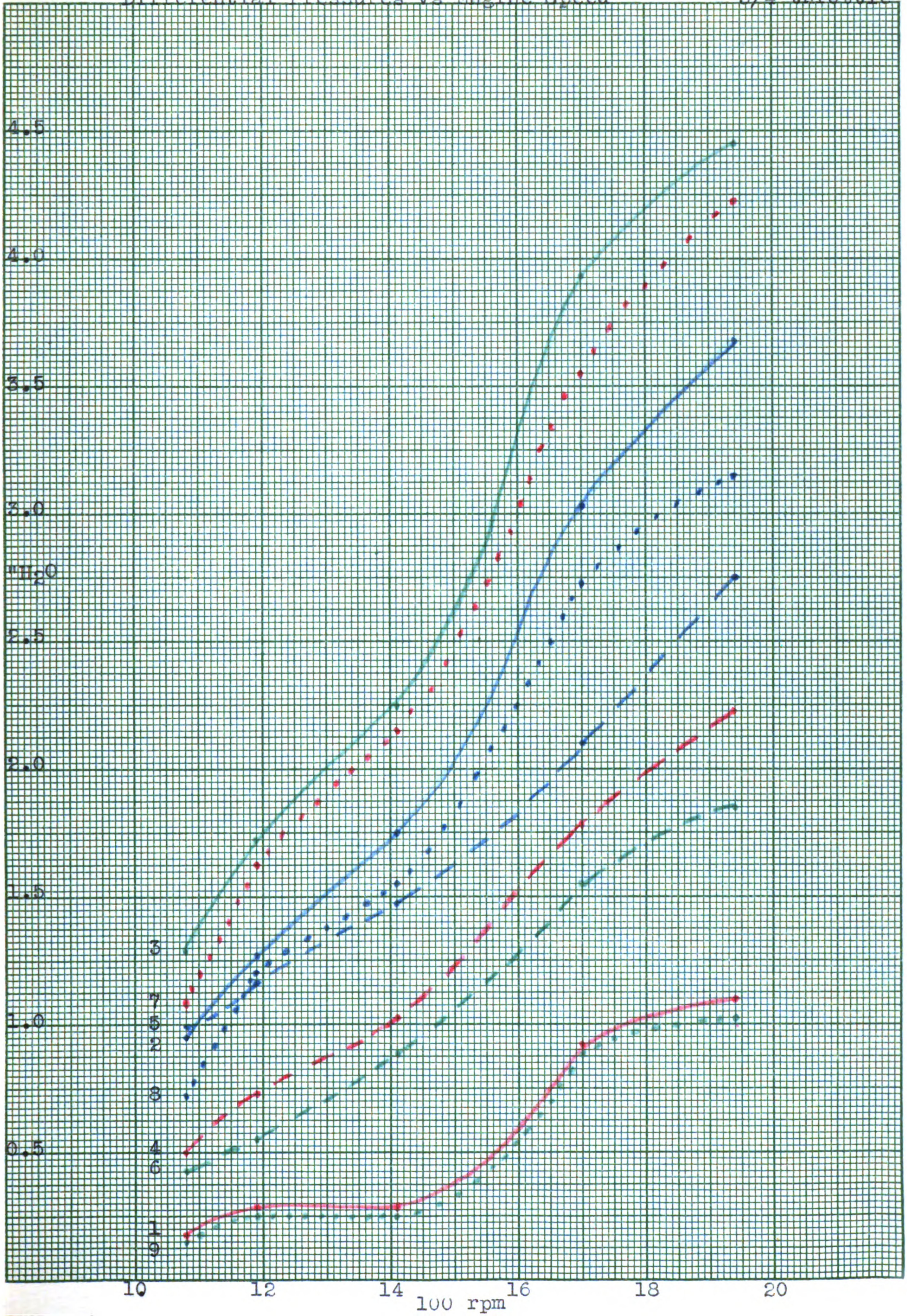
Figure 11

Differential Pressures vs Engine Speed

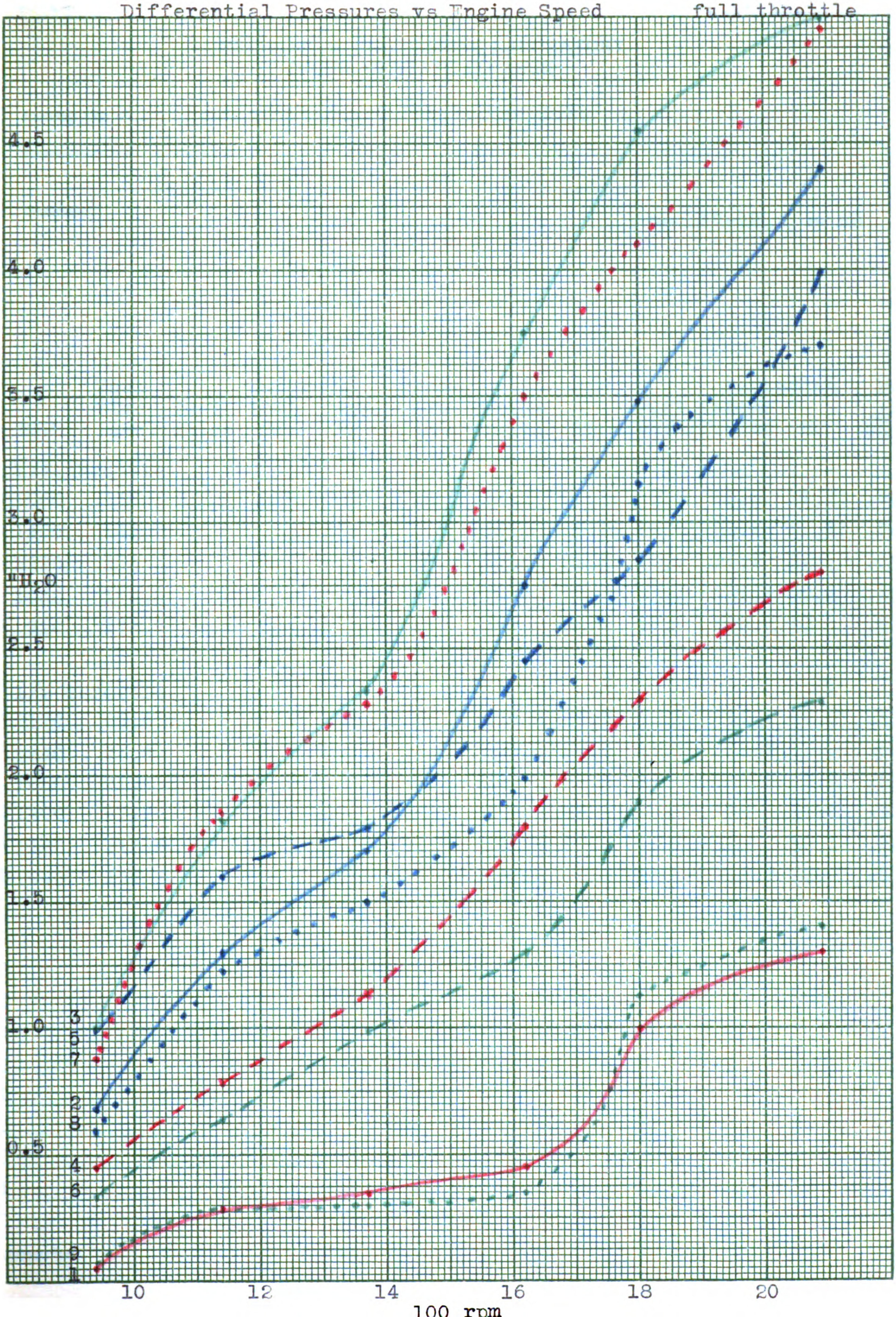
1/2 throttle



Differential Pressures vs Engine Speed   3/4 throttle



Differential Pressures vs Engine Speed full throttle



MECHANICAL ENGINEERING LABORATORY
MICHIGAN STATE COLLEGE

Running Log of Test of Standard Intake Manifold

Observers Olsen, S.A.

Date 12-29, 19 50

No.	RPM	Fuel time sec.	Fuel fl.oz	Water temp °F	Oil temp °F	Brake Arm in.	Torque ft-lbs	HP	Fuel lbs/hr	Corrected Torque ft-lbs	Corrected HP	Fuel lbs BHP-hr
1	1/4 throttle											
2	1660	138	10.5	165	180	21	66.5	21.05	13.2	68.6	21.75	.607
3	1400	145					80.5	21.45	12.55	83.1	22.15	.567
4	1220	153					91.8	21.33	11.9	94.7	22	.541
5	1050	164.5					99.6	19.96	11.1	102.8	20.6	.539
6	920	173.5					105	18.4	10.5	108.4	19	.552
7												
8	1/2 throttle											
9	2170	69.6					115.5	47.7	26.2	119.2	49.3	.532
10	1840	76.8					127.8	44.8	23.75	131.9	46.3	.514
11	1650	82.8					133	41.8	22	137.2	43.2	.509
12	1400	97.7					136.4	36.4	18.65	141	37.6	.496
13	1180	113.5					140.3	31.6	16.03	145.2	32.6	.493
14	940	136					139.1	24.9	13.38	143.8	25.7	.521
15												
16												
17												
18												
19												
20												
21												
22												
23												
24												
25												

Remarks:

MECHANICAL ENGINEERING LABORATORY
MICHIGAN STATE COLLEGE

Running Log of Test of Standard Intake Manifold

Observers

Date , 19

No.	RPM	Fuel time sec.	Fuel fl.oz	Water temp °F	Oil temp °F	Brake arm in.	Torque ft-lbs	HP	Fuel lbs/hr	Corrected Torque ft-lbs	Corrected HP	Fuel lbs / BHP-hr
1	3/4 throttle											
2	1940	65.4	10.5	165	180	21	150.5	55.6	27.85	155.3	57.4	.486
3	1700	72.6					153	49.6	25.1	158	51.2	.49
4	1410	87					152.2	40.9	20.95	157	42.2	.497
5	1190	103					150.5	34.2	17.65	155.3	35.3	.500
6	1080	113.5					150.5	31	16.08	155.3	32	.503
7												
8	full throttle											
9	2090	54					162.8	64.8	33.8	168	66.9	.506
10	1800	60					164.5	56.5	30.4	169.8	58.3	.522
11	1620	67					166.2	51.4	27.6	171.5	53	.522
12	1370	79.2					163.7	42.7	23	169	44.1	.522
13	1140	93					162.8	35.35	19.6	168	37.45	.523
14	940	110					161	28.8	16.6	166.2	29.7	.561
15												
16												
17												
18												
19												
20												
21												
22												
23												
24												
25												

Remarks:

Figure 14

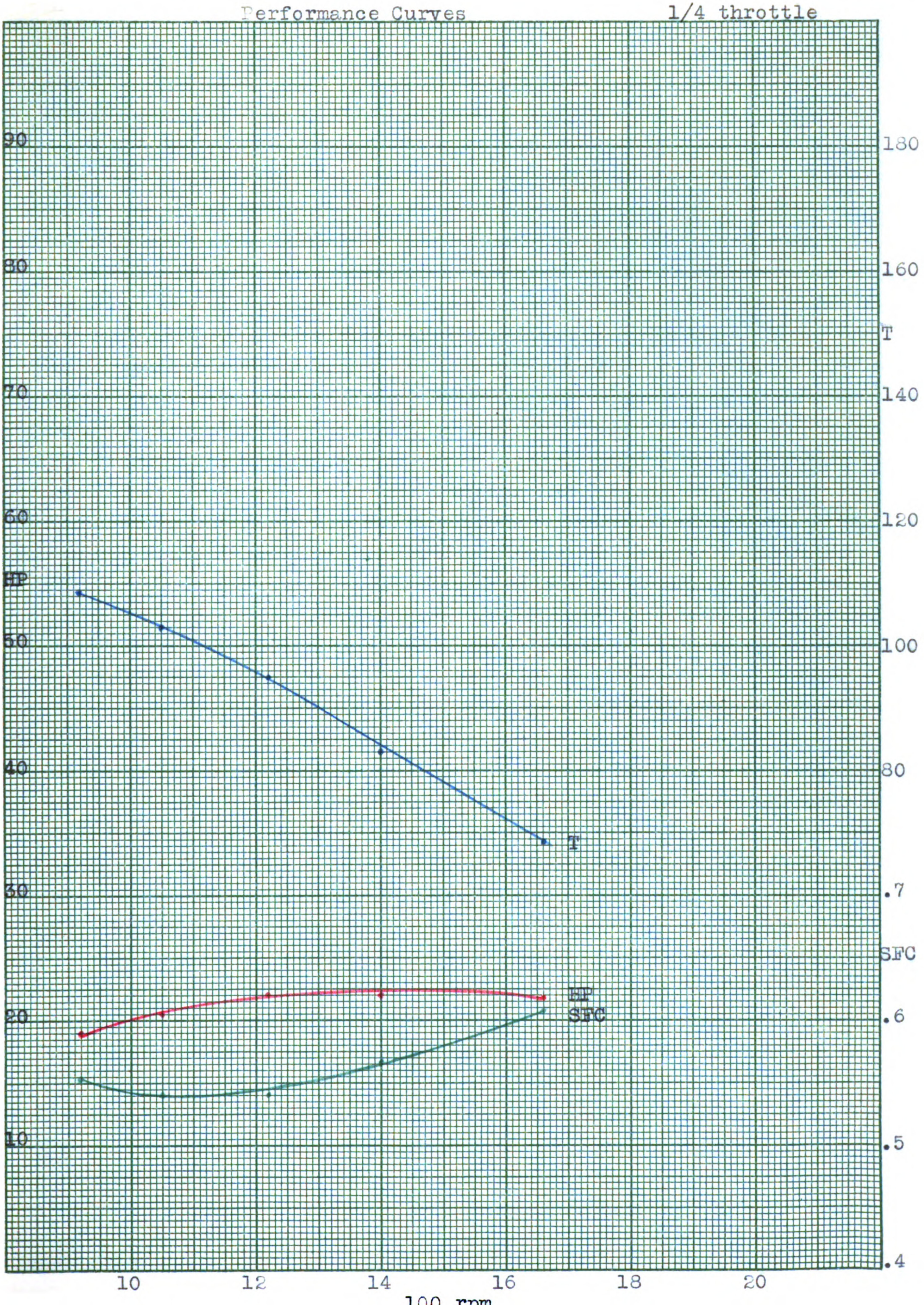

Performance Curves
1/4 throttle
90
80
70
60
HP
50
40
30
20
10
180
160
T
140
120
100
80
.7
SFC
.6
.5
.4
T
HP
SFC
10
12
14
16
18
20
100 rpm

Figure 15

Performance Curves 1/2 throttle

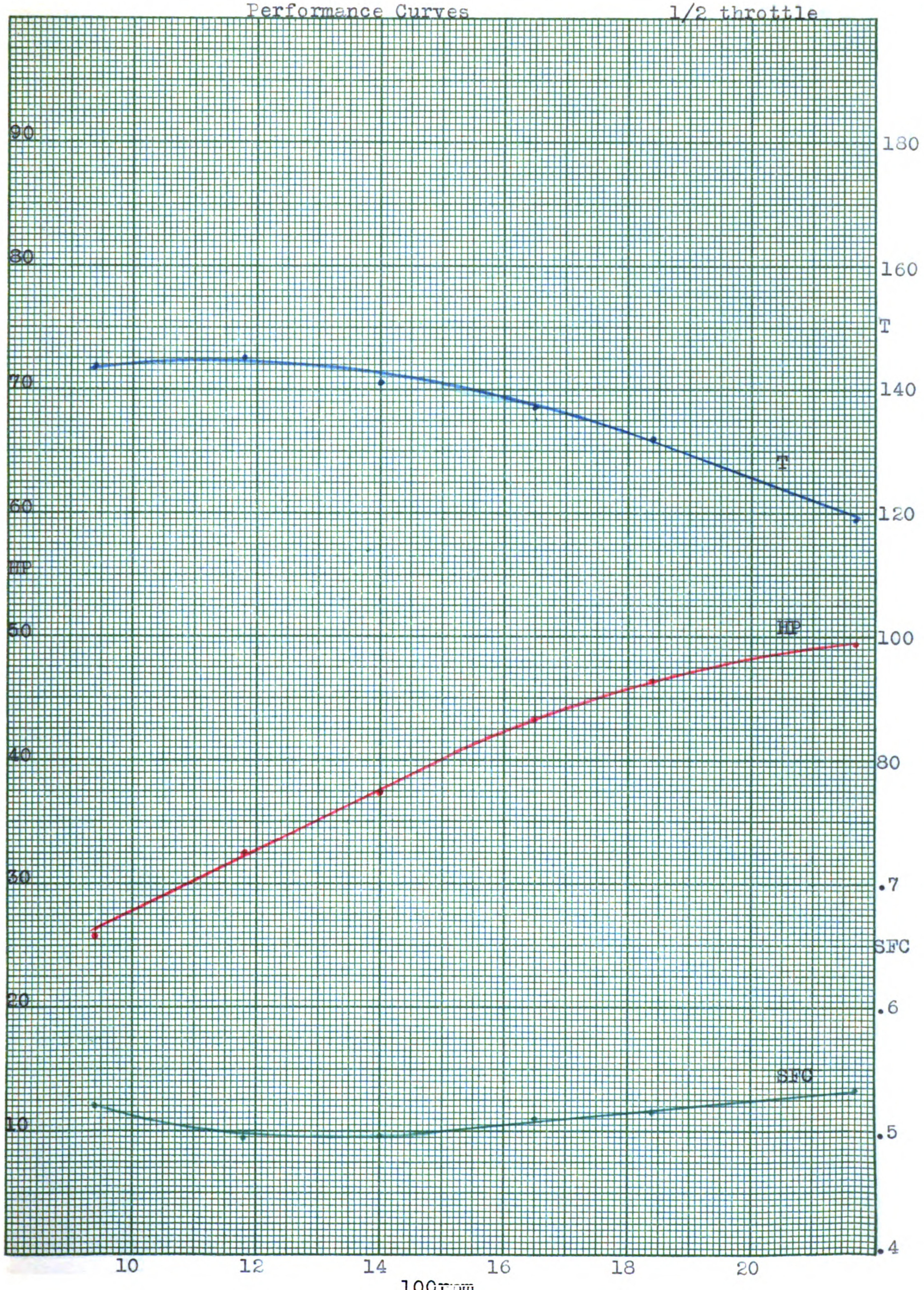


Figure 16

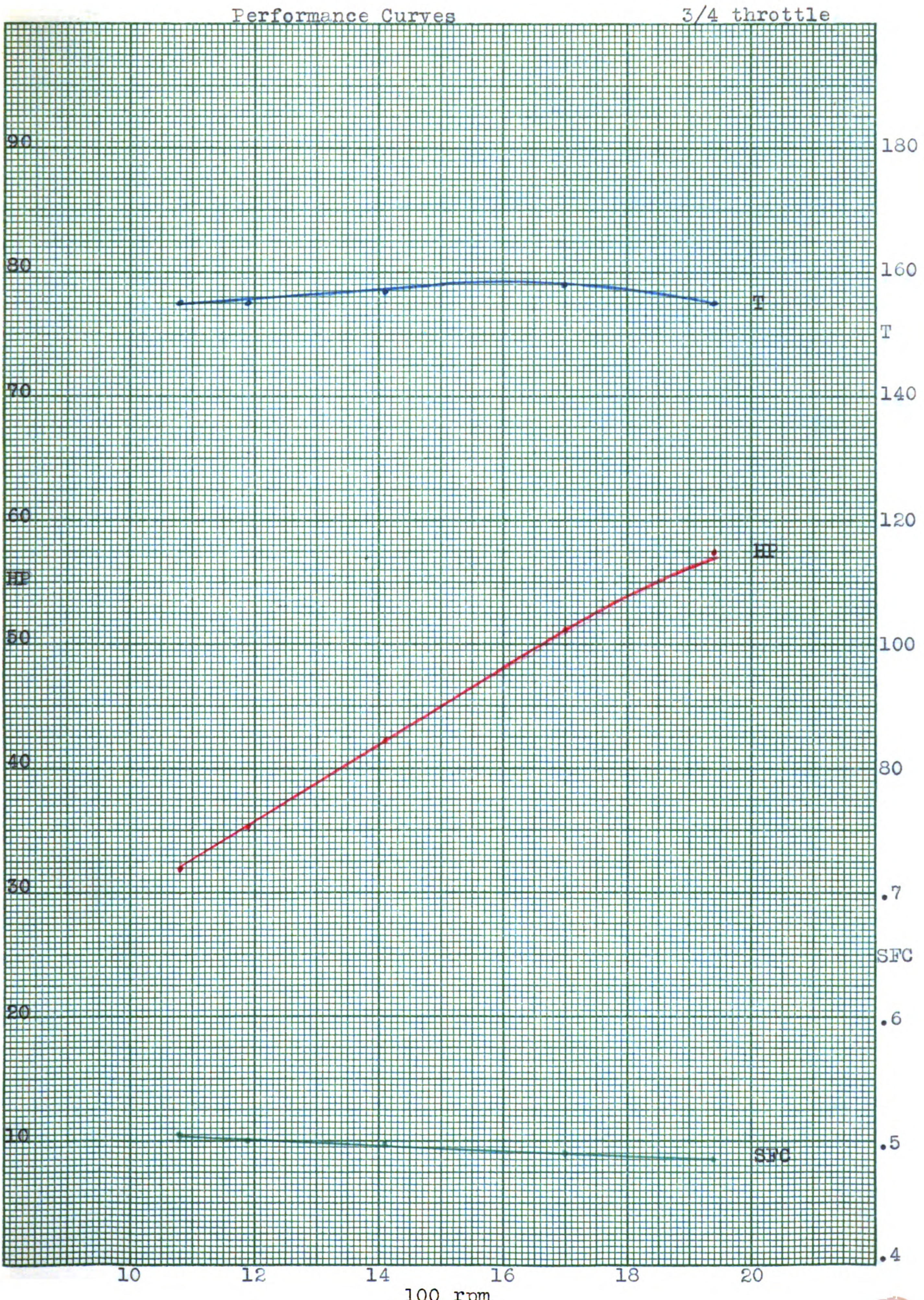
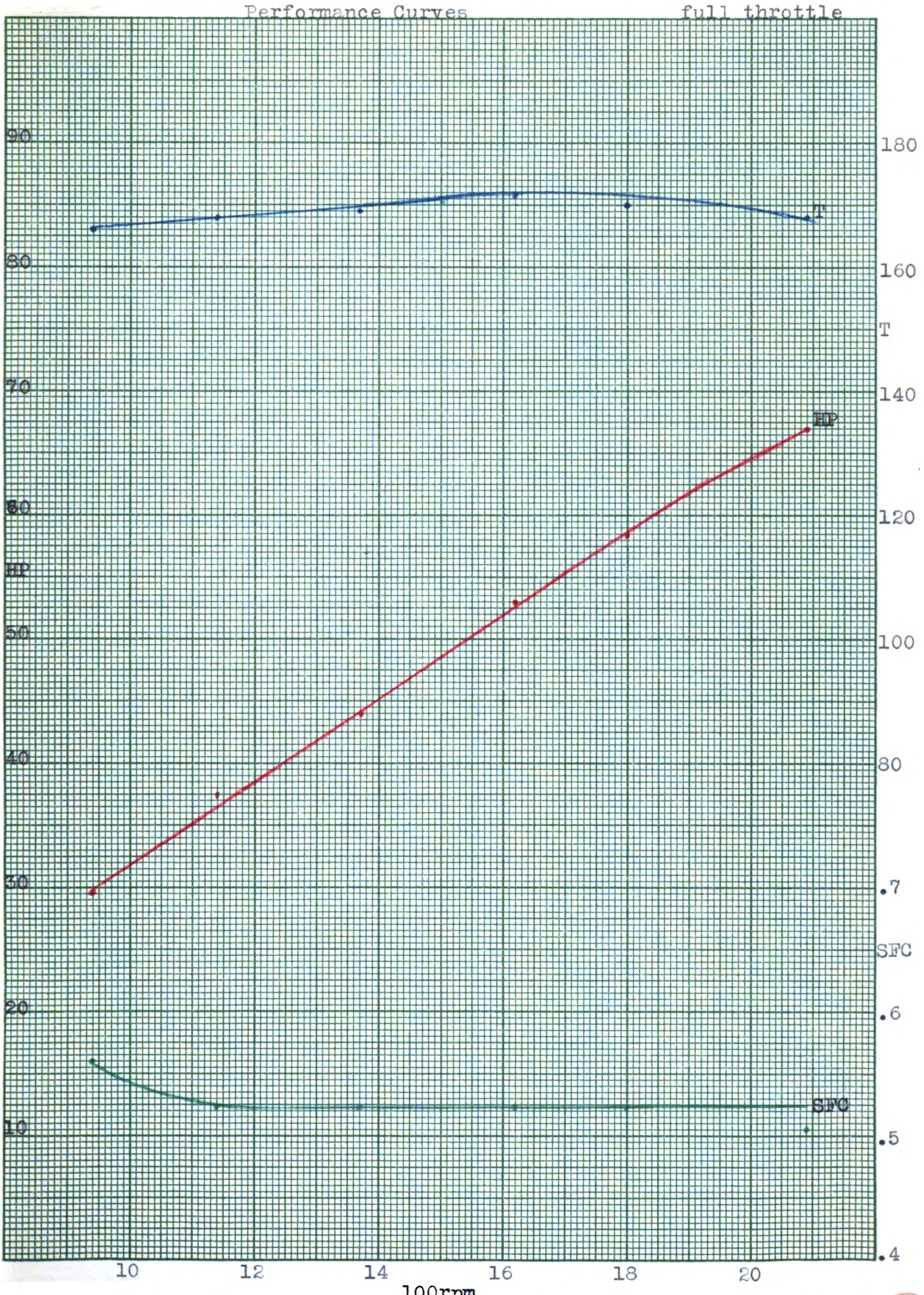
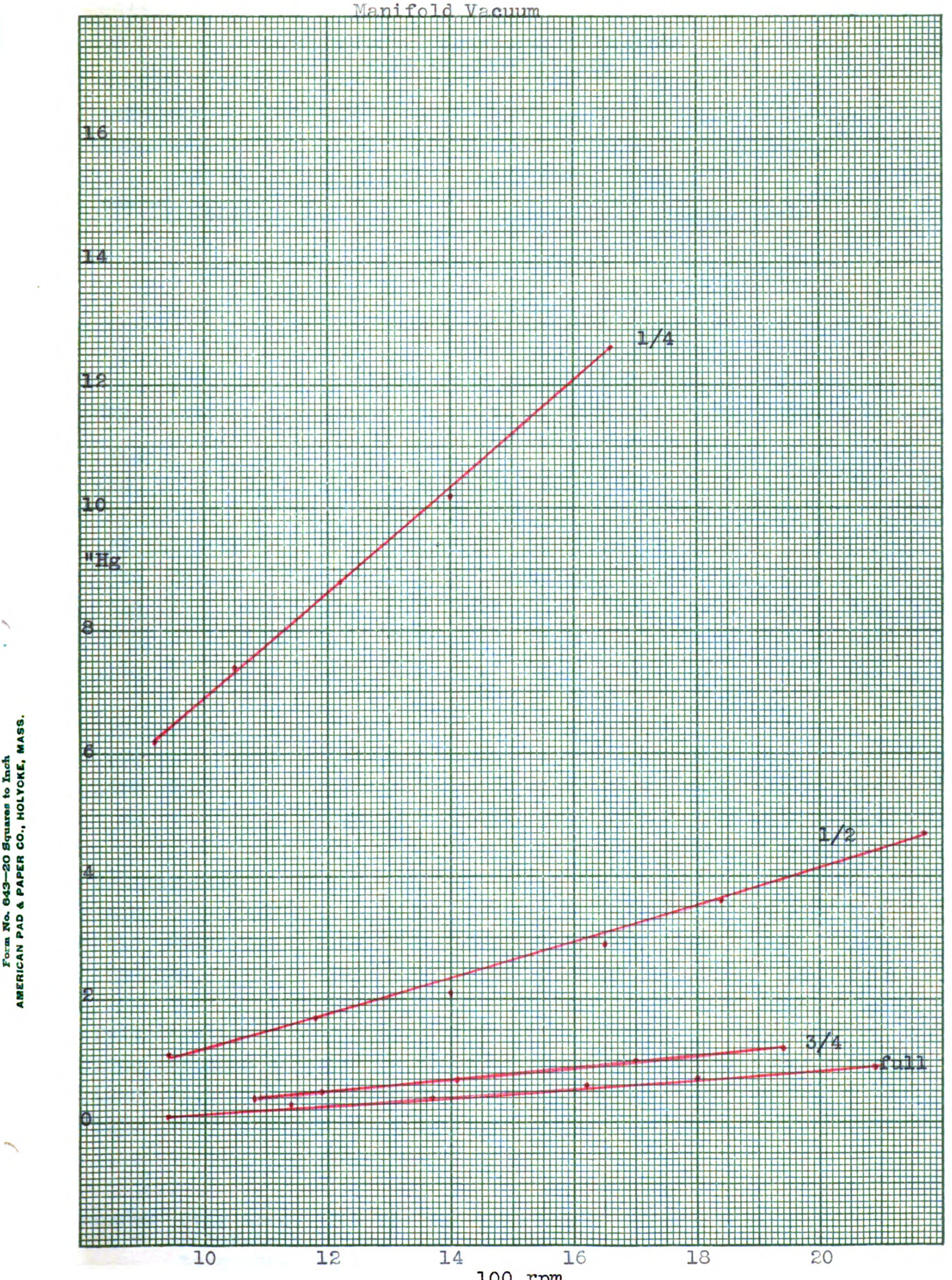

Performance Curves
3/4 throttle
90
80
70
60
HP
50
40
30
20
10
180
160
T
140
120
100
80
.7
SFC
.6
.5
.4
T
HP
SFC
10
12
14
16
18
20
100 rpm

Figure 17



Performance Curves
full throttle
90
80
70
60
HP
50
40
30
20
10
180
160
T
140
120
100
80
.7
SFC
.6
.5
.4
T
HP
SFC
10
12
14
16
18
20
100rpm

Figure 18


Manifold Vacuum
16
14
12
10
"Hg
8
6
4
2
0
1/4
1/2
3/4
full
10
12
14
16
18
20
100 rpm
Form No. 643—20 Squares to Inch
AMERICAN PAD & PAPER CO., HOLYOKE, MASS.

MECHANICAL ENGINEERING LABORATORY
MICHIGAN STATE COLLEGE

Running Log of Intake Manifold #1 Test

Observers: Olsen, S.A.; Johnson, B.A.

Room temp. 74°F

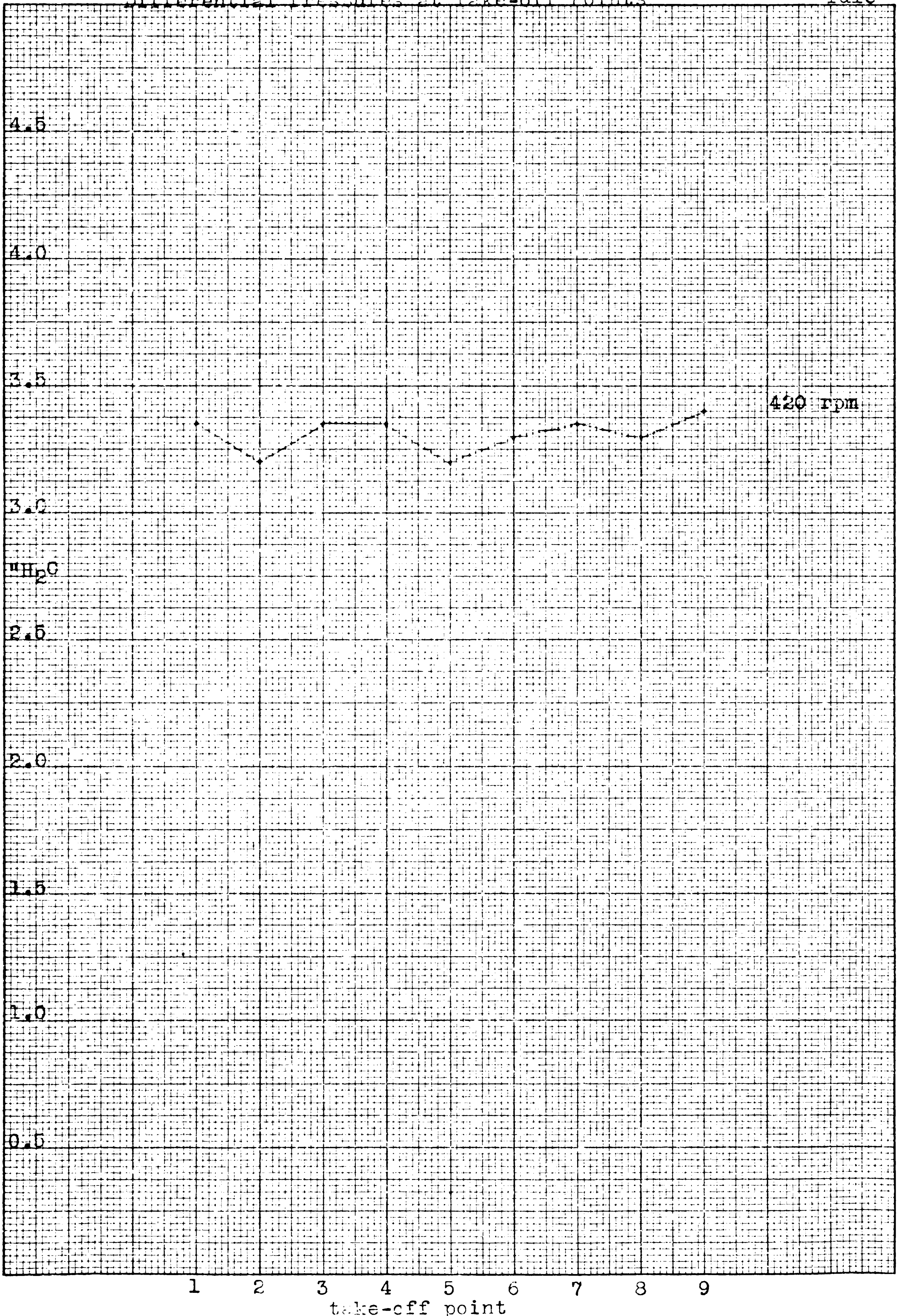
Barometer 29.50"Hg. Date 2-10, 1951

No.	RPM	Beam load lb.	Differential pressure from pt. B "H_2O 1.	2.	3.	4.	5.	6.	7.	8.	9.	pt. B "Hg.
1	Idle											
2	420	0	3.35	3.2	3.35	3.35	3.2	3.3	3.35	3.3	3.4	20
3												
4	1/4 throttle											
5	1835	39.5	- -	- -	- -	turbulent		- -	- -	- -	- -	13.2
6	1620	47	4.5	4.4	5.0	4.1	5.0	4.1	4.9	4.6	4.5	11.6
7	1400	53.5	3.45	3.4	4.0	3.3	3.7	3.1	3.5	3.6	3.35	9.6
8	1180	57	2.6	2.5	2.8	2.3	2.9	2.3	2.7	2.7	2.4	7.6
9												
10	1/2 throttle											
11	1590	77	4.6	4.4	5.0	4.5	4.9	4.6	5.0	4.5	4.7	2.4
12	1430	75.5	3.7	3.9	4.3	3.6	3.9	3.4	3.85	3.6	3.4	1.8
13	1180	72.5	2.5	2.65	3.05	2.35	2.6	2.2	2.75	2.8	2.55	1.2
14												
15	3/4 throttle											
16	1690	91	4.0	4.2	4.5	3.9	3.9	4.0	4.2	4.1	3.9	.8
17	1515	90.5	3.2	3.5	4.0	3.3	3.15	3.0	3.6	3.4	3.0	.6
18	1300	89.5	2.2	2.3	2.9	2.35	2.25	2.15	2.6	2.45	2.05	.5
19												
20	full throttle											
21	1700	95.5	3.8	4.2	4.6	3.4	3.35	3.2	4.7	4.4	3.7	.6
22	1490	94.5	2.6	2.85	3.5	2.75	2.5	2.8	3.4	3.15	2.55	.4
23	1270	91	1.7	1.8	2.35	1.9	1.95	1.85	2.15	2.1	1.55	.3
24												
25												

Remarks: Over about 1700 rpm the flow was to turbulent to record accurately. Under 1100 rpm the engine was to rough running to gather accurate results

Figure 19

Differential Pressures at Take-off Points



Differential Pressures at Take-off Points 1/4 throttle

Figure 21

Differential Pressures at Take-off Points 1/2 throttle

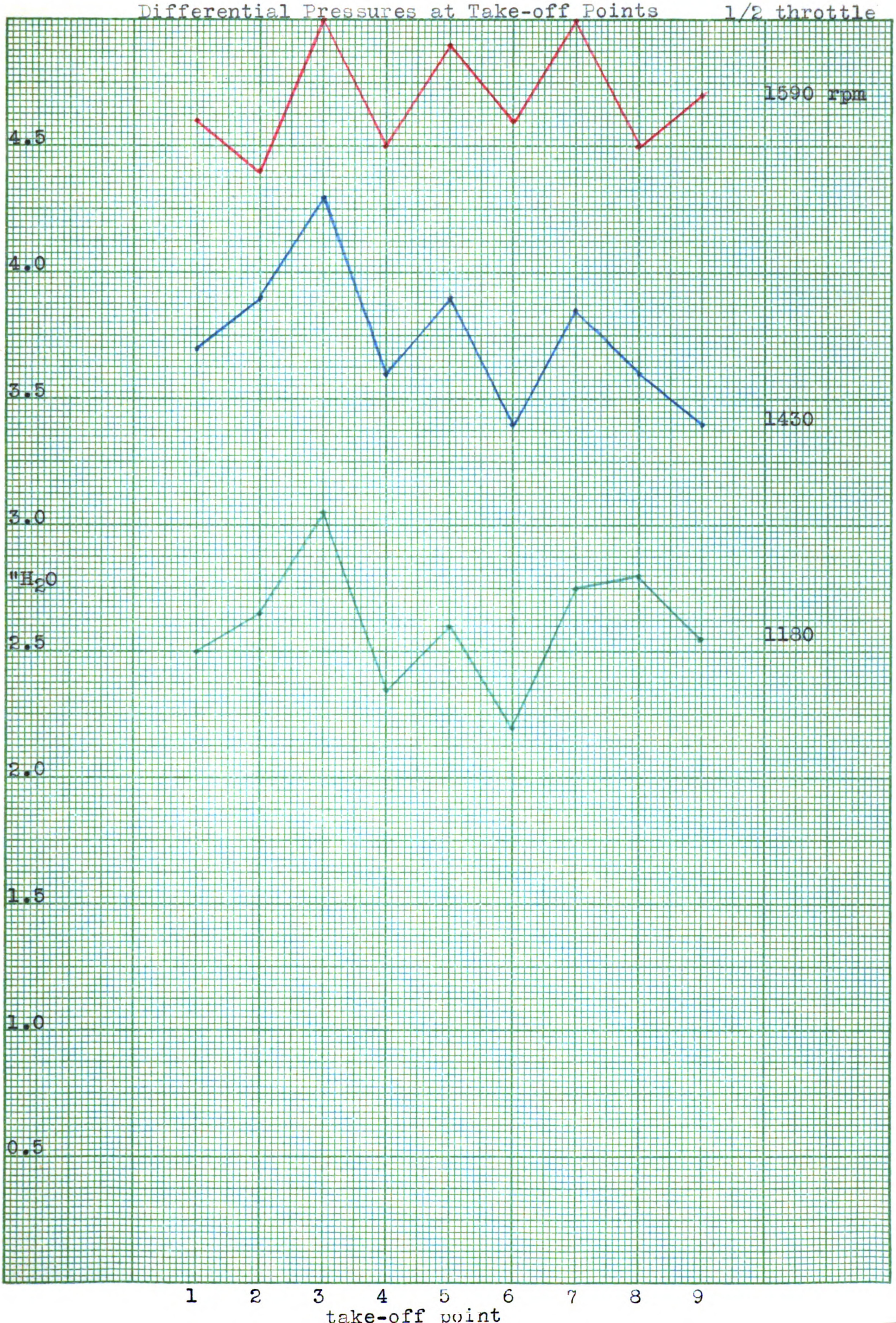


Figure 22

Differential Pressures at Take-off Points 3/4 throttle

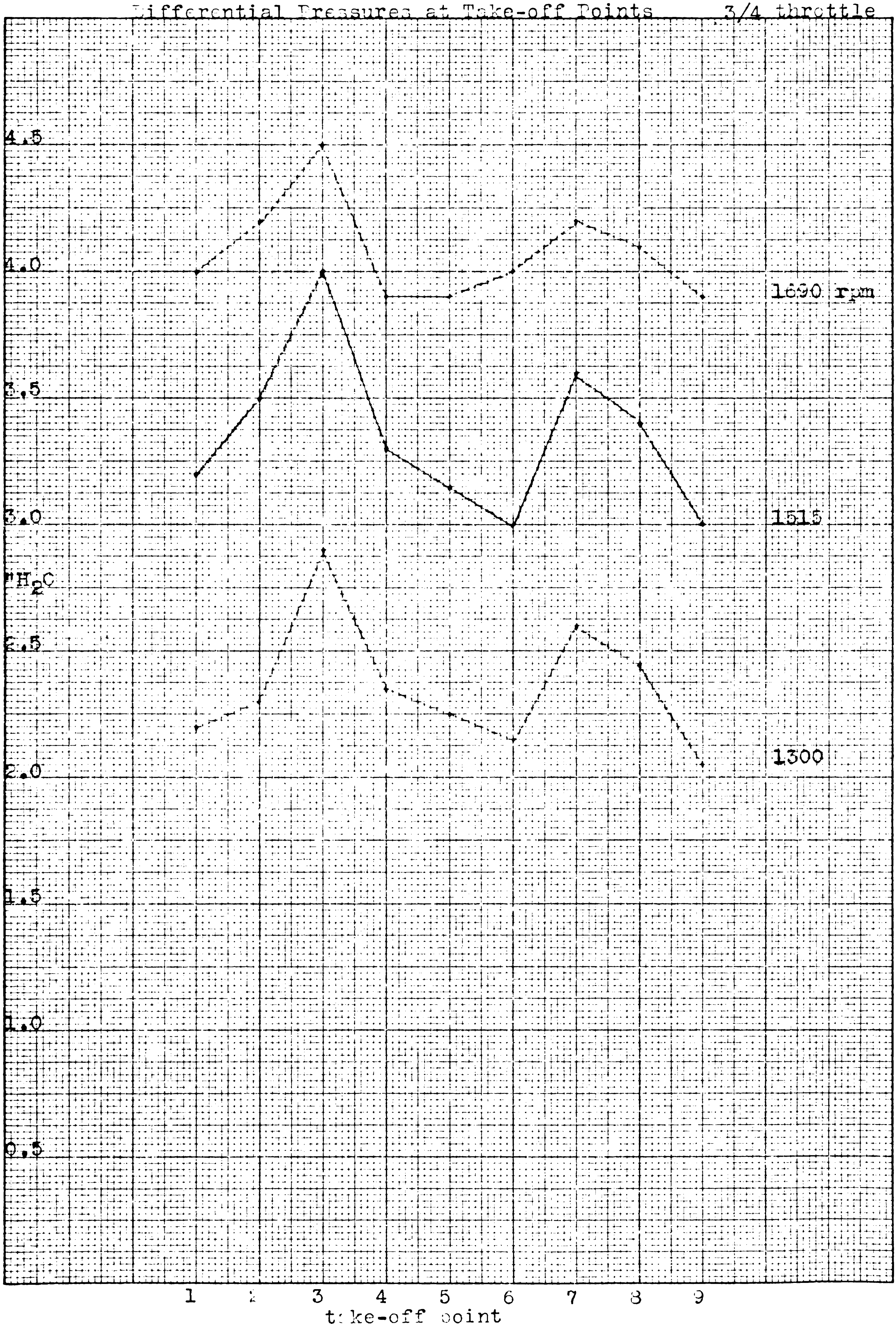


Figure 23

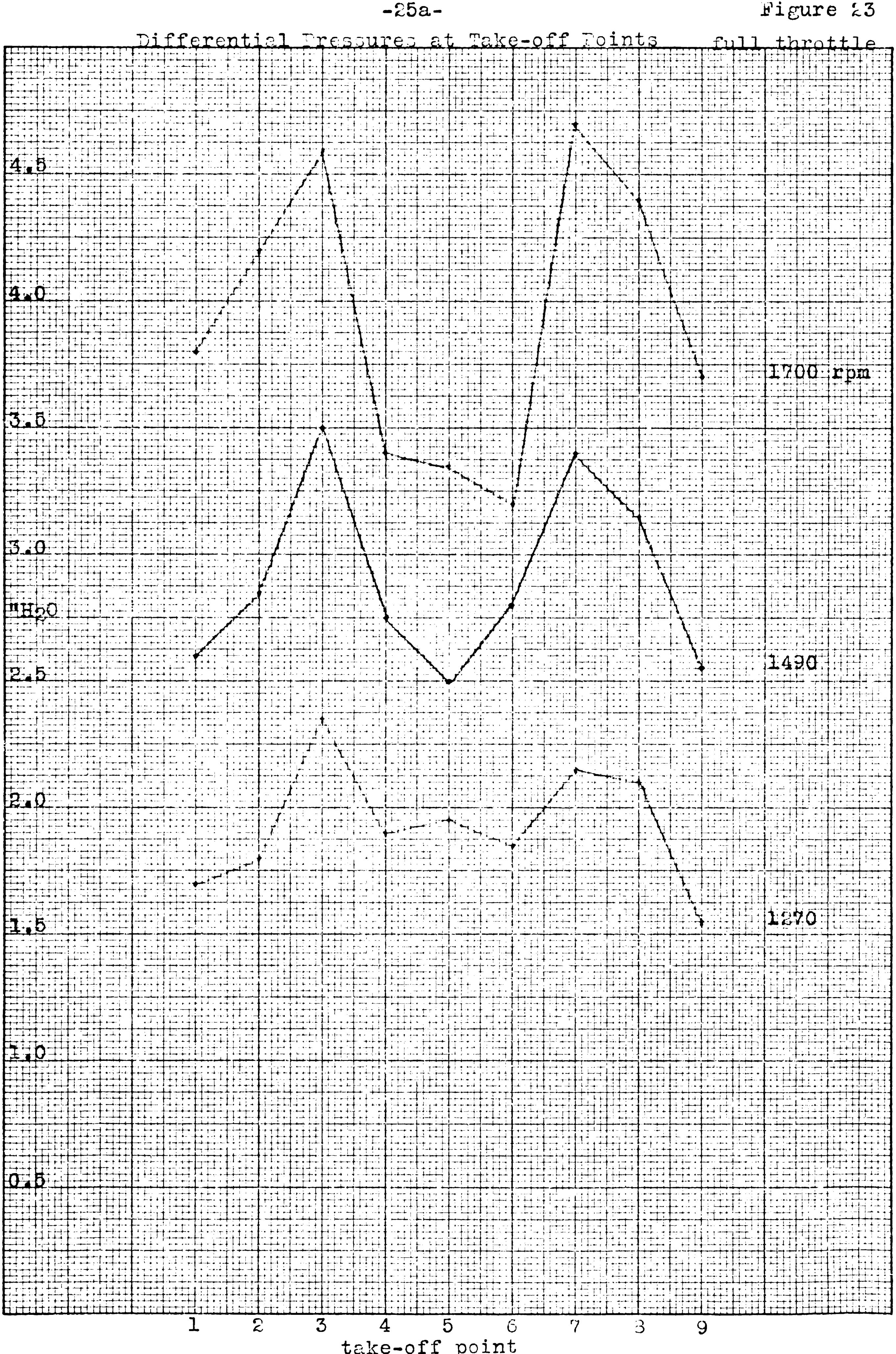

Differential Pressures at Take-off Points
full throttle
4.5
4.0
3.5
3.0
"H_2O
2.5
2.0
1.5
1.0
0.5
1700 rpm
1490
1270
1 2 3 4 5 6 7 8 9
take-off point

Figure 24

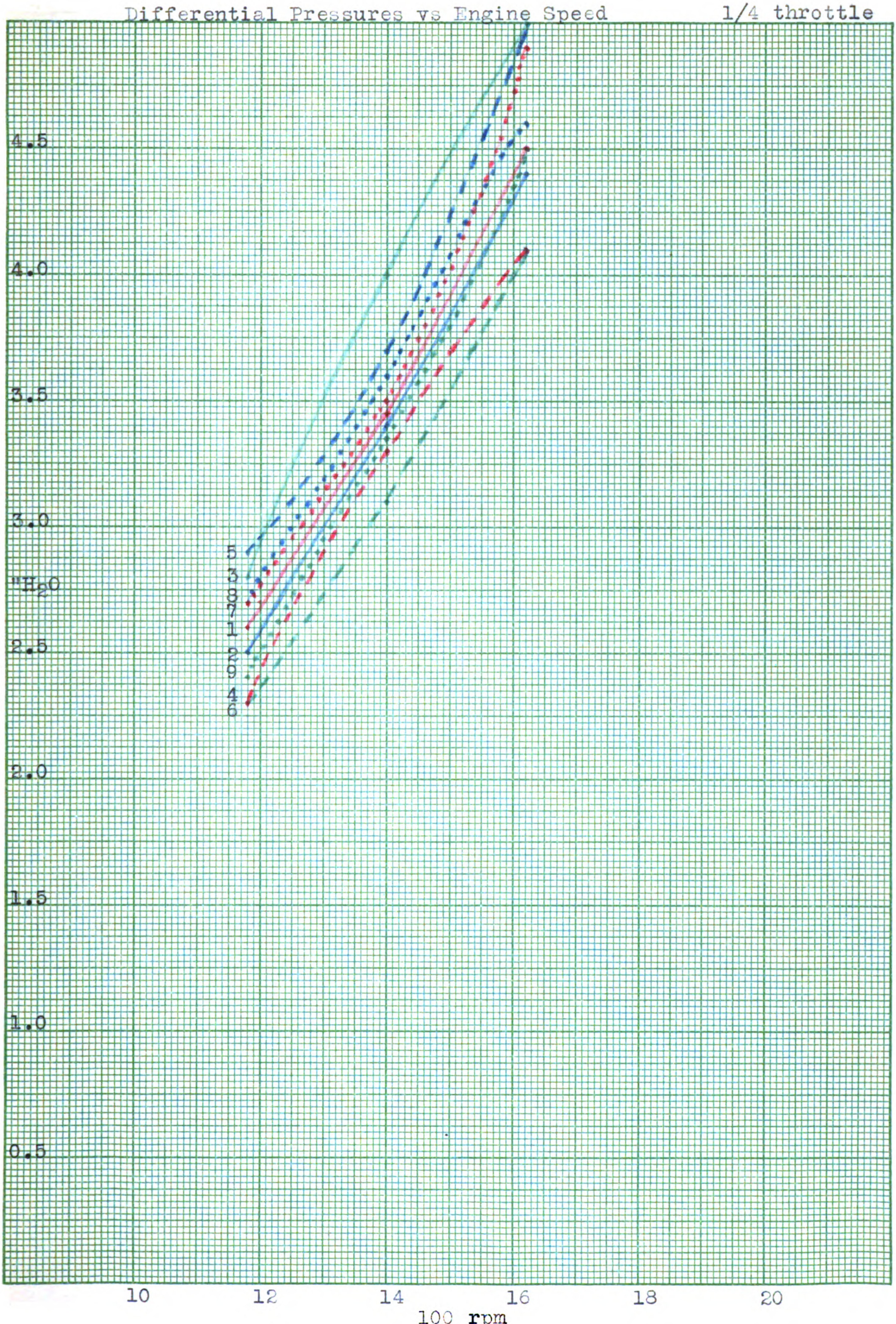

Differential Pressures vs Engine Speed
1/4 throttle
4.5
4.0
3.5
3.0
"H_2O
2.5
2.0
1.5
1.0
0.5
5
3
8
7
1
2
9
4
6
10
12
14
16
18
20
100 rpm

Figure 25

Differential Pressures vs Engine Speed 1/2 throttle

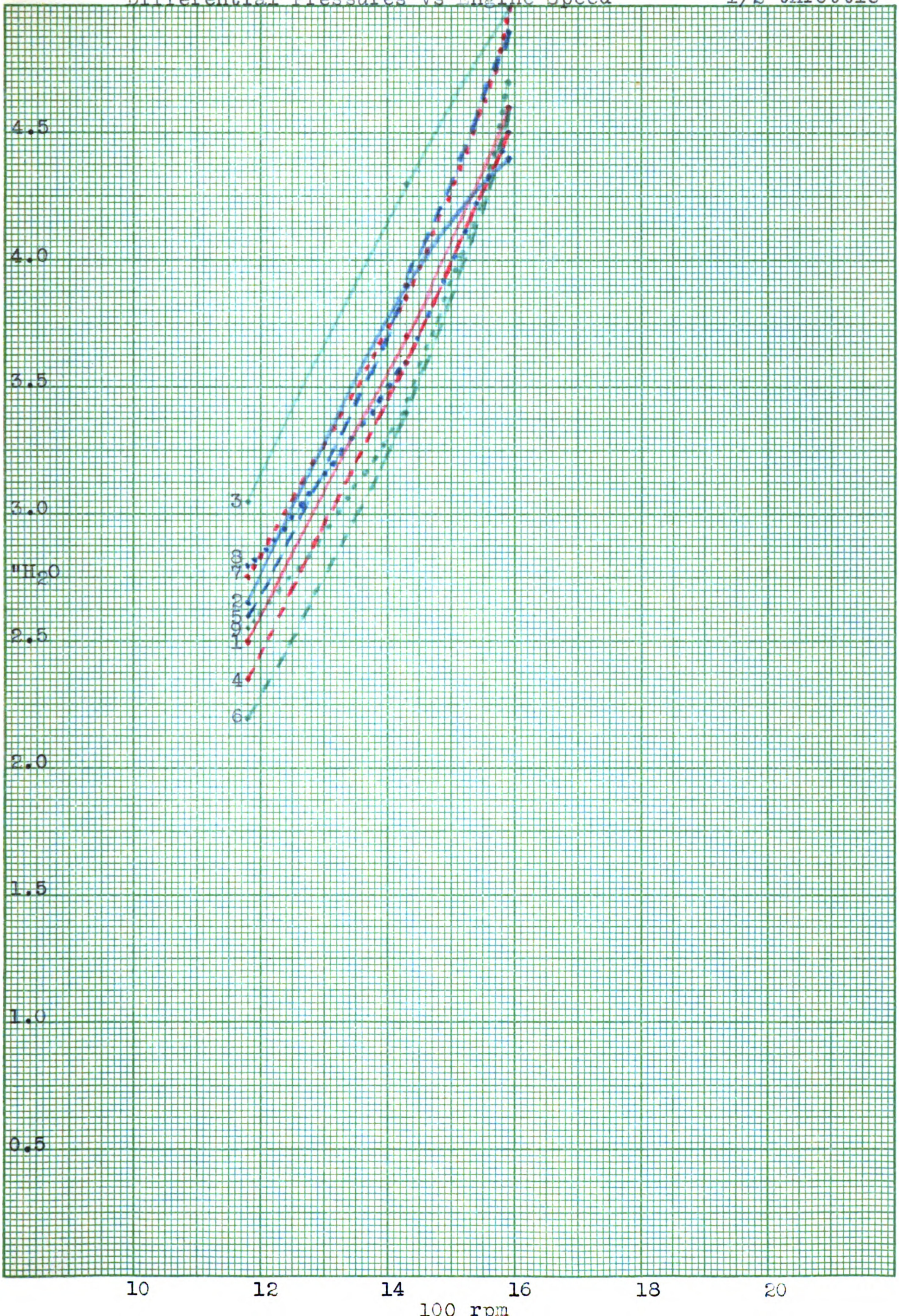


Figure 26

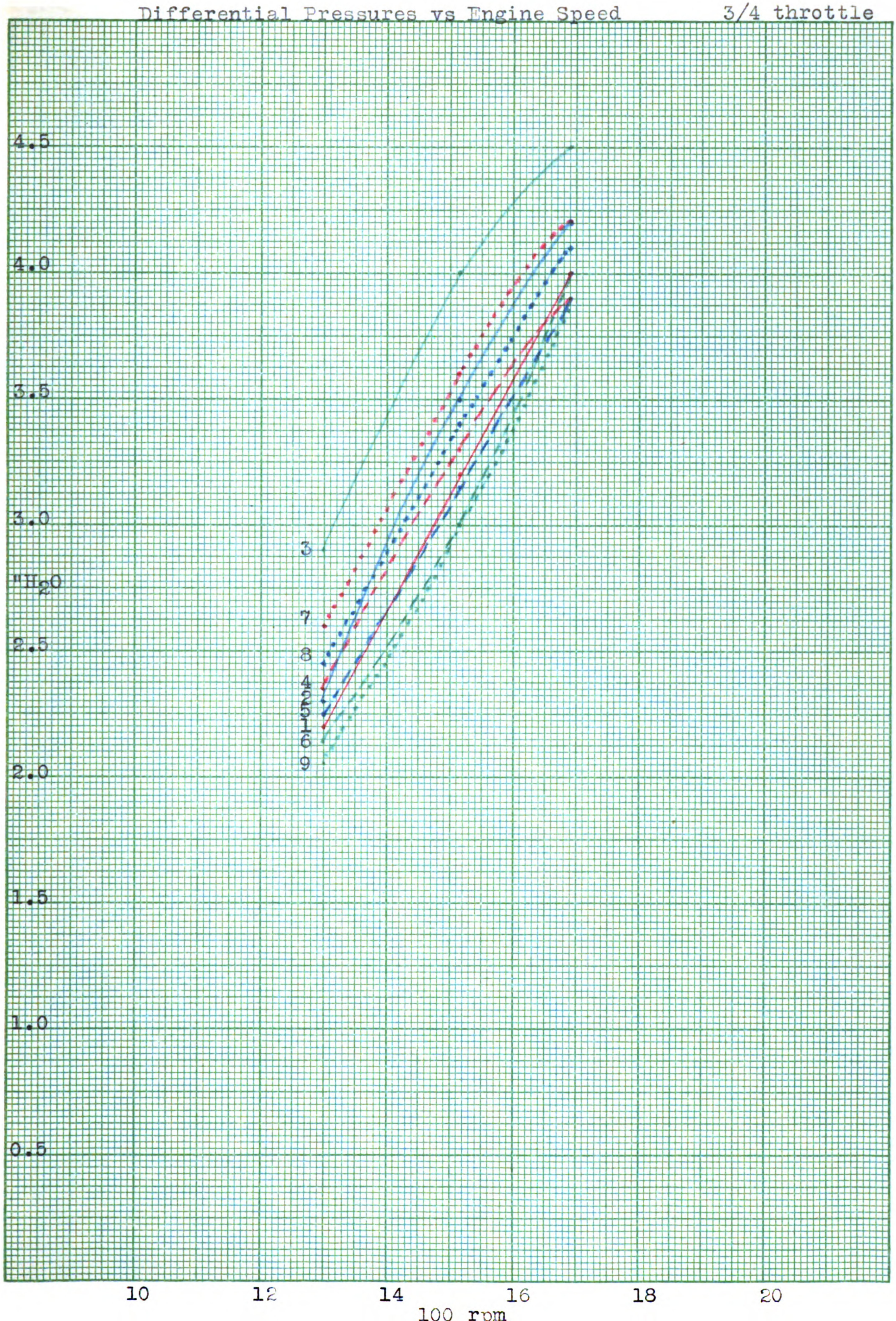
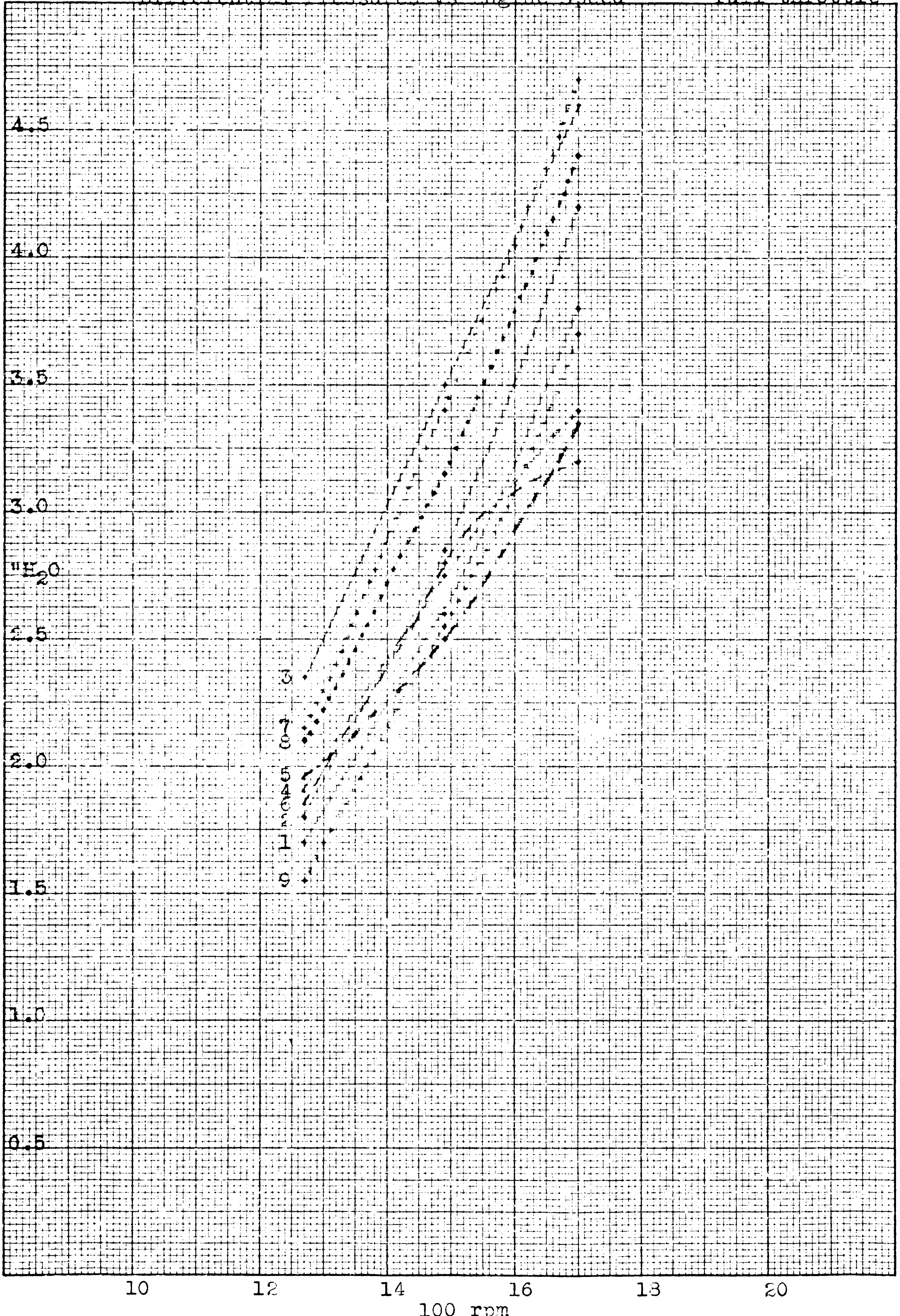

Differential Pressures vs Engine Speed
3/4 throttle
4.5
4.0
3.5
3.0
"H_2O
2.5
2.0
1.5
1.0
0.5
3
7
8
4
2
5
1
6
9
10
12
14
16
18
20
100 rpm

Figure 27

Differential Pressures vs Engine Speed full throttle



MECHANICAL ENGINEERING LABORATORY
MICHIGAN STATE COLLEGE

Running Log of Intake Manifold #1 Test

Observers { Olsen, S.A. / Johnson, B.A. } Date 2-10, 1951

No.	RPM	Fuel time sec.	Fuel fl.oz	Water temp °F	Oil temp °F	Brake arm in.	Torque ft-lbs	HP	Fuel lbs/hr	Corrected Torque ft-lbs	Corrected HP	Fuel lbs/BHP/hr
1	1/4 throttle											
2	1335	119	10.5	165	180	21	69.1	24.15	15.3	71.2	24.85	.616
3	1620	123					82.2	25.4	14.8	84.6	26.15	.566
4	1400	130					93.6	25.0	14.0	96.5	25.75	.545
5	1180	142					99.7	22.4	12.8	102.6	25.03	.556
6												
7	1/2 throttle											
8	1590	81					134.8	40.8	22.5	138.9	42.0	.536
9	1430	89					132.0	36.0	20.5	136.0	37.1	.553
10	1180	108					126.8	28.5	16.85	130.5	29.35	.574
11												
12	3/4 throttle											
13	1690	69					159.2	51.4	26.4	164.0	53.0	.498
14	1515	78					158.3	45.7	23.35	163.0	47.1	.496
15	1300	90					156.5	38.8	20.25	161.2	39.95	.508
16												
17	full throttle											
18	1700	63					167.0	54.1	29.0	172.0	55.7	.520
19	1490	70					165.3	47.0	26.0	170.3	48.4	.537
20	1270	81					159.2	38.5	22.5	164.0	39.6	.568
21												
22												
23												
24												
25												

Remarks:

Figure 28

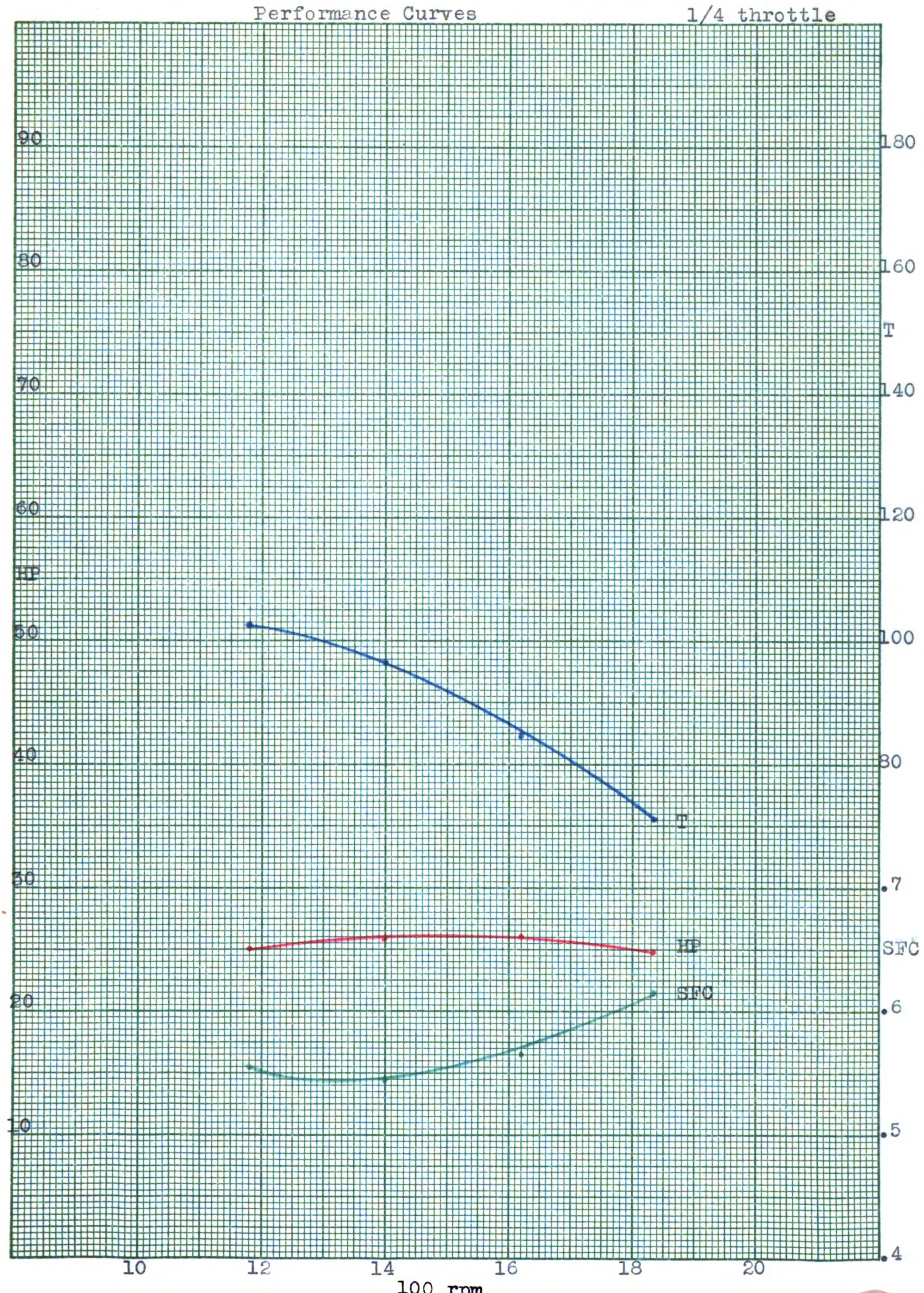

Performance Curves
1/4 throttle
90
80
70
60
HP
50
40
30
20
10
180
160
T
140
120
100
80
.7
SFC
.6
.5
.4
T
HP
SFC
10
12
14
16
18
20
100 rpm

Figure 29

Performance Curves 1/2 throttle

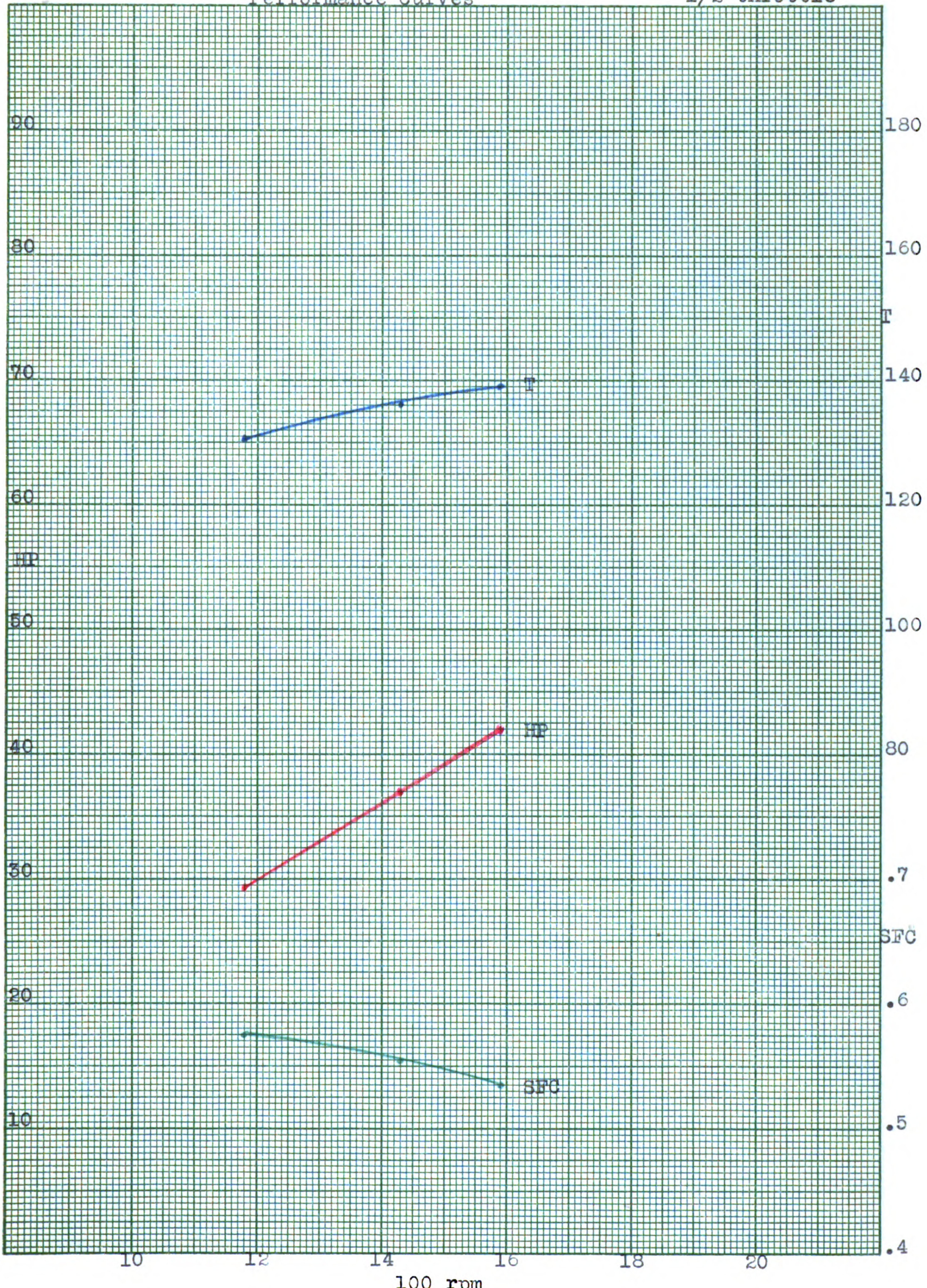
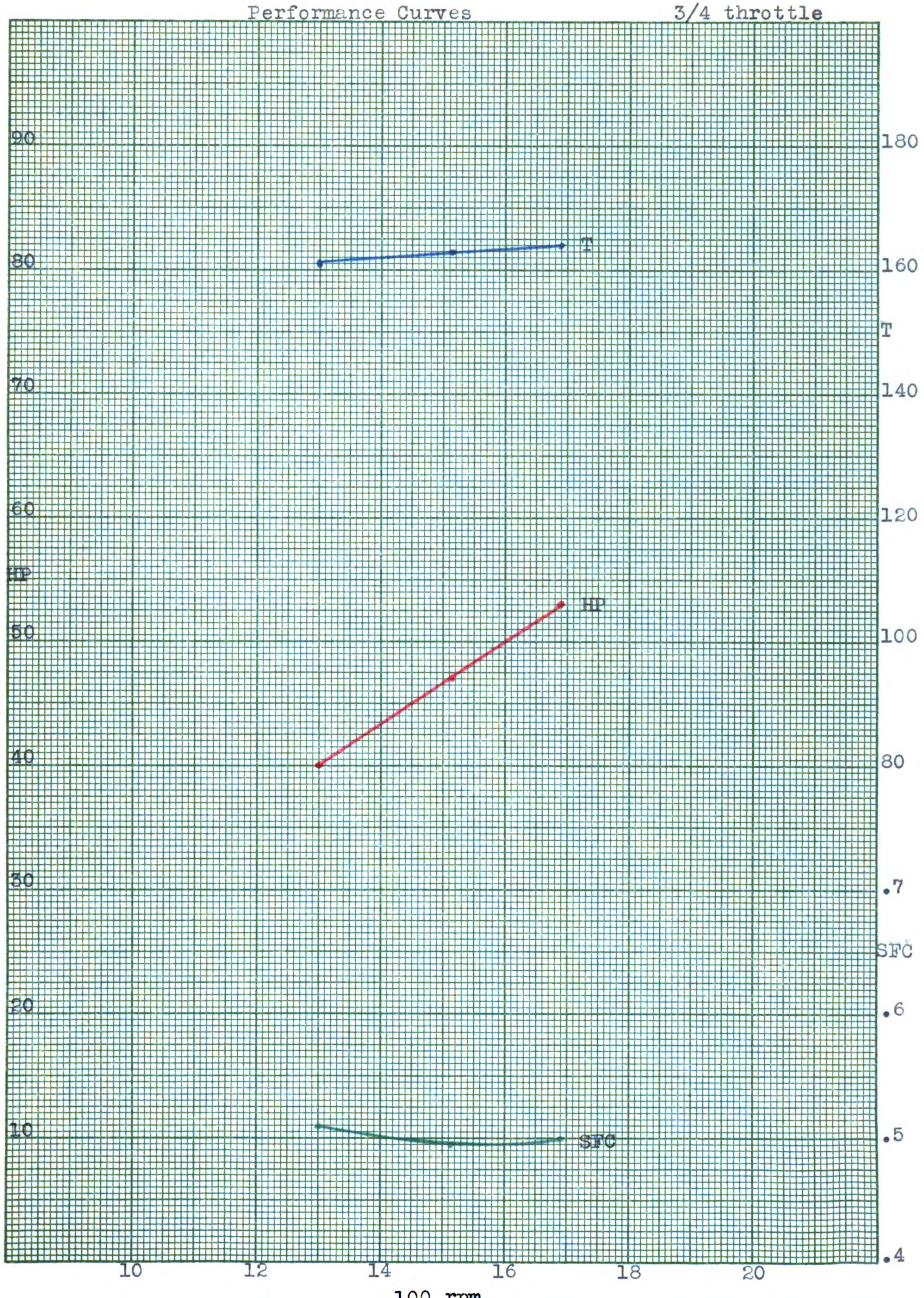
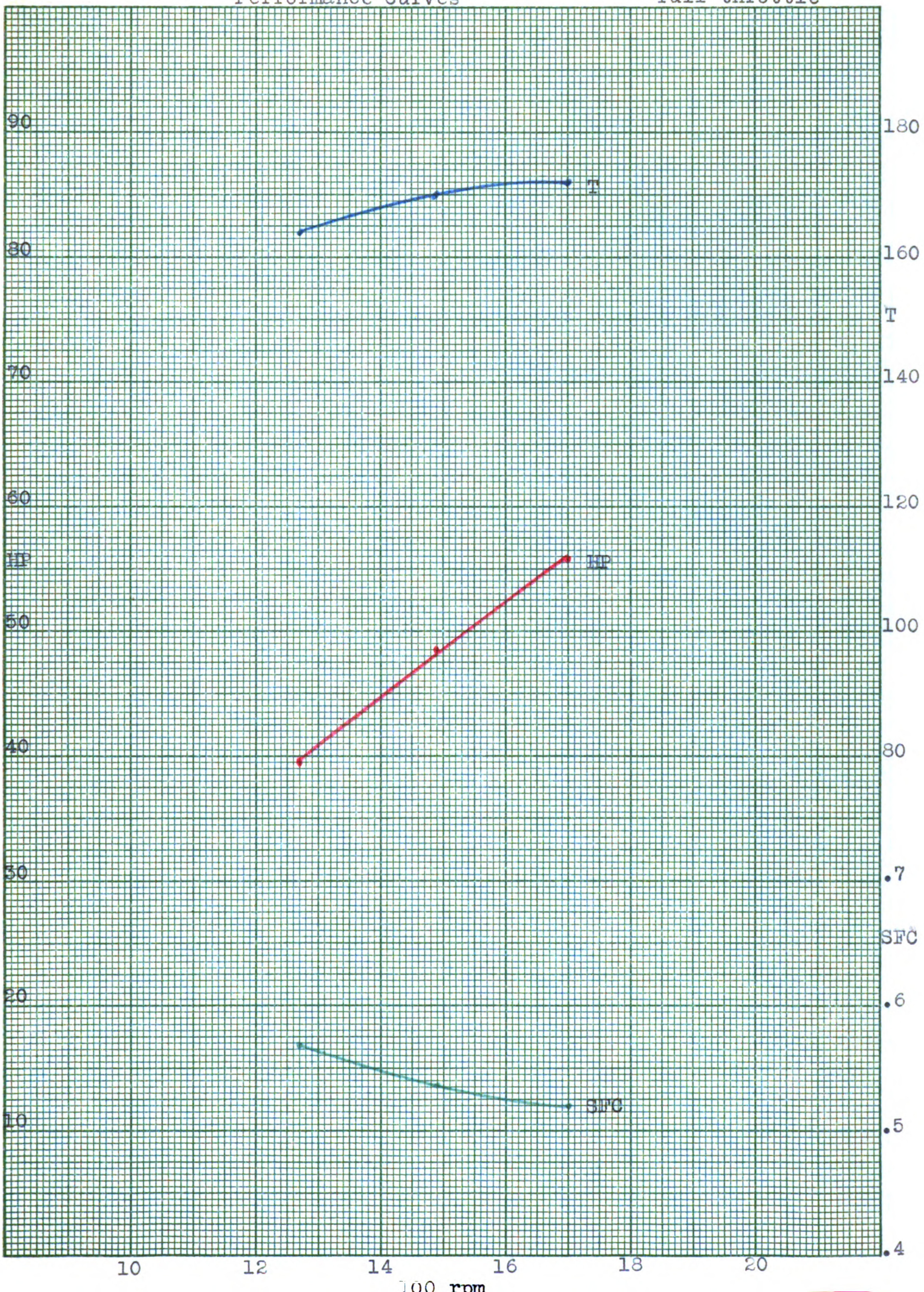


Figure 30



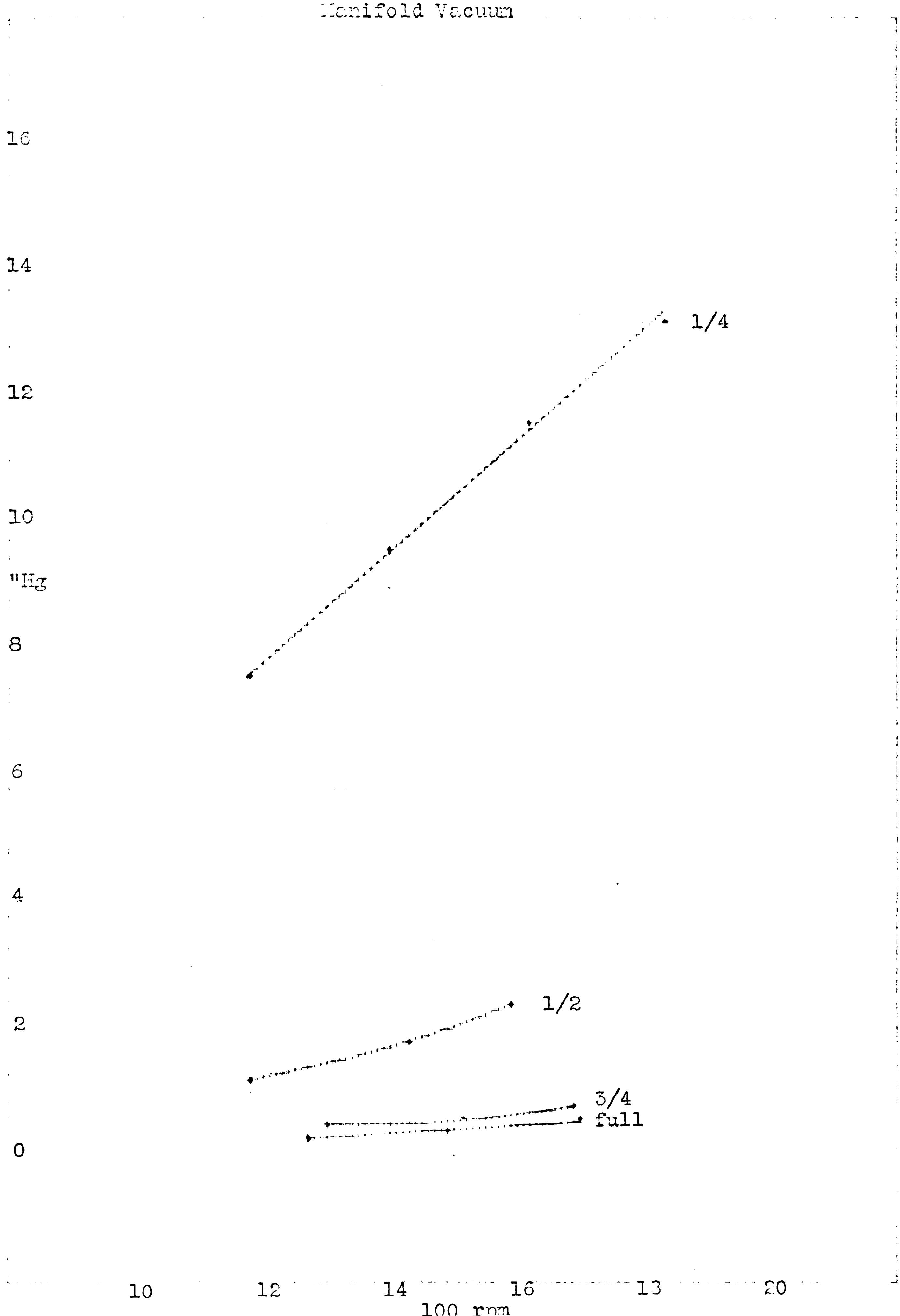
Performance Curves
3/4 throttle
T
HP
SFC
HP
T
SFC
90
80
70
60
50
40
30
20
10
180
160
140
120
100
80
.7
.6
.5
.4
10
12
14
16
18
20
100 rpm



AMERICAN PAD & PAPER CO., HOLYOKE, MASS.

Figure 32

Manifold Vacuum



MECHANICAL ENGINEERING LABORATORY
MICHIGAN STATE COLLEGE

Running Log of Full Throttle Tests

Standard and Type I Manifolds

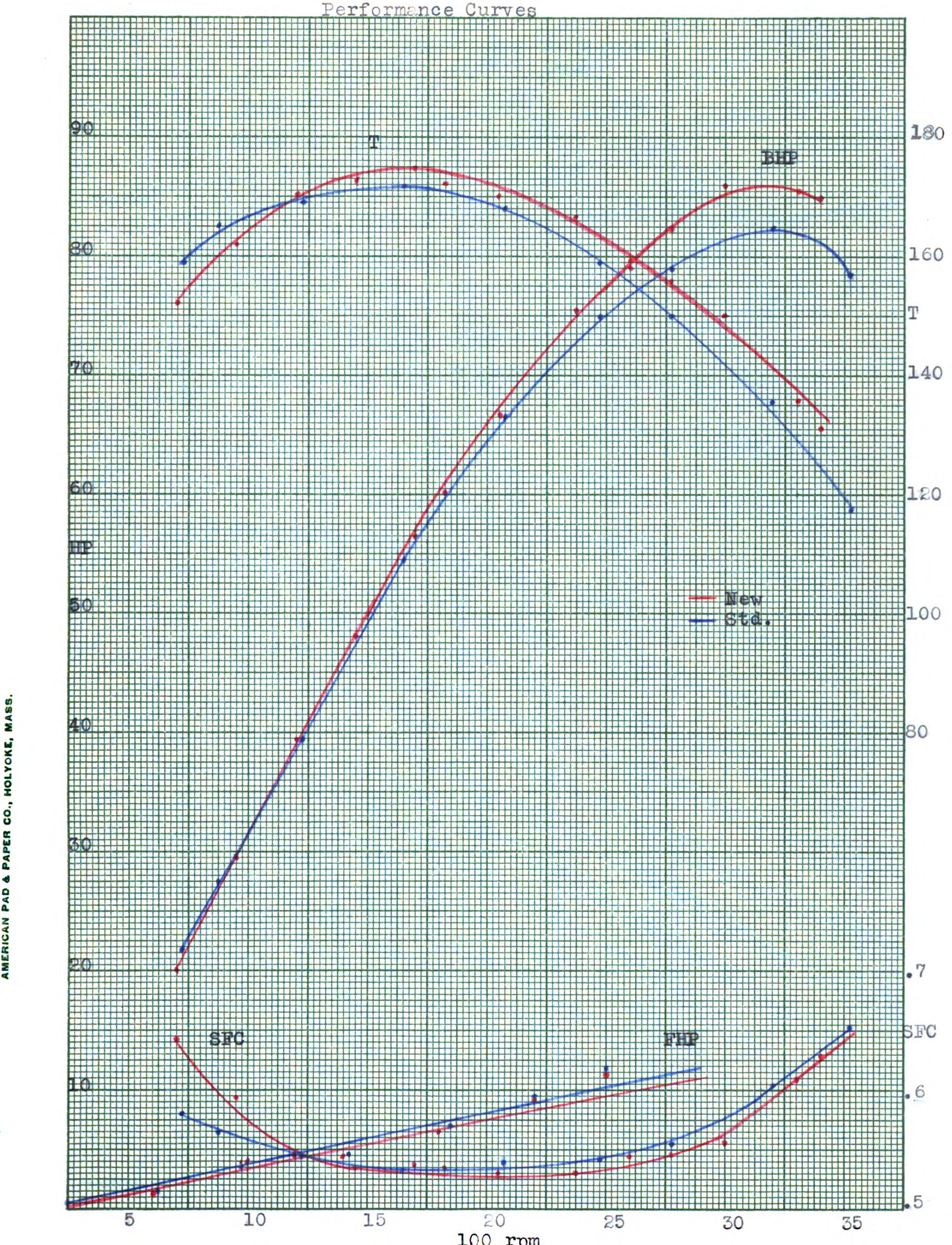
Observers

Date , 19

No.	RPM	lb Beam load	Fuel time sec.	Fuel lb/hr	Torque ft-lbs	HP	Corrected Torque ft-lbs	Corrected HP	Corrected Fuel lb/hphr
1	Standard Manifold				Temp 75°F		Barometer 29.4"Hg.		
2	3530	65	35.5	51.25	113.8	76.2	117.5	78.6	.652
3	3190	75	36.5	49.8	131.2	79.8	135.6	82.4	.605
4	2770	83	41.5	43.9	145.2	76.6	150	79.1	.556
5	2470	88	45	40.5	154	72.5	159	74.8	.542
6	2070	93	51	35.7	162.8	64.2	168	66.4	.538
7	1660	95	63	28.9	166.2	52.6	171.7	54.4	.532
8	1230	93.5	85	21.45	163.7	33.4	169	39.6	.543
9	870	91.5	118	15.43	160.1	26.6	165.3	27.45	.563
10	720	88	144	12.64	154	21.1	159	21.8	.580
11									
12	Type I Manifold				Temp 75°F		Barometer 29.36"Hg.		
13	3400	72.5	34	53.5	126.9	82.2	131.3	85	.629
14	3310	75	35	52	131.2	82.5	135.7	85.5	.608
15	3000	83	38.2	47.7	145.2	83	150.2	85.9	.556
16	2770	86	40.5	44.9	150.5	79.4	155.8	82.2	.546
17	2600	88	42.4	43	154	76.4	159.3	79	.544
18	2370	92	45.6	39.9	161	72.7	166.7	75.3	.530
19	2060	94	51.5	35.35	164.5	64.6	170.2	66.8	.528
20	1830	95	56.7	32.1	166.2	58	172	60.1	.534
21	1690	96.5	61.4	29.65	168.8	54.4	174.7	56.3	.537
22	1460	95.5	71	25.65	167	46.5	172.8	48.1	.534
23	1210	94	84.8	21.45	164.5	38	170.2	39.3	.546
24	960	89.5	103.6	17.55	156.6	23.6	162	29.6	.593
25	695	84	141	12.9	147	19.4	152	20.1	.642

Remarks:

Figure 33


Performance Curves
T
BHP
HP
T
SFC
FHP
New
Std.
100 rpm
AMERICAN PAD & PAPER CO., HOLYOKE, MASS.

MECHANICAL ENGINEERING LABORATORY
MICHIGAN STATE COLLEGE

Running Log of Motoring Tests

Standard and Type I Manifolds

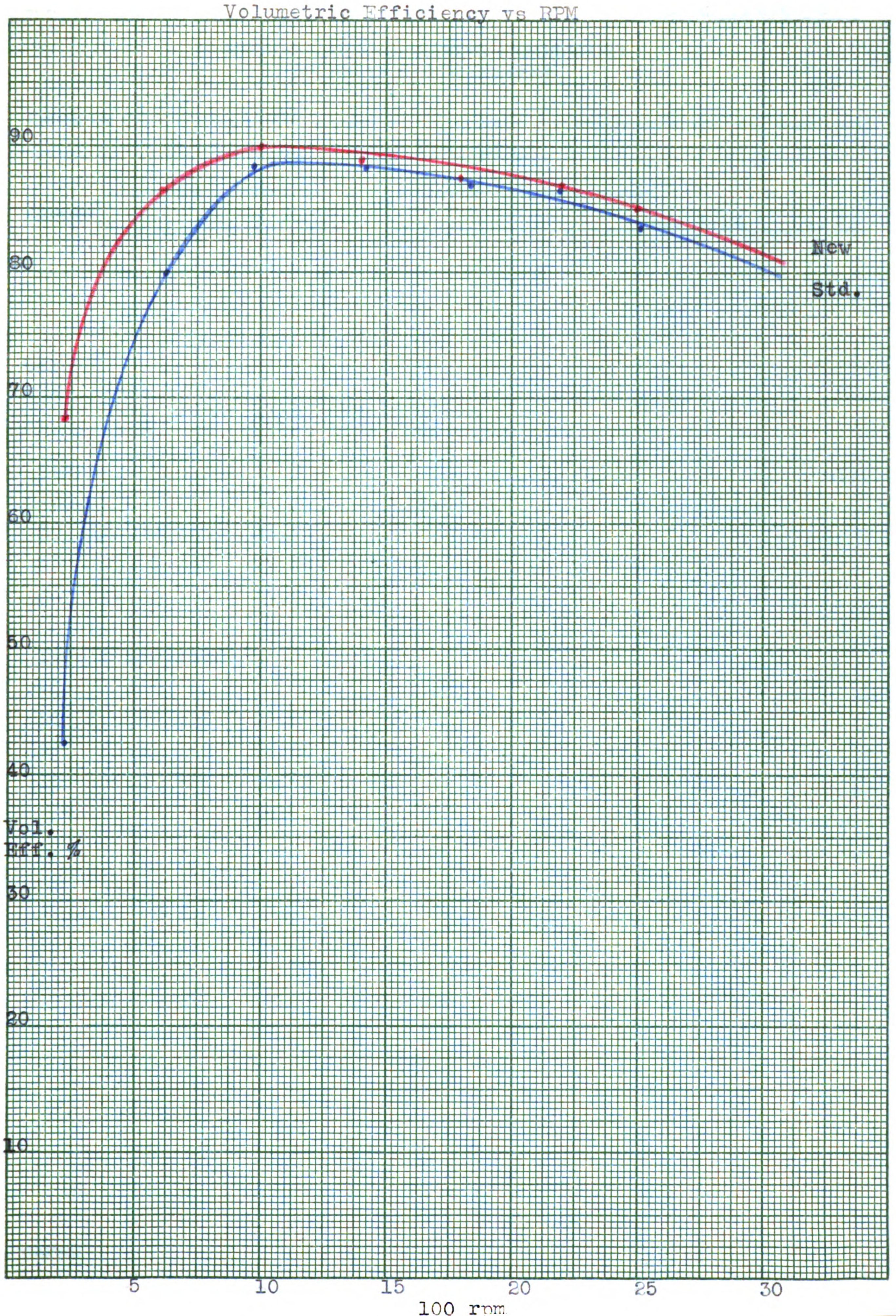
Observers

Temp 79°F

Barometer 29.20"Hg Date 3-19, 1951

No.	RPM	Actual Air flow lb/hr	Beam load lb	FHP	Corrected Ideal Air flow lb/hr	Volumetric Efficiency %
1	Standard Manifold					
2	230	25	10.1	.47	58.6	42.7
3	630	128	12.2	1.55	160.5	79.8
4	970	218	13.3	2.61	247.5	88.3
5	1420	320	16.1	4.62	362.5	88.3
6	1840	407	19.1	7.1	469	87
7	2195	485	21.3	9.45	560	86.6
8	2510	533	23.3	11.82	640	83.3
9						
10	Type I Manifold					
11	230	40	9.9	.46	58.6	68.3
12	615	136	11	1.37	157	86.6
13	990	227	13.3	2.66	252.5	89.9
14	1400	316	15.3	4.33	357	88.5
15	1790	400	13	6.51	456	87.6
16	2200	433	20.7	9.2	501	87
17	2500	542	22.5	11.35	633	85
18						
19						
20						
21						
22						
23						
24						
25						

Remarks:



Volumetric Efficiency vs RPM
90
80
70
60
50
40
Vol.
Eff. %
30
20
10
5
10
15
20
25
30
100 rpm
New
Std.

MECHANICAL ENGINEERING LABORATORY
MICHIGAN STATE COLLEGE

Running Log of Road Tests

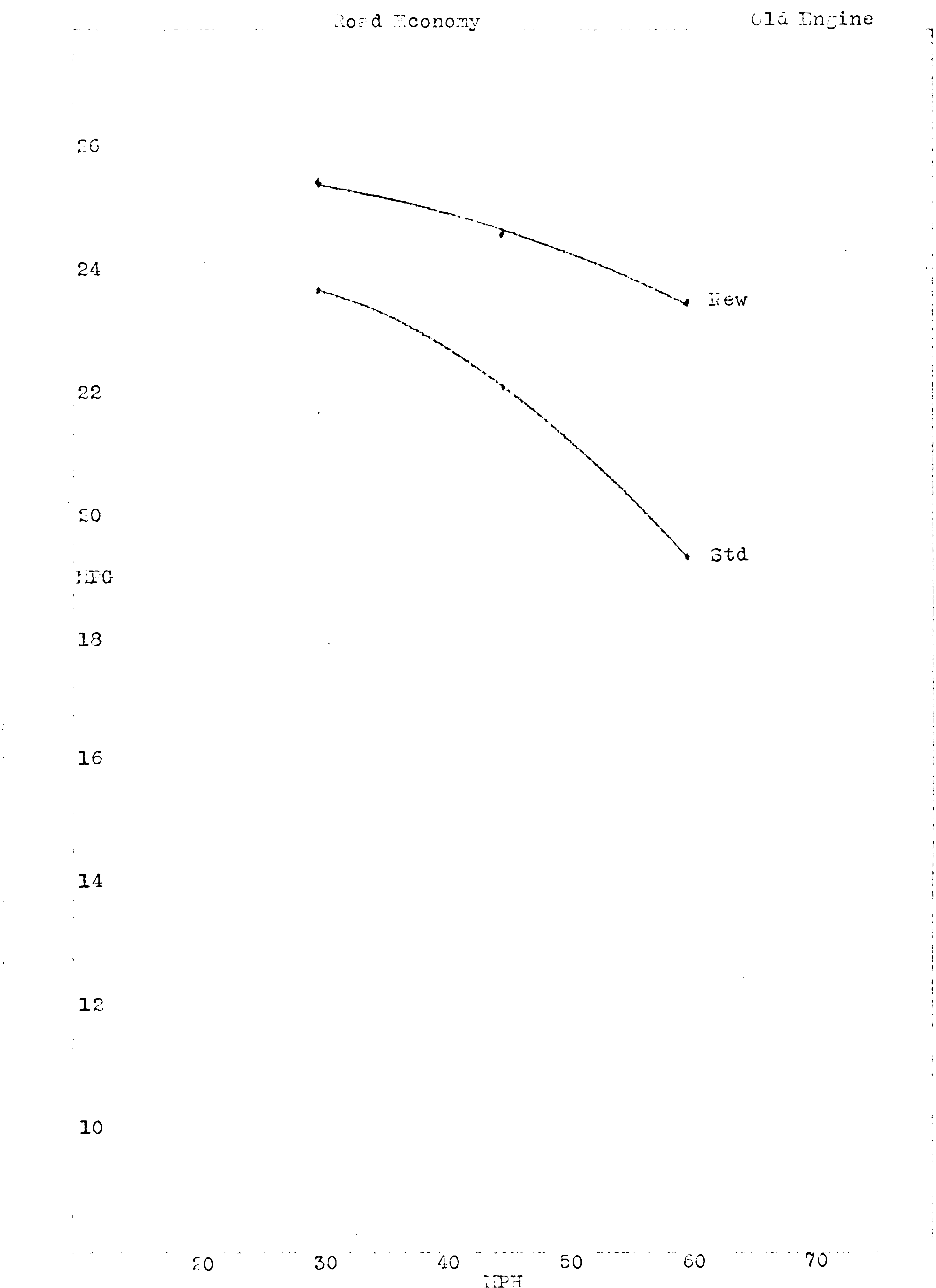
Standard and Type I Manifolds

Observers: Olsen, S.A.; Warrell, F.J.; Olsen, M.G.

Date 3-24, 1951

Car Speed MPH	Fuel time sec	Fuel cc	miles / gallon	Acceleration time sec.
Standard Manifold				
0-60				19.63
30	120.79	160	23.75	
45	75.09	160	22.15	
60	49.38	160	19.4	
Type I Manifold				
0-60				13.35
30	120.35	160	25.5	
45	83.5	160	24.65	
60	59.3	160	23.55	

Remarks:



Road Economy
Old Engine
26
24
22
20
MPG
18
16
14
12
10
New
Std
20
30
40
50
60
70
MPH

MECHANICAL ENGINEERING LABORATORY
MICHIGAN STATE COLLEGE

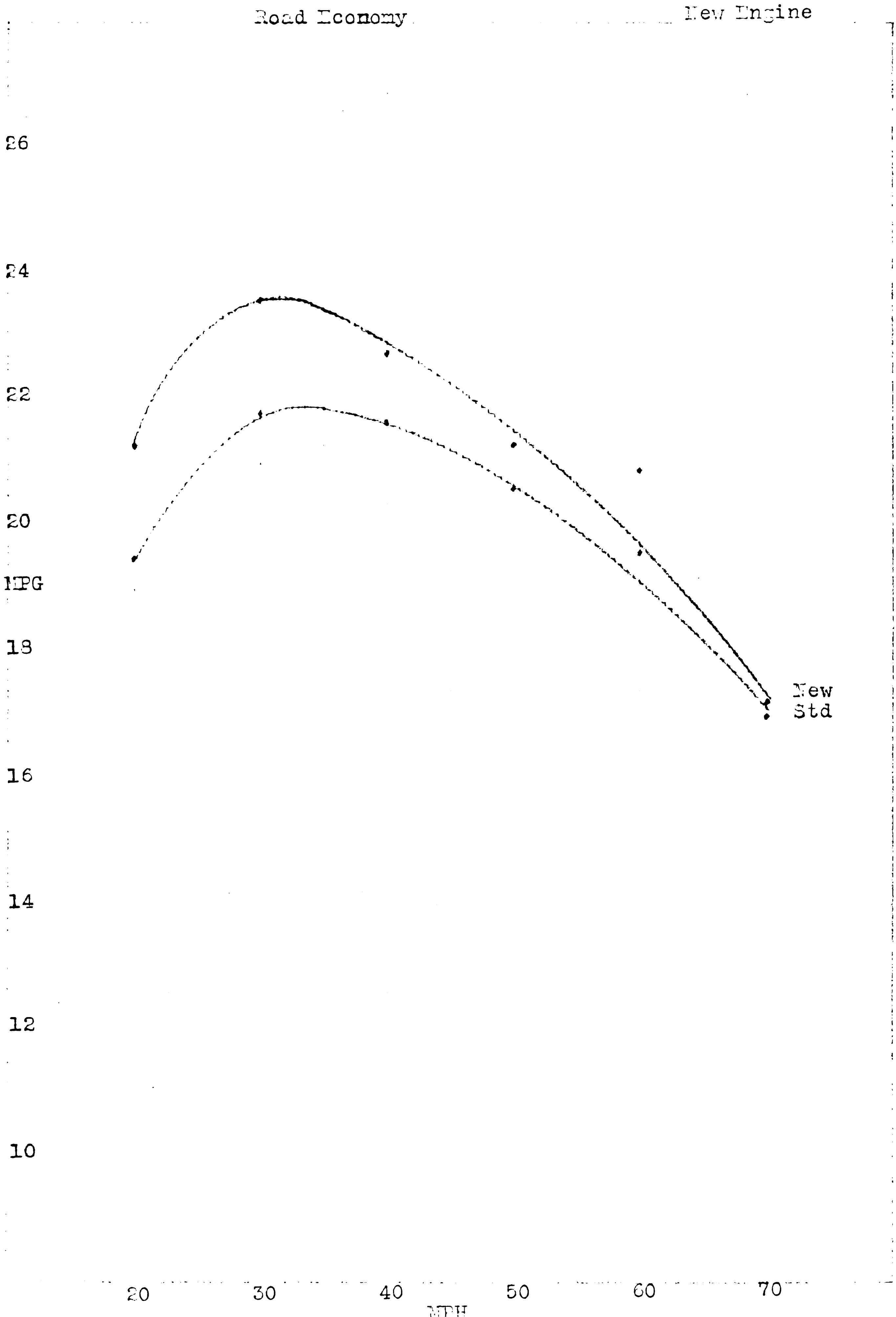
Running Log of Road Tests New Engine
Standard and Type I Manifolds

Observers Olsen, S. A. / Warrell, F. J. Date 4-21, 1951

No.	Car Speed MPH	Fuel time sec.	Fuel cc.	miles/gallon	Acceleration time-sec.
1	Standard Manifold				
2	0-60				18.6
3	20	148.5	160	19.5	
4	30	110.9	160	21.8	
5	40	82.5	160	21.65	
6	50	62.8	160	20.6	
7	60	49.8	160	19.6	
8	70	37	160	17.0	
9					
10	Type I Manifold				
11	0-60				16.8
12	20	162	160	21.3	
13	30	120	160	23.6	
14	40	86.7	160	22.75	
15	50	65	160	21.3	
16	60	53	160	20.9	
17	70	37.4	160	17.2	
18					
19					
20					
21					
22					
23					
24					
25					

Remarks:

Figure 36


Road Economy
New Engine
26
24
22
20
MPG
18
16
14
12
10
New
Std
20
30
40
50
60
70
MPH

SAMPLE CALCULATIONS

Dynamometer Test

Given: 216.5 cu in, 6 cyl, OHV, automotive engine

driving beam load = 93 lb

motoring beam load = 21 lb

dynamometer constant = 3000

brake arm = 21 in

room temperature = 75° F

atmospheric pressure = 29.4 "Hg

fuel time = 51 sec for 310 cc

fuel constant = .001628 lb/cc

measured air consumption = 440 lb/hr @ 60° F and 14.7 psi

actual rpm = 2000

standard dry air density = .0764 lb/cf

negligible vapor pressure

$$BHP = \frac{WN}{3000} = \frac{93 \times 2000}{3000} = 62 \text{ HP}$$

$$\text{Torque} = WR = 93 \times 1.75 = 162.8 \text{ ft-lb}$$

$$FHP = \frac{WN}{3000} = \frac{21 \times 2000}{3000} = 14 \text{ HP}$$

$$\text{Fuel lb/hr} = \frac{310\text{cc}}{51 \text{ sec}} \times \frac{3600 \text{ sec}}{1 \text{ hr}} \times \frac{.001628 \text{ lb}}{\text{cc}} = 35.7 \text{ lb/hr}$$

$$\text{Correction factor} = \frac{29.92}{29.4} \sqrt{\frac{535}{460}} = 1.097$$

$$\text{Corrected BHP} = 1.097 \times 62 = 68 \text{ HP}$$

$$\text{Corrected T} = 1.097 \times 162.8 = 178.3 \text{ ft-lb}$$

$$\frac{\text{lb fuel}}{\text{bhp-hr}} = \frac{35.7}{62} = .525 \frac{\text{lb}}{\text{bhp-hr}}$$

Theoretical air consumption = m_t

$$m_t = \frac{216.5 \text{ cu in}}{2 \text{ rev}} \times 2000 \frac{\text{rev}}{\text{min}} \times \frac{60 \text{ min}}{\text{hr}} \times \frac{1 \text{ cf}}{1728 \text{ cu in}} \times \frac{.0764 \text{ lb}}{\text{cf}}$$

$$m_t = 575 \frac{\text{lb}}{\text{hr}}$$

$$\text{Volumetric efficiency} = \frac{m_a}{m_t} \times 100\% = \frac{440}{575} \times 100\% = 76.5\%$$

Road Test

Given: actual car speed = 60 mph

fuel time = 49.38 sec for 160 cc

$$\text{mpg} = \frac{60 \text{ miles}}{\text{hr}} \times \frac{1 \text{ hr}}{3600 \text{sec}} \times \frac{49.38 \text{ sec}}{160 \text{ cc}} \times \frac{3790 \text{ cc}}{\text{gal}} = 19.45 \frac{\text{miles}}{\text{gal}}$$

MAJOR EQUIPMENT SPECIFICATIONS

Engine -- 1949 Chevrolet

Serial Number -- GAA 81241

Six cylinder -- overhead valves

Bore and stroke - 3 1/2 x 3 3/4 in

Displacement -- 216.5 cu in

Compression ratio -- 6.5 to 1

Engine -- 1948 Chevrolet

Serial Number -- FAA 270706

Six cylinder -- overhead valves

Bore and stroke -- 3 1/2 x 3 3/4 in

Displacement -- 216.5 cu in

Compression ratio -- 6.5 to 1

Dynamometer -- General Electric

Serial Number -- 1504076

Range -- 100 HP absorbed @ 1050 to 3500 rpm

MANIFOLDING AND DISTRIBUTION

General Conditions

Before any intelligent investigation can be made of the automotive intake manifold, it is necessary to be concerned with some of the general conditions of the manifolding and distribution problem. The problem does exist in endeavoring to achieve equality of fuel distribution to all cylinders of the engine. That is to say that the gasoline vapor, air, and liquid fuel phases present in the manifold do not divide themselves equally among all of the cylinders.

The function of the carburetor is basicly to meter the fuel to the supply air that is drawn into the engine. The sole purpose of the manifold is to distribute the mixture of fuel and air without too many losses from friction, bends, or the like. It is necessary for the manifold to accomplish its function with a high enough flow velocity to hold the liquid fuel in suspension. It is also necessary to add enough heat to the mixture to obtain good fuel vaporization, yet maintain high volumetric efficiency. There is a definite point to reach here, depending upon whether the engine is intended for good economy of operation or for high power output.

The air-fuel ratios used today vary anywhere from 8 to 20 parts of air to one part of fuel, by weight. In the average automobile engine, maximum power is obtained with an air-fuel ratio in the vicinity of 13 to 1, and maximum economy at wide open throttle is reached with a ratio near 14 to 1. With the engine operating at part throttle, economy is obtained with A-F relationships varying from 15 to 18 to 1, depending, of course, on the actual engine speed. Naturally the engine speed and load requirements design the mixture necessities.

In order for any engine to develop its maximum power, speaking of full-throttle operation, it is necessary for all cylinders to operate at optimum air-fuel settings simultaneously. It is not too difficult to visualize that if some of the cylinders are operating lean and some rich it is impossible for the engine to develop high power. Bad distribution characteristics of the manifold brings about the necessity of supplying the engine with an overall rich mixture to insure most of the cylinders getting near the correct power mixture. An engine can operate more readily on the rich, or power side, of the air-fuel ratio settings. Of course, engine smoothness has to be considered so as to not get too rich a mixture that will choke up the already rich cylinders and induce misfiring and overall roughness. Tests have shown that the operating air-fuel ratios control the general knock susceptibility of the engine also. From this alone the spark requirements for the different cylinders in some engines may vary as much as 15 degrees between them. The distributor setting has to be retarded so that the cylinder most likely to knock will not do so, but the one difficulty is that some fuels knock on richer or leaner mixtures than others.

Unfortunately, no one has yet developed a carburetor that will perfectly vaporize the fuel. The mixture of vapor, air, and liquid particles may be homogeneous at some locations in the manifold, and if the homogeneity could be maintained most of the distribution problems would be solved. A perfectly vaporized liquid, or use of a gaseous fuel, would greatly aid the solution. However, in the actual condition, bends in the manifold branch and the mixture inertia throw off the balance between the various components of the flow. Some cylinders end up rich, some lean, depending upon the manifold conditions. A lot of effort has been directed towards improving the distribution through addition of heat to the mixture. It was once thought that the hotter the manifold and carburetor air the better would be distribution. Recent tests, however, have proven that this is not necessarily so in all cases. In any event, if the mixture temperature is raised, torque and power drop off. This heating will increase the volume and reduce the weight of fuel available for combustion. The volumetric efficiency will drop off about 1% per 10° F temperature rise at constant spark advance, and the drop is larger if the spark is retarded to avoid ping in any cylinder. Because of its adverse affect on the volumetric efficiency, just enough heat is used to make the engine operation commercial.

One solution to the problem of proper manifolding and distribution would not necessarily fit for all conditions of speed, load, and mixture ratios. The need is to maintain homogeneity throughout all operating ranges, and distribute the mixture equally to all cylinders. The shape of the intake manifold branches must be so designed as to avoid any segregation of the liquid particles from the vapor and air. In fact, it is desirable to have a design that through one means or another will remix any particles that have separated out of the flow. Attempts have been made at manifold designs with emperically developed smooth curved bends to avoid this segregation. In most cases, however, the liquid fuel still deposits on the inner radius of the bend.

Almost all of today's in-line engines utilize siamese intake ports; therefore, the inner cylinder usually ends up the richer of the two. Sometimes though, the air velocity will twist the liquid particles over to the outer radius and they will enrichen the outer cylinder. Ethyl Corporation has set up an engine with a clear plastic intake manifold and has observed the phenomenon to be true at various loads and engine speeds. It is not too difficult to see that this one factor alone would be hard to control or correct. It was once thought that the outer cylinder was always the richer one, for the simple reason that the very inertia of the liquid particles themselves would force them to the outside on a bend. However, the various parts of the mixture are traveling down the manifold branches at unequal speeds. The velocity varies from a maximum at the center of the tube to zero at the walls. At the bend, centrifugal force pushes the center portion of the mass to the outside and by this the slow moving sections are forced to the inner circle. The liquid particles can separate themselves from a slow moving section more easily, so they drop out on the inside and usually run into the inner cylinder.

Some manifolds, such as the rake type, utilize sharp corners as a condusive toward creating a turbulence to hold the heavier liquid particles in suspension in the flow, or to remix any of those that may have separated along the way. It almost goes without adding, that the size of the manifold section determines the velocity of the mixture flowing through it. Common practice holds that the minimum velocity tolerable is about 15 feet per second.

Along with a host of other items, even the firing order of the engine affects the fuel distribution. Siamesed cylinders should fire at equal intervals so as to tend toward equal fuel distribution between the two of them. Any cylinder following a longer time interval between inductions, naturally receives the richer mixture because of a greater build up. Firing order and bends often combine their effects to make one particular cylinder excessively rich. This is especially true in an in-line six cylinder engine. In an eight cylinder engine it is not difficult to arrange the firing order so as to counter the bend effects on distribution.

It goes without saying that a balanced carburetor is necessary before any accurate conclusions can be drawn concerning the problem of fuel manifolding and distribution. Even some slight eccentricity of the air cleaner can manifest itself into an unbalanced condition in manifold branch distribution. All of the factors mentioned here affect this difficult problem of distribution, and many factors have not yet been segregated as such. No way has yet been devised to insure a good manifold design. Much thought and work with emperical testing has gone toward achieving good operation, but the concrete evidence of fact has yet to come forth with a sure-fire method for intake manifold design.

Manifold Testing and Design

Automotive intake manifolds are tested mainly on a basis of power and torque as compared to another manifold. They are always tested this way and many proponents of the method say that it is the most important. This author doubts that stand and holds that economy is just as important as any power or torque output, if not more so. In these days of rapidly rising costs it seems that the engineers should cease pushing power and take up the challenge of striving for greater economy of operation in the automobile engine. But regardless of the goal sought, the big shortcoming of any test is not knowing what is going on inside the manifold and this is necessary to any intelligent conclusions.

Equality of distribution between all cylinders of the engine is the prime considerate. At the present time there are two main methods used to measure this equality, or inequality, of distribution - exhaust gas analysis and spark plug temperature measurements. Both methods are dependable, but the first mentioned is the more rapid.

An exhaust gas analysis depends on a definite relationship between the products of combustion and the mixture conditions prior to combustion. A chart made up by A. D'Aleva and W. G. Lovell and published in the March, 1936 issue of the SAE Journal, illustrates one such relationship. Most commercial fuels in use today correspond closely to the C_8H_{17} hydrocarbon used in this example, and the chart can be taken to be indicative in general to all. However, as the hydrogen-carbon relation varies, so do the products of combustion. Naturally any method of testing would not be accurate without a hydrogen-carbon test of the fuel used being made first. It is a tedious process to take an Orsat analysis of the combustion process from each cylinder over a speed range. There has been developed an automatic Orsat sampler which will give CO_2 and CO_2 plus O_2 quantities in 15 to 30 seconds. This new equipment will also handle leaner mixtures than some other instruments on the market will.

Several automatic analyzers indicate the air-fuel ratio as the gases pass through them. They generally make use of chart relationships such as mentioned before, and measure one or more constituents of the gas. One of the more dependable instruments of this type makes use of the Wheatstone bridge. One leg of the bridge is exposed to the exhaust gases and the other to water-saturated air. Both of the elements are heated to 200° F initially and through this the percentage of CO and H_2 in the gases are measured by their property of heat conductivity. CO has 1.6 and H_2 has 4 times the heat conductivity of air. As a concentration of these gases increases around the exhaust leg it is cooled. This cooling lowers the resistance in the bridge leg, causing more current to flow, which is measured on a connected galvanometer which reads directly in air-fuel ratios. Unfortunately this convenient apparatus operates only below 14.7 to 1 ratios -- the theoretical point of complete combustion. Most engines operate at part throttle on a ratio from 14 to 18 to 1, and the analyzers have proven to be of no good use for a road-load analysis.

The spark plug temperature measurement method of distribution determinations makes a fuel-temperature fish-hook for each cylinder. The maximum temperature point of each fish-hook will occur at approximately the same air-fuel ratio, and the plugs have a thermocouple at one terminal point to measure this temperature. The relative order in which the maximum temperature is reached in each cylinder as the total fuel delivery to the engine is changed shows the relative distribution to each cylinder at any given fuel rate. One disadvantage is that the method is slow. Four or five points must be taken with different fuel deliveries for each speed and throttle position. Manifolds have to be checked on the road for acceleration and top speed, as well as on the dynamometer. Distribution checks and cold start tests are essential in both phases.

The final internal shape of a manifold can be determined emperically by the long and tedious process of testing models of different sizes and shapes. One company has devised a less expensive method for this practice. They split the manifold section and then build up or change the interior by cutting away the metal, or adding to it with wax or solder.

The addition of heat is essential to bring the intake manifold up to operating temperature after starting a cold engine. The most common method of heating is to use a hot spot between the intake and exhaust manifold. Hot exhaust gases are directed around the intake riser by a control valve. The valve is operated by the temperature of the manifold or the velocity of the exhaust gases. It has been found that the optimum full throttle operating temperature of the fuel mixture is between 100 and 120° F. For this reason the present hot spot arrangement has not proven too satisfactorily. There is always too much heat when not wanted, and not enough when it is needed. The ideal situation would be to have just enough heat supplied to the manifold to heat up the liquid ends of the fuel -- yet leave the vapor reasonably cool. In this manner the volumetric efficiency could be kept up to a high level, and the overall cool intake mixture can add to both power and economy by the utilization of a higher compression ratio.

One motor company considered the problem of the fuel vapor condensing out of the stream instead of remaining as a gas. It has been stated before that condensate formation is a direct function of the manifold internal surface area, and that the velocity influences its movement, especially at bends. It has been suggested that the manifold be designed with a single division zone as close to the carburetor as possible. In this way there is a minimum of condensate to distribute, and any which is formed has to go to the cylinder for which it is intended, there is no choice once the flow is started. The proposal is to have the shortest possible length of manifold branch to each cylinder, this minimizes the area and tends to a reduction in the quantity of condensate. With the shortest lengths, higher flow velocities are desirable. The Hudson six-cylinder engine has an intake manifold of this type. In all cases the port areas are modified to compensate for the different lengths. Gradually increasing the velocities from the distribution zone to valve has proven slightly beneficial, but the treatment of the port adjacent to the seat has the major effect upon the gas flow. By designing the port in an approach to a true venturi, the best effect is accomplished, but this is difficult to attain. One final factor to remember is that valve timing and the manifold areas must be matched to take full advantage of the ramming effect which results from the pulsations in the intake system.

In any event one thing seems certain, namely, that a good deal more research is necessary on induction systems. Not along the well-trodden path, which is already amply charted, but in the direction of uniflowing the current in a practicable manner and cutting out the wasteful effects of, as yet, comparatively uncontrolled charge column vibrations and rebounds, and generally bad mixture distribution.

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