AN EXPERIMENTAL INVESTIGATION OF PRESSURE VARIATION, OF THE CAUSATION AND SUPPRESSION OF NOISE IN EXHAUST SYSTEMS OF INTERNAL COMBUSTION ENGINES.

BY

JOHN C. MORRISON.

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I N D E X.

INTRODUCTION.

One of the essential features of a high speed internal combustion engine is that of silent operation, and the suppression of exhaust noise is therefore of the utmost importance. Before any remedy can be applied it is obviously necessary to ascertain the fundamental conditions underlying the causes of exhaust noises, and with this in view the following series of investigations on a high speed petrol engine were undertaken.

The scheme of experiments entailed in the first instance the design and application of a special type of indicator to ascertain the pressures in the exhaust pipe at high engine speeds. After this apparatus was tested out and proved reliable a large number of experimental pressure tests were made, first on a motor car engine using only one cylinder, and then on a specially equipped single cylinder motor cycle J.A.P. engine. The investigations were carried out on various lengths of open pipe with and without silencers.

Other experiments were made on sound production by the engine, and on the effectiveness of its suppression by the various types of silencer employed.

The results of these two series of experiments are dealt with conjointly in the latter part of the discussion.

The details of the special pieces of experimental apparatus which were found to be necessary as the work progressed were all designed and partly made by the writer.

SECTION I.

PRELIMINARY INVESTIGATION OF THE EXHAUST PRESSURE CONDITIONS OBTAINING IN A STANDARD PETROL ENGINE.

As the writer was unaware of any special experimental work in this field he decided to carry out certain preliminary experiments on an existing engine using a simple apparatus. The only engine available in the Laboratory was a disused six cylinder car engine of about 3,400 c.c. capacity. The cylinders were in separate blocks of two each with one branch of the inlet manifold feeding the two cylinders. Each cylinder had a separate exhaust flange. See Drg. 1. The cylinder at the end near the starting handle was used for the test. The exhaust from the other cylinder in the same block was led in a long pipe outside the building. The other four cylinders were not used and consequently acted as a brake. An attempt was made to regulate the engine speed by means of a Prony brake on the flywheel but this was not a success. As the carburettor belonging to the engine was much too big for two cylinders a motor cycle carburettor was substituted. The top dead centre of the working piston was marked on the circumference of the flywheel and was found as follows:- A vertical piece of

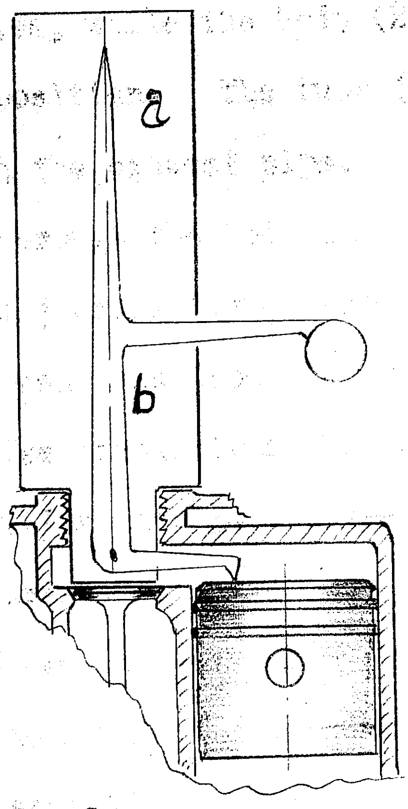


Fig. 1.

wood (a) Fig. 1. was set on the top of the valve cover port and a bell crank (b) was hinged to the bottom of (a) so that one end touched the top of the piston, and when the latter was moved over the T.D.C. the vertical arm of the bell crank magnified this motion. The exact position of the T.D.C. was thus easily obtained. The point at which the exhaust valve began to open was also marked on the flywheel. The rate of opening was found to be comparatively slow. The valve was small and the compression of the engine was low, so that the exhaust was not so noisy as that of more modern engines.

THE PRESSURE INDICATOR.

For small pressures at high speeds the ordinary piston indicator is quite unsuited on account of the inertia forces which are great in comparison with the gas pressures. The following simple device was tried out and found to be sufficiently accurate for the immediate purpose of the experiments. The name given to it is the Single-Point Indicator to distinguish it from the later design called the Multi-Point Indicator. See Fig. 2. It consists essentially of a cock, the centre or journal (1) of which is screwed on the projecting end of the engine cam shaft, and rotates, while the body (2) of the cock is held in any desired position. The tube (3) connects the centre of the cock with the exhaust pipe. Three 1/16" dia. holes are drilled through the body and journal of the cock so that while the journal turns through a small angle gas under pressure can pass from the exhaust pipe to a manometer. After a few revolutions the pressure recorded by the manometer for this particular point in the exhaust stroke becomes constant. The body of the cock can be angularly adjusted and the pressures found for a number of points in the engine cycle. A complete curve of pressures for any desired part of the engine cycle can then be drawn.

A similar scheme has been developed by other investigators for other purposes, but the writer was not aware of this when he designed the above apparatus.

A large size scale of angles (4), Fig. 2, was fixed to the front of the engine, each degree reading two degrees since the camshaft rotates at half engine speed. It was

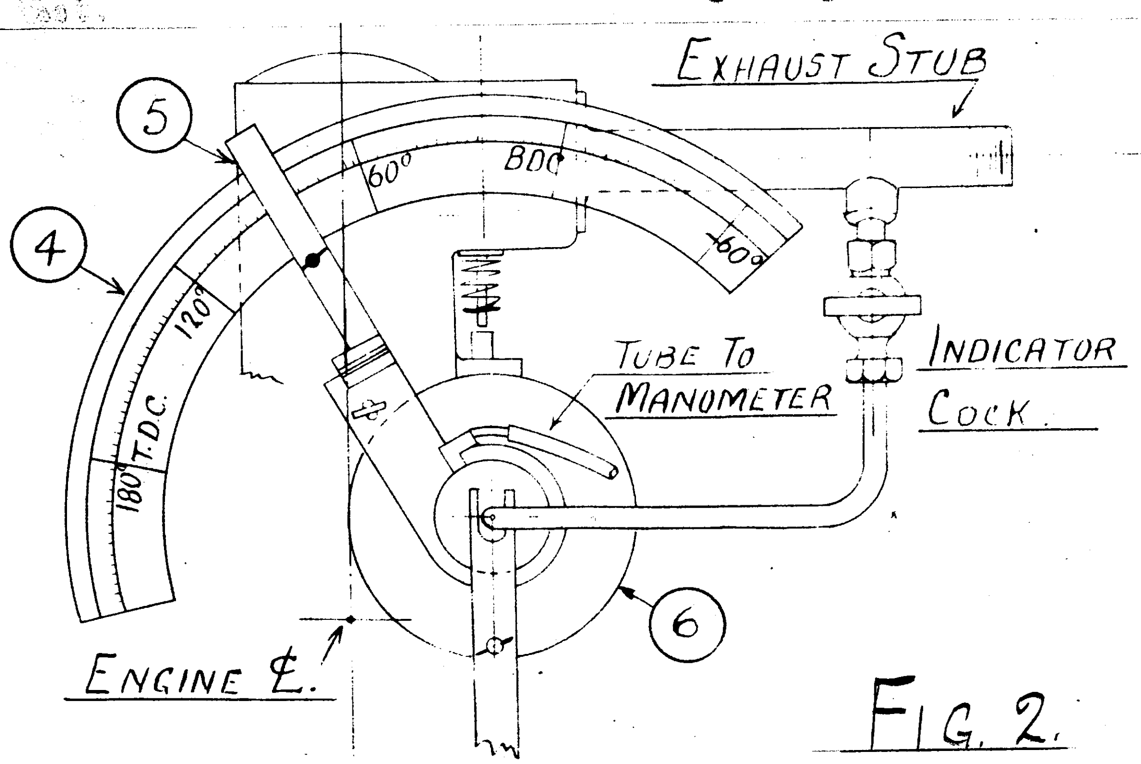


FIG. 2.

ARRGT OF INDICATOR ON ENGINE.

not possible to get a complete circle of this scale, so the arm (5), giving the position of the indicator, was made free to rotate on the indicator, and could be clamped to a disc (6) soldered to the body (2). The method was to turn the indicator by the arm through 240° on the scale (4), then to slacken the clamp and, holding the indicator in this position to put the arm back to the zero (i.e.-60°) and clamp it again. When this was done twice the engine had turned through 3 x 240° = 720° or two revolutions. This was rather a hindrance but it could not be avoided.

The exhaust system to be tested could be screwed or clamped to the end of the exhaust stub. A small pipe was led from the cooling system to drip on the indicator cock and prevent heat travelling down the copper pipe to the indicator. The first experiments with this indicator were performed for the purpose of getting a general knowledge of pressure conditions rather than as part of any complete test.

The curves of pressure are shown in sheets 1, 2, 3 and 4, and are all plotted on a base of crank angles. Since the speed of the engine is known the base also serves as a time scale.

DESCRIPTION OF EXPERIMENTS AND DEDUCTIONS FROM THE CURVES.

Curve No.1. is the result of the first attempt at using the indicator. Points were taken only 4° apart, but this was much closer than was necessary. Afterwards the practice was to take two or three points on the crest of a wave and one at zero, or rather at atmospheric, pressure.

Curves No.2. A and B obtained from the same open pipe at different speeds gave the impression that the frequency of the pressure waves was independent of the engine speed. The two curves were plotted on the same time base, see Nos. 2. A and C, and might have coincided if 2. A had been regular.

Curves Nos. 3. and 4. are taken for the full two cycles of the engine. Curve No.4. is irregular as regards pressure, but it has the same average frequency as No.3. of 50 waves per second.

Considering the exhaust pipe as an organ pipe closed at one end and open at the other the wave length is four times the pipe length and is equal to the velocity of sound divided by the frequency. For the pipe 6'-9" long having a frequency shown by the curves of 50 waves/sec. the velocity of sound in the gas = 6·75 x 4 x 50 = 1350 ft./sec. This velocity is discussed later in the section on Sound. For the present it is sufficient to conclude that the wave motion in an exhaust system is in accordance with ordinary acoustic phenomena.

The two silencers tested were the Howarth and the Carbjector. They are described on pages 46-7.

The Curves Nos. 5 and 6, Sheet 2, obtained by using them on the end of a long pipe are remarkably similar to one another, which is at first sight surprising since the construction of the two silencers is entirely different. Also the initial pressure at B. D. C. is almost the same for the 6 ft. 9 inch pipe when it is open and when a silencer is attached to the end. The reason for this was not fully understood at this time, and the explanation is held over till later, see pages ~~38-40~~ 41-2.

In Curve No.3. the first pressure wave starting from the cylinder goes to the open end of the pipe and is reflected as a negative wave, but when the silencer is attached, Curve No.5, the pressure wave is reflected from the silencer as another pressure wave which appears at 'A'. The pipe in this case behaves like an organ pipe closed at both ends although the silencer is not quite the same thing as a closed end. The silencers have a strong damping effect on the vibrations.

Curves Nos. 7 and 8, Sheet 3, taken with the Howarth silencer attached direct to the exhaust stub give curves somewhat similar to No.5. The Howarth has an ejector device which is supposed to help out the exhaust gases, and in order to see if this had much effect the silencer was tested first in the normal position and then with the ends reversed so that the gas was flowing the wrong way.

Curves Nos. 9. A and B show that there is a smaller initial pressure with the silencer reversed and that the period of the waves is longer than with the silencer in the normal position. This test was repeated with the new engine and the results are discussed later. This applies also to the next experiment in which the two silencers are used in series, Curves Nos. 10. A and B.

Curves Nos. 11, 12, and 13 are intended to show the effect of the addition of a single baffle plate to the end of an open pipe.

The baffle consisted of a disc (a), Fig. 3, of sheet steel fitted in a circular box (b) which could be slipped on the end of the exhaust pipe (c) while the engine was running. The gas passages at points 1, 2 and 3 had the same cross sectional area as that of the exhaust pipe.

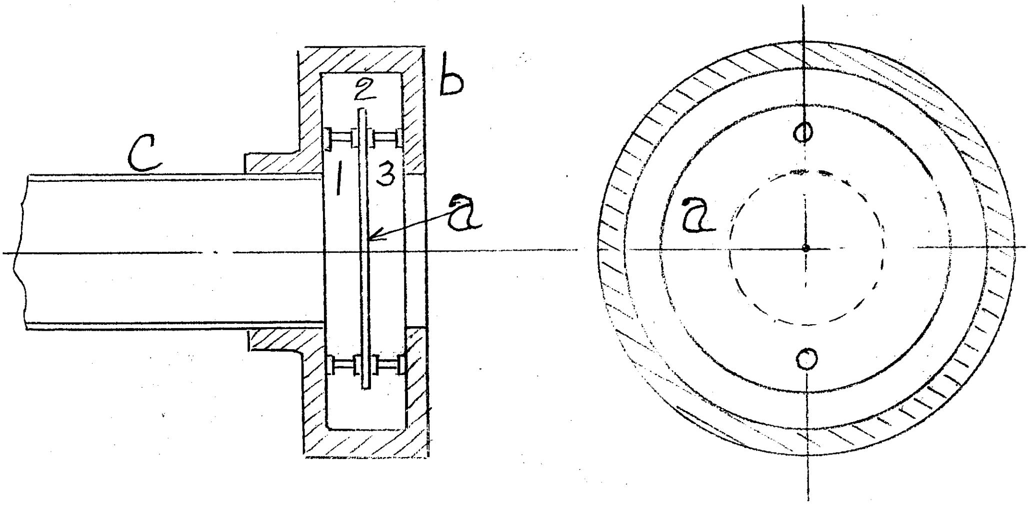


Fig. 3.

One curve was taken with the baffle in place and another without the baffle. The exhaust pipe in this case was 1 $^5/_8$" dia. and 4 ft. 9 ins. long.

The first notable feature of the curves is that the maximum pressure at B. D. C. is not increased by the presence of the baffle. This is the same phenomenon that was observed in the case of the silencer curves, and is discussed on pages 40-42. . The baffle does not influence the frequency of the waves but it has a considerable damping effect on them.

THROTTLED EXHAUST.

It is sometimes stated that throttling the exhaust under certain conditions increases the power of an engine. It was not possible to observe the power variation of this engine, but curves of exhaust pressure were taken for two degrees of throttling. A flange (a), Fig. 4, was screwed on the exhaust stub, and a flange (b) could be bolted to it. This flange (b) had a central hole the area of which for Curve No. 14. was a quarter of the area of the pipe, and for Curve No. 15. A was half the area of the pipe.

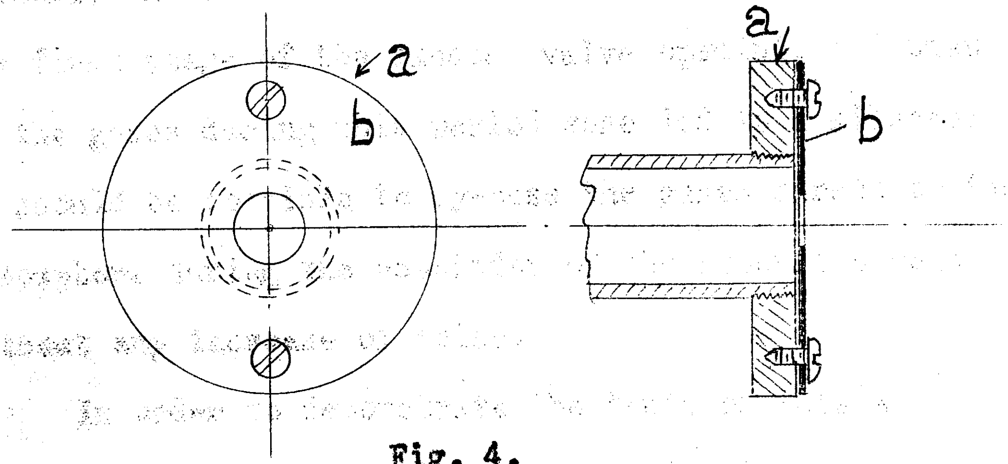


Fig. 4.

The throttling increases the initial maximum pressure and stops the wave motion, which latter effect would be beneficial in cases where the wave due to an open pipe was above atmospheric pressure at T. D. C.

EXPANDING PIPES.

Exhaust pipes have often been made funnel shaped with a view to getting a Venturi effect as in the case of the flow of water.

To test this theory two funnels were made, one having an expansion ratio of 1 to 4 and the other 1 to 8. They were tested in comparison with a parallel pipe of the same length, see Curves Nos. 16. A, B and C.

There is very little difference between the Curves B and C for the funnels. Both have a smaller initial maximum pressure than the parallel pipe. It would not seem that tapering an exhaust pipe is of any value.

An experiment performed with this engine in connection with the noise problem belongs really to the section on Sound, but it will be more convenient to deal with this experiment now.

Certain phenomena had suggested the idea that the explosive noise of the exhaust was produced only during the first stage of the exhaust valve opening, and that if the gases during this period were led to a silencer it should be possible to by-pass the gases direct to the atmosphere during the remainder of the exhaust stroke without any increase of noise.

In order to demonstrate the truth of this a special piston valve was made which directed the gases to a silencer during the first part of the exhaust stroke, and then uncovered a port allowing the gases free passage to atmosphere. The arrangement is shown diagrammatically in Fig. 5.

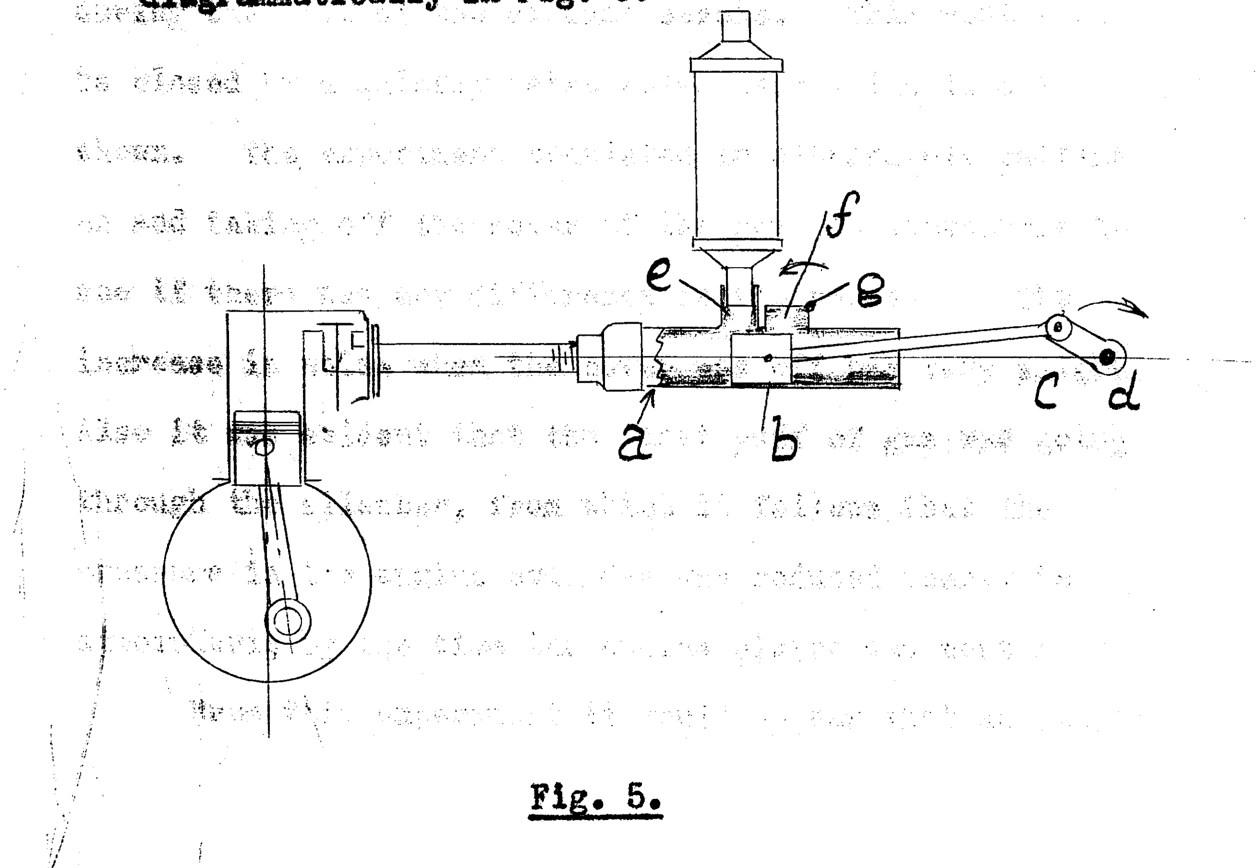


Fig. 5.

A cylinder (a) is attached to the exhaust stub and the piston (b) is driven by a crank (c) on a countershaft (d) which is connected by a chain drive to the engine and runs at engine speed. When moving out the piston uncovers first the port (e) leading to a silencer, and secondly the port (f) which may be open to the atmosphere or may be shut by a cover (g). The details of the apparatus are shown in Drg. No.3, and the arrangement on the engine in Drg. No.4.

FIRST EXPERIMENT.

The object was the locating of the noisy period of the exhaust.

The piston valve was timed in such a way that it began to open the port to the silencer at the same instant that the exhaust valve began to open. The port to the silencer was full open when the engine crank was 10° past B.D.C. Then the piston valve began to open the port to atmosphere, and this port remained open during the rest of the exhaust stroke. This port could be closed by a quickly detachable cover which is not shown. The experiment consisted in alternately putting on and taking off the cover of the port to atmosphere to see if there was any difference in the noise. The increase in noise when the cover was off was very slight. Also it was evident that the first puff of gas was going through the silencer, from which it follows that the pressure in the engine cylinder was reduced nearly to atmospheric by the time the engine piston was past B. D. C.

From this experiment it would appear that an engine may be silenced by putting the first portion of the exhaust gases through a silencer, and allowing the latter portion of the gases to go direct to the atmosphere. This would overcome any objection to a silencer on the score of back pressure, but it would necessitate a valve driven by the engine, and this would certainly be a strong objection to the scheme.

SECOND EXPERIMENT.

Pressure effects with the Piston Valve in use.

The indicator was used to obtain pressure curves for the two conditions (1) Port 'g' to atmosphere open; (2) Port 'g' closed.

Curve No.17. for condition (1) shows a considerable drop below atmospheric pressure after the port to atmosphere is open. The latter part of the curve is the same as would be obtained if there were no silencer or piston.

Curve No.18. for condition (2) is somewhat similar at first, but after the initial rise and fall in pressure there is another pressure wave reflected from the silencer. This dies out and does not vibrate as in Curve No.17.

It might be possible to arrange matters so that the drop below atmospheric pressure, as in Curve No.17, occurred when the exhaust valve was closing at T. D. C. which would give better scavenging than is possible with the air port closed. Further speculation in this matter would not be of any use as conditions in the J.A.P. engine were found to be quite different.

THIRD EXPERIMENT.

Effect of delayed opening of the Piston Valve.

This investigation was suggested by an experiment which had been performed by the writer some years before. The engine used then was a small $1\frac{1}{2}$ H.P. motor cycle engine which had a very early opening exhaust valve. The exhaust note had a very sharp "crackle" and was very suitable for testing silencing schemes. The following device was very successful :-

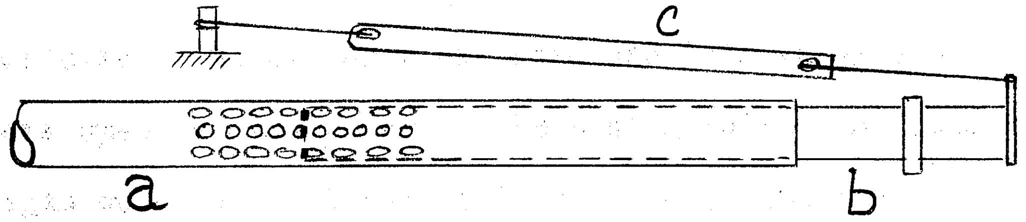


Fig. 6.

A number of holes were drilled in the exhaust pipe (a), and a very light wooden plunger (b) was forced into the pipe by a piece of elastic (c). The plunger covered the holes when full in. When the exhaust valve opened the first rush of gas pushed the piston back and uncovered the holes. This did not appear to affect the running of the engine. The "crackle" was entirely eliminated, and only a slight puff was heard. This appeared to be due to the fact that there was no opening to atmosphere at the instant when the explosive effect of the gas entering the pipe set up violent sound waves.

A similar scheme was tried on an engine of about three times the cylinder capacity of the small one, but the same result could not be reproduced.

When the noise made by a large engine had been studied for some time it became evident that while the "crackle" or explosive bang was the main feature there was another component, namely a hiss similar to that of compressed air escaping through a nozzle. It is shown in the section on Sound that this is a very important factor. In the small engine the hiss is not so marked, and its absence probably accounts for the success of the plunger method in this case.

In order to get more conclusive evidence as to the possibility of preventing the "bang" without the assistance of a silencer the piston valve apparatus was employed. No silencer was used and both ports 'e' and 'f' were open to atmosphere. The piston valve was timed to begin opening the first port when the engine piston was at B. D. C., thus preventing any connection between the exhaust gases and the atmosphere till the noisy stage was nearly past. This did eliminate most of the "bang", but the "hiss" was very pronounced. The "hiss" appeared to be made at the piston valve openings, so it would be necessary to design a special form of piston or valve which would allow a quieter exit of gas.

On the whole it did not seem likely from this experiment that it would be possible to get a really quiet exhaust without some type of silencer.

A curve of pressure No.19. was also taken during the above experiment.

The initial rise in pressure is not followed by a drop below atmospheric as in most other cases. There is a gradual decrease in pressure accompanied by a wave motion but all above atmospheric. The scheme, therefore, does not tend to improve scavenging.

No further use was made of the Piston Valve

Apparatus as it was felt that experiments on similar lines might be carried out with more profit on the J.A.P. engine. This concludes the record of the work carried out on the car engine.

CONCLUSIONS DRAWN FROM WORK ON THE CAR ENGINE.

1. A pressure detecting device such as the Single Point Indicator is suitable for use on an exhaust system of an engine running at speeds of 2500 r.p.m., and it would probably be satisfactory at much higher speeds.

2. The pressures existing in a long exhaust pipe are almost entirely due to wave motion caused by the intermittent nature of the discharge into the pipe.

The frequency of this wave motion is a function of the length of the pipe and of the velocity of sound in the gas, and is independent of the engine speed.

3. In the case of a long exhaust pipe the initial maximum pressure in the pipe near the exhaust valve is not affected by the addition of a silencer or baffling device at the end of the pipe. Placed in such a position silencers of the two particular types used appear to act simply as wave reflectors. They allow the gases a fairly free passage, and consequently their silencing effect lies in their ability to reflect sound waves rather than to allow a gradual expansion which is the claim made for them.

4. Expanding pipes appear to have no effect either in reducing noise or in producing a suction.

5. The nature of the exhaust noise is complicated but can be resolved into two main elements: (a) a "bang" or "impact" noise due to sudden rise of pressure in the pipe when the exhaust valve is opened, (b) a "hiss" produced by the gases rushing at a high velocity through the valve passages.

It has not been possible to estimate the relative importance of these two components. In the case of small high speed engines the "hiss" is not so noticeable as in the case of larger engines where the period of valve opening is longer.

6. A mechanically operated valve to direct the gases into a silencer during the first part of the exhaust stroke and afterwards to bye-pass them directly to the atmosphere may be a possible solution. It has the disadvantage, however, of requiring moving parts.

7. All manufacturers of Commercial Silencers claim that these silencers give no back pressure. This claim is based on the fact that in certain tests the addition of a silencer to an open pipe did not result in a loss of power or in a higher average pressure in the exhaust pipe. The curves taken with the writer's Single Point Indicator show that an average back pressure gives no definite information about the actual variation of pressure during the exhaust stroke. The addition of a silencer may in some cases decrease the pressure at the important point in the cycle, viz., the T. D. C., and may thus give the impression that the silencer is having an ejector effect. The phenomenon appears to be

due not to any ejector action but to the timing of the pressure wave which is reflected from the silencer.

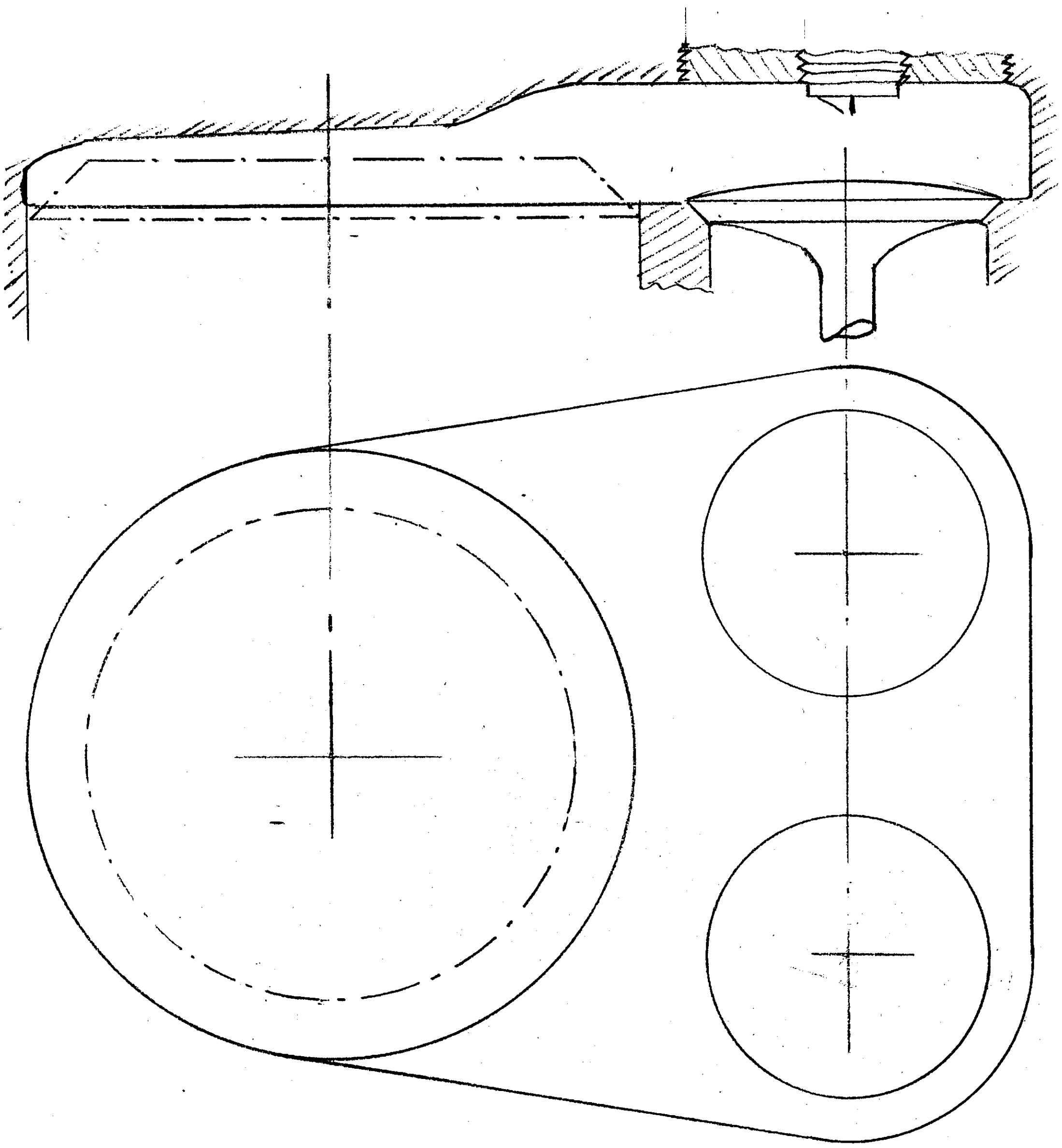


FIG. 7.

SHOWING POSITION OF PISTON
WHEN LONG CON. ROD. WAS FITTED.

SECTION II.

WORK CARRIED OUT ON THE J.A.P. ENGINE.

SELECTION OF NEW ENGINE.

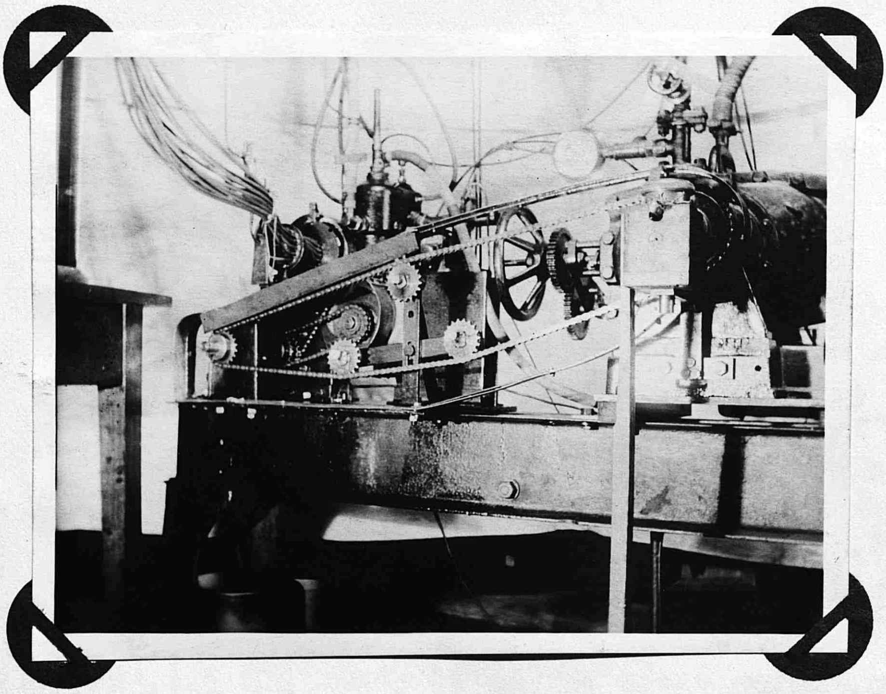
In order to bring in all the factors of noise, pressure, power and speed the most suitable type of engine was a single cylinder overhead valve, high compression, motor cycle engine of about 500 c.c. capacity. But all engines of this type are air cooled, and watercooling was absolutely necessary. A compromise was effected by using a watercooled side valve cylinder which belonged to a twin-cylinder engine, with the crank-case of a single cylinder air cooled engine.

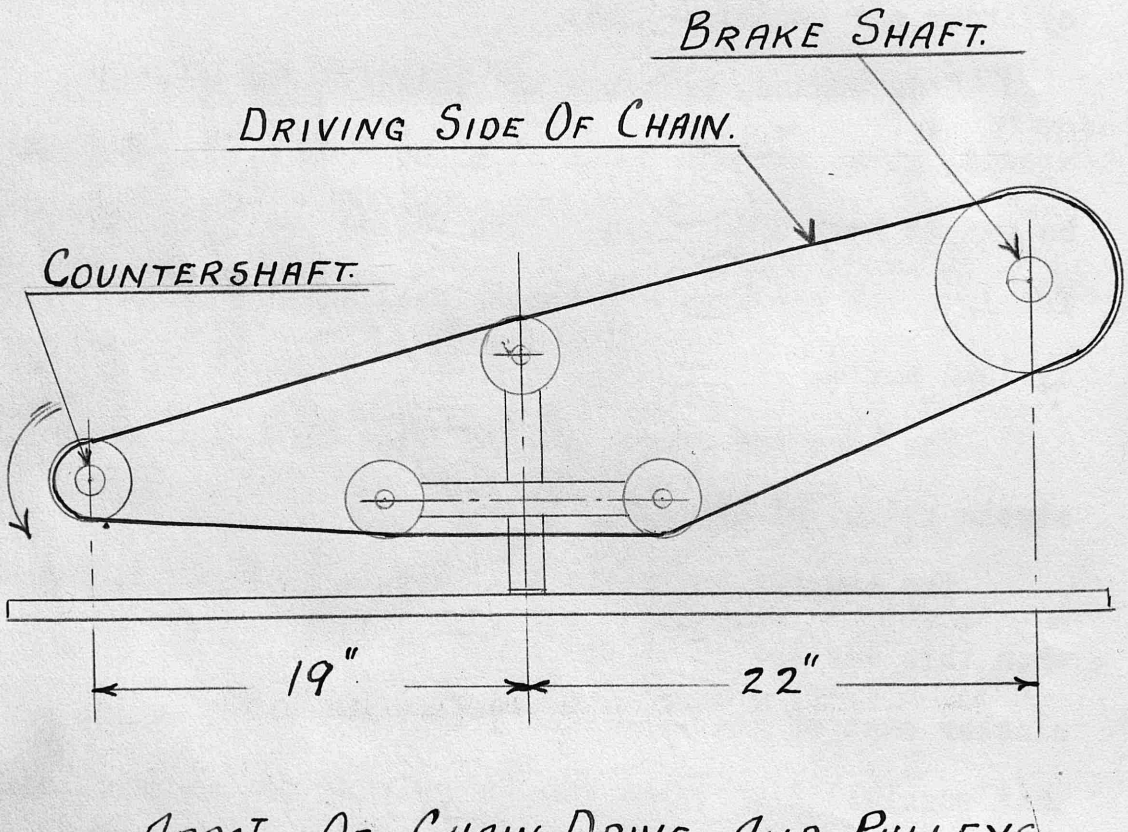
This engine, although not intended for very high speeds, gives maximum power at about 3200 r.p.m. and can be run to over 4000 r.p.m. The valves are large and fairly quick opening, and the exhaust makes a comparatively loud noise.

The bore and stroke are 85·7 and 85 m.m., giving a stroke volume of 490 c.c.

The compression ratio was said to be 5½ to 1, but when this was tested it was found to be 4½ to 1. A plaster cast of the clearance volume was taken to see what possibilities there were of raising the compression. The engine was returned to Messrs J.A. Prestwich who fitted a longer con. rod; see Fig. 7., which, they said, gave a compression ratio of 5·7 to 1. It was found to be 5¼ to 1 which was high enough.

A Heenan and Froude Hydraulic Brake was used for





ARRGᵀ OF CHAIN DRIVE AND PULLEYS.

FIG. 8.

measuring the power. This brake was intended for powers up to 100 H.P., and very fine readings at the power of this engine (about 10 H.P.) were not expected, but as will be mentioned later, the brake proved to be quite sensitive. An examination of the Power Curve of the brake showed that a reduction of speed of 2 to 1 between the engine and the brake would be required. This reduction was taken advantage of in designing the starting device.

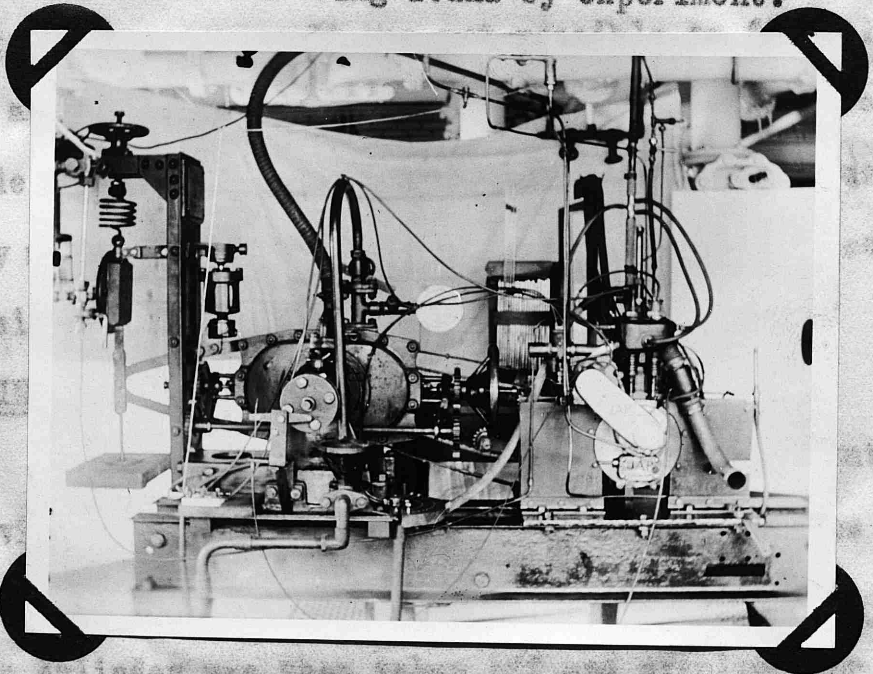
COUNTERSHAFT.

In order to facilitate driving the Indicator and any other apparatus that might be found necessary, a countershaft running at engine speed was interposed between the engine and brake. See Drawing No.5. It is carried in Skefko Self-aligning Ball Bearings the housings of which can be moved horizontally for purposes of chain alignment. For various reasons the countershaft had to be in front of the engine and the brake behind, giving an inconveniently long chain drive. Owing to the jerky nature of power delivery it was evident that a jockey pulley would have to be fitted about the middle of the drive.

Various arrangements of pulleys were tried, and even the 3 pulleys as shown, see Fig. 8, allows a good deal of whip in the chain at low speeds. The only way to keep the chain from jumping the pulleys is to have it pretty tight. Plain pulleys were used at first but these were very noisy and wore quickly and were replaced by chain sprockets.

STARTING DEVICE. See Drawing No.6.

This type of engine is usually started by a kick starter on the gearbox. This arrangement not being possible here the most convenient one was a hand starter on the brake spindle. A blind flange (1) with a centre catch stud (2) was bolted to the brake shaft coupling flange (3). The starting handle clutch end (4), was slipped over this stud, and it simply dropped off when the engine started. As it is not possible to pull an engine of this size over compression with a full charge of gas an exhaust valve lifter worked by a Bowden wire is supplied with the engine. It is difficult to work this by hand and drop it at the right moment, so a cam (5) was bolted to the starting handle clutch which operated a lever (6), and pulled the Bowden wire, the best duration of the opening of the exhaust valve being found by experiment.



ARRANGEMENT OF ENGINE & BRAKE.

Some difficulty was at first experienced in starting the engine owing to the B. & B. carburetter not giving a rich mixture at low speeds. A special jet was made by altering the existing slow running jet so that when it is fully opened a proper spray mixture is inhaled by the engine. As soon as the engine fires this jet is shut down, and the carburettor functions normally. This arrangement works very well.

LUBRICATION.

There was no oil pump on the engine, so a gravity feed was arranged with the tank suspended from the roof. A drip sight feed was used to regulate the supply.

PISTON POSITION AND CRANK SHAFT ANGLE INDICATOR.

There was no outside flywheel or other convenient rotating part for marking the angles through which the engine turned.

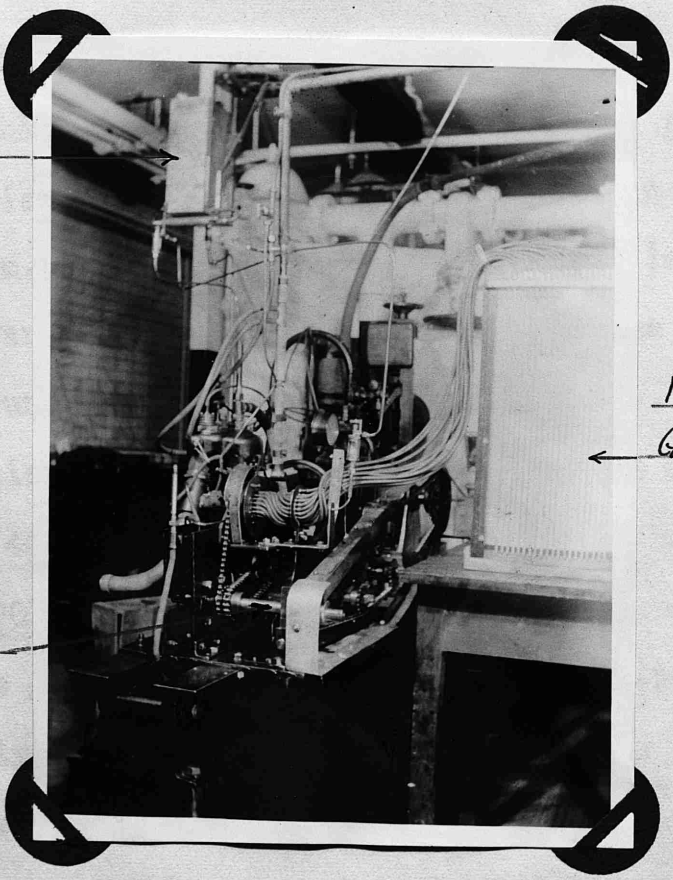
A pointer (1), see Drawing No.7, was jammed between the driving sprocket and the flange against which the sprocket screws. It was not possible to fix this exactly in any position, so the disc (2) marked in degrees, was made adjustable, and when the Top Dead Centre was found (by method, p. 2) the disc was turned to the correct position and clamped there.

RUNNING IN OF THE ENGINE.

The piston and bearings were not tight, but it was advisable to run the engine easily for some time before putting it on full load at top speed. This was done, and the cylinder was then taken off and examined for high spots. Neither the piston nor the cylinder required touching up, and the rings had taken up a good polish. The valves also bedded down well, and the compression was and is perfect.

COOLING DIFFICULTIES.

After periods of running on full throttle the


OIL TANK.
MERCURY GAUGES.
COUNTER-SHAFT.

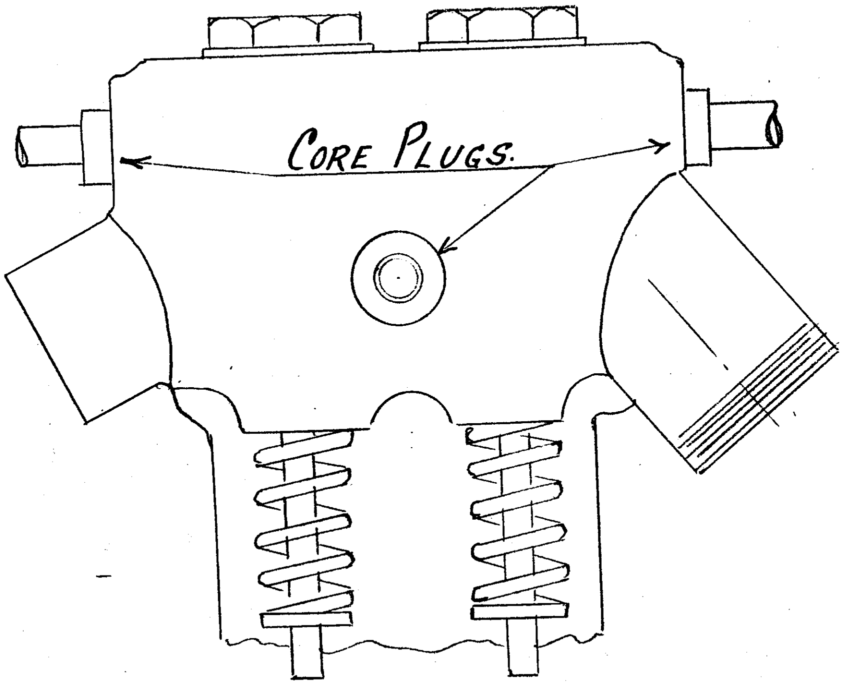
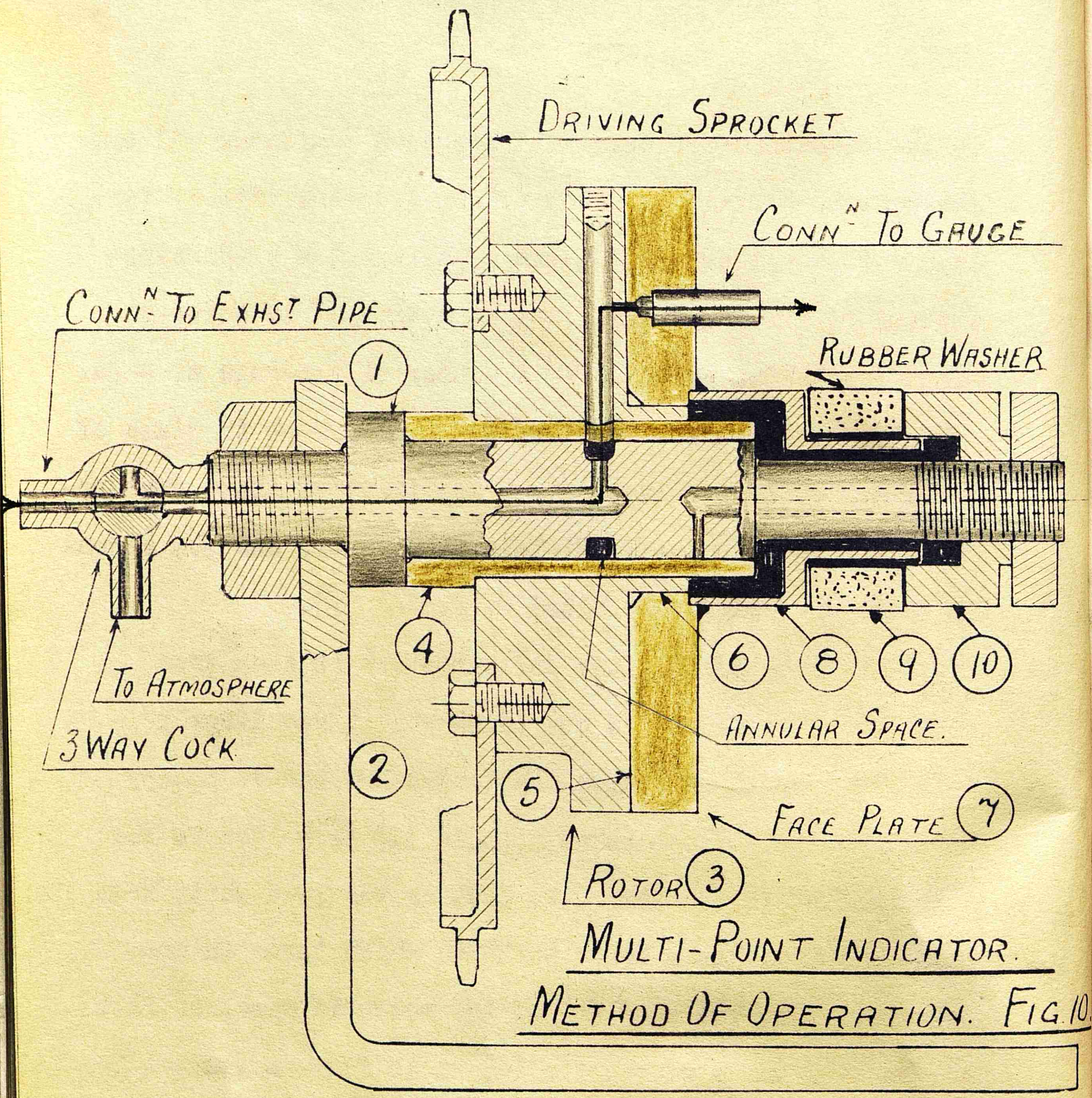


FIG. 9.

engine would begin to run irregularly and sometimes stop or else settle down to a slow speed. This was puzzling as there was no knock, but on one occasion it was noticed that the exhaust was cool which showed that preignition was taking place. This was proved by switching off the magneto and the engine continued to run. A better plug was tried with no improvement. The flow of cooling water was increased till the body of the engine was quite cold, but the part round the valves remained hot, and steam was being formed. There was nothing to induce a quicker flow in this part, and after trying various expedients the core plugs were bored out and connected to the cold water supply, see Fig. 9, while the standard inlet connection was throttled. The exhaust valve cap was also watercooled, but the plug still got very hot and preignition continued. Finally a very cool plug - a Lodge Aero - was tried, and this cured the preignition. The cylinder head is very bad for high compression being very shallow and extensive, see previous Fig. 7, p. 17. The ignition could not be advanced more than 15° at 1000 r.p.m., and 20° at 3000 r.p.m., which is an unusually small advance, and probably accounts for the drop in power occurring soon after 3000 revolutions. An engine of this size with big valves should develop maximum power at about 4000 r.p.m.

THE MULTI-POINT INDICATOR.

The great disadvantage of the Single-Point Indicator previously described was the length of time required to take a full set of readings for two revolutions of the engine. The success of this type of indicator depends absolutely



MULTI-POINT INDICATOR.

METHOD OF OPERATION. FIG. 10.

on the conditions of speed and pressure remaining uniform. Consequently it is necessary to cut down the time as far as possible without making the apparatus too elaborate. Instead of getting one reading at a time it is quite a simple matter to arrange for a number of readings at once. Also the readings can be marked off on a graph in place of being copied into a logsheet. The first of these improvements is found in the Multi-Point Indicator, and the second in the Gauge Frame (described on p. 28).

The main idea of the Indicator can be seen from Fig. 10. The design is not ideal but it was largely influenced by the position the instrument had to occupy on the engine. The centre spindle (1) is bolted to the main bracket (2). The rotor (3), driven by a chain from the counter shaft, has a brass bush (4) which turns on the spindle. The end face (5) of the rotor is machined flat and smooth, and a projection (6) on it forms a journal for the face plate (7) which is of brass. the face plate is pressed against the rotor by a thrust ring (8) bearing on a rubber washer (9), the pressure on which may be varied by screwing a nut (10) along the spindle. The face plate can be held in any desired position by a stop (not shown here). Gas under pressure from the exhaust pipe comes down the centre of the spindle into a groove round the spindle and through a passage in the rotor. From this passage there is a small hole, 5/64" dia., to the rotor face. There is a corresponding hole in the face plate, so that when the two small holes coincide the gas pressure is transmitted through the face plate to a pressure gauge.

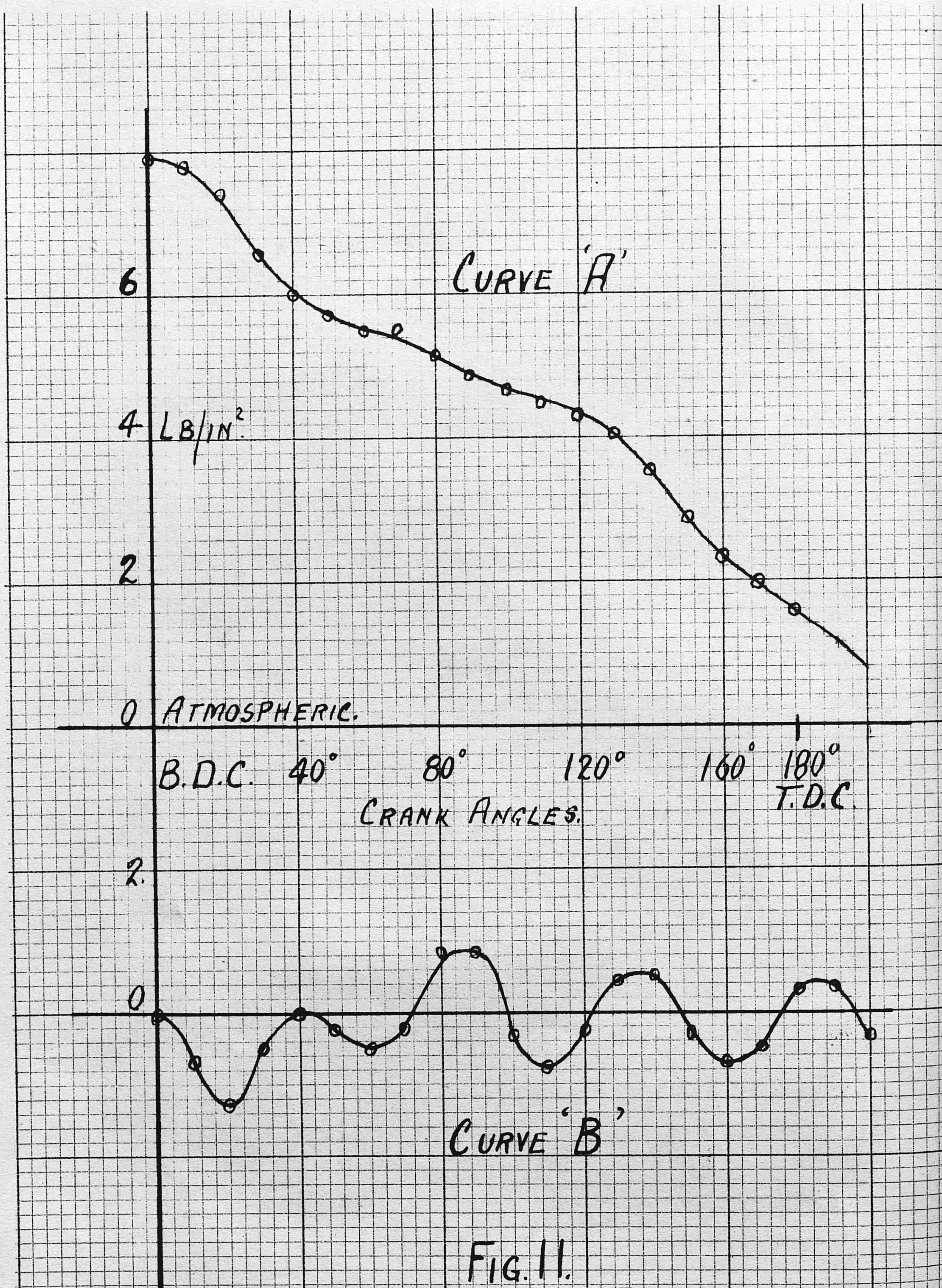

6
4 LB/IN²
2
0 ATMOSPHERIC.
CURVE 'A'
B.D.C. 40° 80° 120° 160° 180°
T.D.C.
CRANK ANGLES.
2.
0
CURVE 'B'

FIG. 11.

Provided that the gas pressure is the same each time the holes coincide the pressure in the gauge will soon build up till it is the same as that in the exhaust pipe.

A number of holes can thus be bored in the face plate giving pressures at different crank angles.

The rotor must run at half engine speed for a 4 stroke engine.

The Indicator as used is fully detailed in Drawing No.8, and Drawing No.9.

The first question is - how many degrees of engine revolution should there be between each point at which the pressure is taken? This depends on the rapidity of pressure variation, e.g., in curve 'A', Fig. 11, where the variation is slow and the small variations do not matter much a point every 40° would be sufficient, while in curve 'B' where the waves have a high frequency they could not be drawn with proper regularity with less than a point every 10°.

In almost all cases the spacing adopted was 10°. The original intention was to have two rows of holes (staggered) half way round the plate 10° apart, giving readings 20° apart for one revolution of the engine. The face plate was then to be turned through 180° to give readings for the second revolution. But when the Indicator was tested the outer row of holes did not give true readings - as described in 'Testing of Indicator' - p.22., and the inner row was extended all round to give readings every 40° of engine revolution for the two revolutions. The intermediate readings at 10° are got by advancing successively the face plate three times through 5°.

LUBRICATION.

Gas tightness of the bearings - journal and face plate - depends on the presence of a good oil film, and at the same time there must not be too much oil because the gas passages are easily clogged. The method of introducing the oil is shown in the sectioned sketch of the spindle, Drawing No.8. The oil passing through the collar of the spindle is intended for the left bearing only. The oil coming down the centre from the right spills from the end of the spindle bearing into the thrust ring from which it passes through the face plate journal to the rotor and face plate thrust face. Oil can also go down the gas passage to the latter bearing.

At first the oil supply was by gravity feed from two sight feed regulators. This did not prove sufficiently regular, and a hand operated spring loaded pump was fixed on the oil tank, and this did much better.

TESTING OF INDICATOR.

The Indicator was tested before any of the auxiliary fittings were made. The bracket was screwed down on a board, and the chain was driven by a sprocket on a countershaft, see Drawing No.9. The bearings and brackets for the countershaft were those used for the Piston Valve drive, Drawing No.3. The countershaft pulley was driven by a light belt from the pulley of a small electric motor, the speed of which could be easily varied. Before testing was begun the Indicator was run for a time to run in the bearings. The rotor face which was of mild steel, would not smooth up, so it was

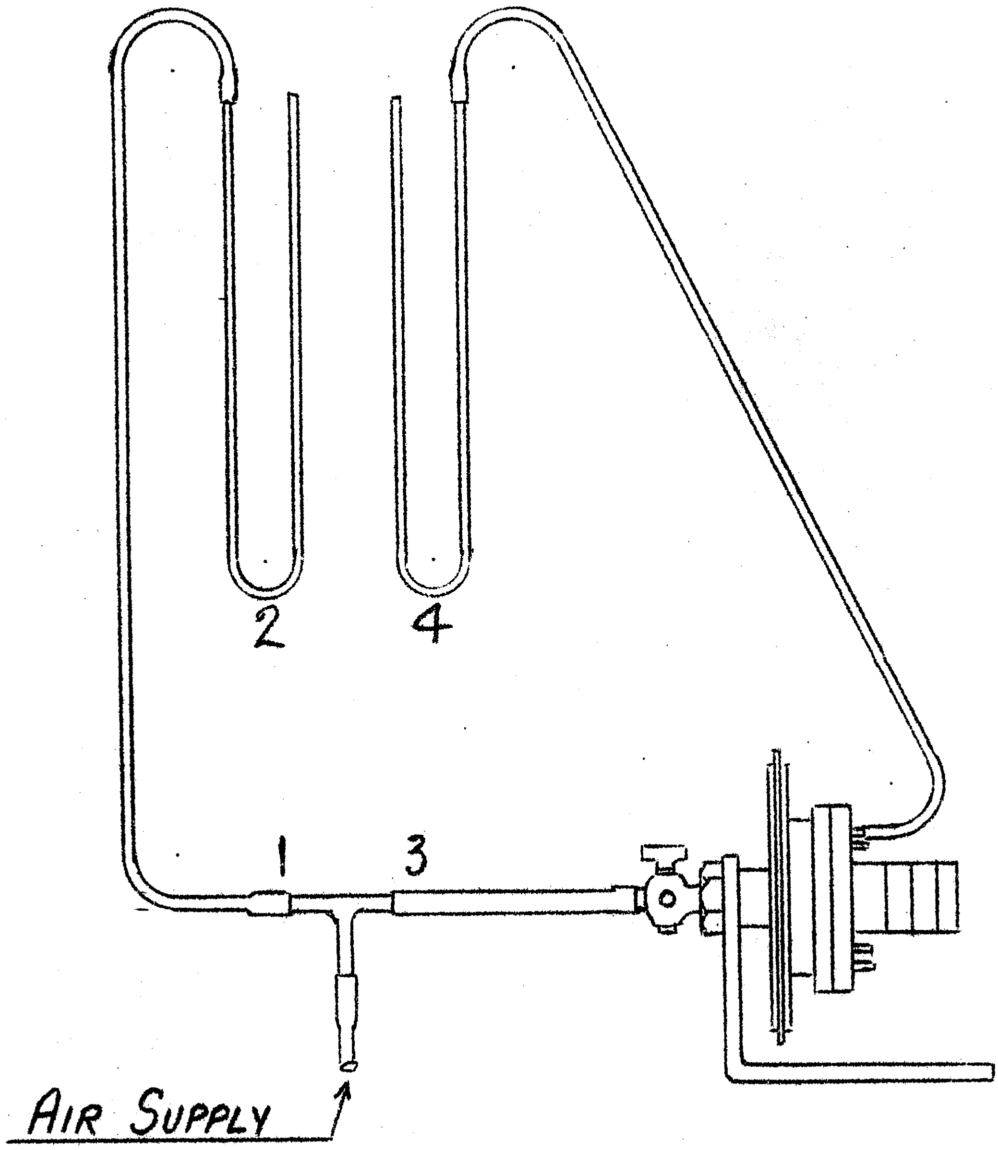


FIG. 12.

casehardened and ground.

Air at any desired pressure was got from the air reservoir of the air compressor in the Laboratory.

From the branch (1), of a Tee piece, Fig. 12, air went direct to a manometer (2), and from (3) it went to the inlet pipe of the indicator, then from any of the outlet pipes on the face plate to another manometer (4).

When the indicator was running the pressure in the two manometers should have been the same.

At first the pressure was very slow at coming through the indicator. This was due to oil sticking in the small holes in the face plate. These holes were $^{5}/_{64}"$ dia. and about $^{5}/_{32}"$ deep, and the capillary action of the oil was too strong for gas at low pressures. The holes were bored out till they were less than $^{1}/_{16}"$ deep. This was much better. The inner row of holes gave accurate pressures, but the outer row gave pressures too low, and it was evident that gas was leaking towards the circumference.

It was evident that the face plate had been badly turned and was slightly open at the circumference, but there was no hope of improving this by more turning so the only thing to do was to grind the faces with emery powder. Although this improved matters the outer row would not retain full pressure and was given up as unreliable.

Another face plate could have been made but it meant a great deal of work spacing and centering and drilling the small holes correctly as they had to fit clearance as it had to be gas tight, so provision was made to prevent a general wreck in case it did seize.

The chain wheel was not attached directly to the rotor but could rotate on the rotor spigot between the rotor and an adjusting plate. Two small brass pins were screwed through the adjusting plate into the chain wheel so that if the spindle (1) seized these pins would shear and the chain wheel could go on turning without doing any harm. <u>See Drawing No. 8.</u> The spindle actually did have a slight seizure but not enough to shear the pins.

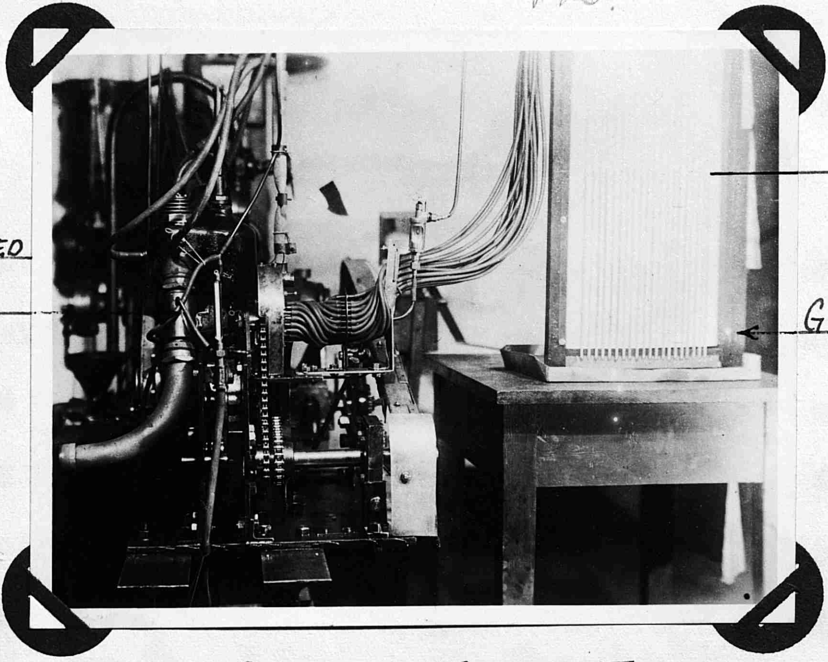
The spindle was afterwards casehardened and ground since when there has been no trouble. If the face plate seized it had only to bend the small stop for fixing its position, tear off the gauge tubes, and then it could run free. Fortunately this did not happen.

<u>OFFTAKE FROM EXHAUST PIPE TO INDICATOR.</u> <u>See Drawing No. 9.</u>

A short piece of the exhaust pipe close to the exhaust stub was watercooled with the double purpose of giving a cool offtake to the Indicator and of showing any difference in heating of the pipe due to the various silencers to be tested. The water from the jacket was led past a thermometer pocket.* A rubber tube connected the offtake with a three way cock screwed into the indicator spindle. This rubber wasted quickly with the heat so a cloth was wrapped round it and kept wet by a drip of water.

Another offtake for use at other points on the pipe was not watercooled (see same Drawing at the foot) but was simply clamped to the pipe with asbestos insertion

* There was also a thermometer pocket on the top of the engine jacket outlet.


Mercury Level
Water-Cooled Off-Take
Gauge Frame

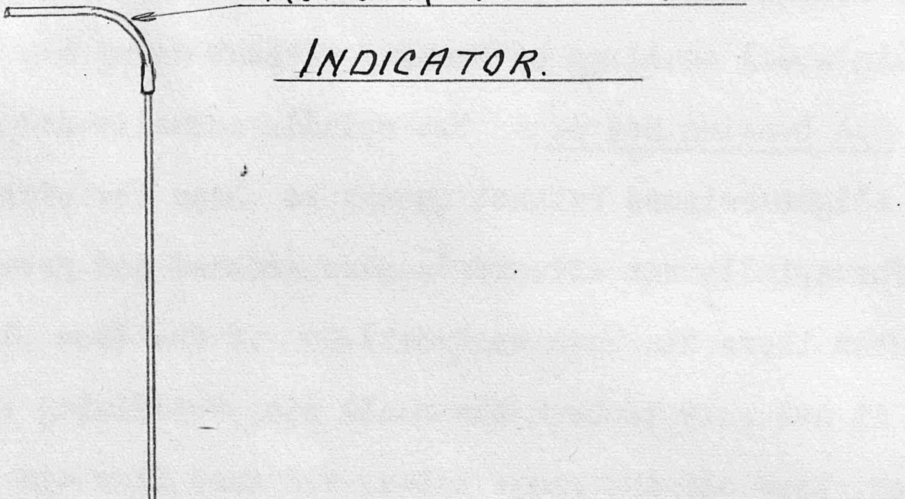

Rubber Tube To Indicator.
Mercury Bath.

Fig. 13.

to retard heat flow. The rubber connecting pipe melted during a single test, but as few tests of this kind were made it did not matter.

GAUGE FRAME. See Drawing No. 9.

The pressures obtained with the single-point indicator on the car engine were in the neighbourhood of 1 or 2 lb./ in.^{2}. A preliminary run of the new engine and multi-point indicator gave up to 8 inches of mercury - approximately 4 lb./ in.^{2}. Mercury therefore was the most suitable liquid to use, and a height of 13" pressure and 12" vacuum was allowed for. The 18 manometers were placed side by side in such a way that when a piece of tracing paper was placed in front of the tubes and a light behind the height of the mercury in each tube was clearly defined on the tracing paper where it could be marked in pencil. One limb only of the manometer was noted, the other being led to a mercury bath which was at the side and out of the way of the beam of light. See Fig. 13. As all the tubes lead from the same bath the height of mercury in each tube should be the same for atmospheric pressure. When the indicator was running with the three way cock turned to atmosphere the level in each tube was almost the same, but a zero line was always drawn before taking a set of readings. Also there was an extra tube on each side of the row of working tubes, leading from the bath and open to atmosphere, and this gave a check on the zero reading.

The tubes were $^1/_8$" bore and held about $^1/_4$ lb. of mercury each so that the net amount of mercury in the bath and tubes was a big item. In order to minimise the danger of losing all the mercury in the bath due to the breaking of a tube the ends of the tubes were led up and bent down in syphon form into the bath, so that there were no holes in the bottom of the bath.

This brought an unexpected difficulty - an air bubble formed in the top of the bend of the tube and no amount of swinging the mercury in the tube by means of an air pump would drive it out. Some tubes, for no apparent reason, were not affected this way. It was seen that the bore of the tube was so large that the mercury could pour past the bubble, so the offending tubes were reduced in diameter by heating and drawing and this cured the trouble. As only some of the tubes were affected this way it was evident that this bore was about the critical diameter for this phenomenon and that a diameter slightly less might be on the safe side. Several diameters (of bore) were tried and it was found that a reduction of about $^{1}/_{64}$" or less prevented bubble formation. The mercury sometimes deposited an opaque film in the tubes, and as they could not be taken out and cleaned a special ramrod was made. It is very risky to tie packing on a rod to go inside a thin glass tube $^{1}/_{8}$" bore and 2 ft. long, because there is little likelihood of getting it out if it sticks. The following trick was successful :- A long thin wire was bent double at one end to grip the end of a narrow tape. This entered the tube easily. When the wire was pulled up the tape packed up and filled the bore of the tube and would have stuck but the free end was pulled which released the packing till it was just tight enough to clean the tube.

ALTERATION TO MANOMETER SYSTEM.

A preliminary trial had given 8" of mercury so it was thought that an allowance of 13" should be ample. When the engine was properly tuned up, however, the pressure in the tubes near Bottom Dead Centre was so great that the mercury was blown out, and the suction also raised the level beyond the height allowed for. It was not possible to increase the height of the tubes without remaking the whole gauge frame and tube system so the tubes which had to take the high pressures were disconnected from the bath and the rear limbs made the same height as those on which the readings were taken, the ends being left open thus making them ordinary U tubes in which the change in level of one limb is half the total change in height.

The readings in these tubes had therefore double the value of those connected to the bath and this spoilt the appearance of the curves when readings were being taken, although it made no difference when they were plotted to the correct scales.

TUBE HOLDER. See Drawing No. 9.

The tubes were liable to be wasted by oil where they joined the Indicator, and in the event of the face plate seizing the tubes would wind up and break all the glass tubes in the gauge frame. To prevent this a tube-holder was attached to the Indicator bracket. The tubes from the Face Plate connections go in spiral directions to a disc carrying short lengths of copper tube. The long tubes from the Gauge Frame are attached to the copper tubes on the off side of the disc. This disc can be rotated but is normally clamped in position. The short pieces of rubber tube can be easily replaced when damaged, and the long tubes are comparatively safe. It would really have been better to have had the centre spindle in the vertical position so that oil getting into the rubber tubes would run down and out again, but this would have necessitated bevel gearing and other complications.

METHOD OF TAKING A DIAGRAM.

The engine was run till conditions were steady at the desired speed.

The Indicator Cock was opened to atmosphere and the zero line drawn on the tracing paper which had vertical lines corresponding to the manometer tubes.

The Indicator Cock was opened to the exhaust pipe and time was allowed for the mercury to rise or fall in the tubes. This time depended on the distance the mercury had to go and was on an average about 5 seconds.

The first gauge readings were marked with an o. The face plate of the Indicator was then turned through 5 ~~10~~° and the second set of readings marked with two strokes thus = . The third set were marked L, and the fourth with a cross + .

These marks were adopted because they were quicker than writing 1. 2. 3. 4., and they could not be confused. Finally the face plate was returned to the first position to see if the pressures had altered. A mirror was placed at the back of the speedometer so that it could be watched while the operator was taking the readings. After getting accustomed to the noise of the engine it was quite easy to spot even a very slight change in speed without looking at the speedometer.

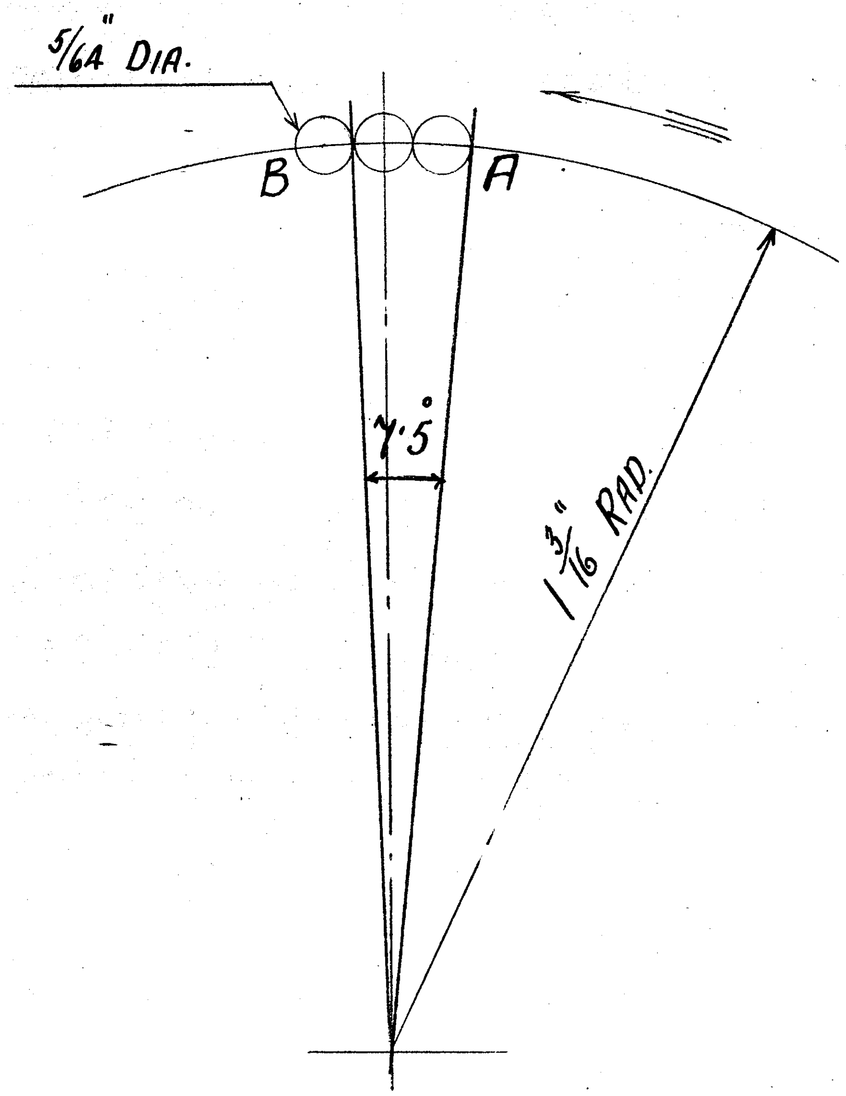
The pressures were afterwards scaled off the tracing paper and transferred to graph paper. They have been further reduced in scale in the collected curves.

The net time required for taking an "Indicator Diagram" varied from two minutes for a very wavy curve to one minute for the curves from some of the silencers.

The curves are discussed in Section III.

ACCURACY OF READINGS.

A point on the pressure-time curve is fixed by the pressure recorded by the gauge and by the crank angle position. Taking these factors separately and beginning with the latter, the accuracy of the crank angle position depends on the correct angular position of the holes in the face plate, the duration of opening of these holes, the setting of the crank angle indicator plate, the setting of the driving sprocket on the Rotor, and the slack in the driving chains from engine to countershaft and from countershaft to rotor. The angular position of the small holes in the face plate is correct to within $\pm 1/4$ degree. The finding of the Top Dead Centre for the setting of the crank angle indicator plate has already been described. The error might be $\pm 1/2$ degree, and if it is actually a little out it will give a constant error which will not affect the curve. The opening of the face plate holes relative to the crank angles was fixed as follows :- With the engine at B.D.C. the Indicator was turned till the hole in the face plate for the B.D.C. Manometer connection registered approximately with the hole in the rotor, the adjusting plate of the driving sprocket meantime being slack. The Indicator Cock was opened to atmosphere and



SCALE :- 4 TIMES FULL SIZE

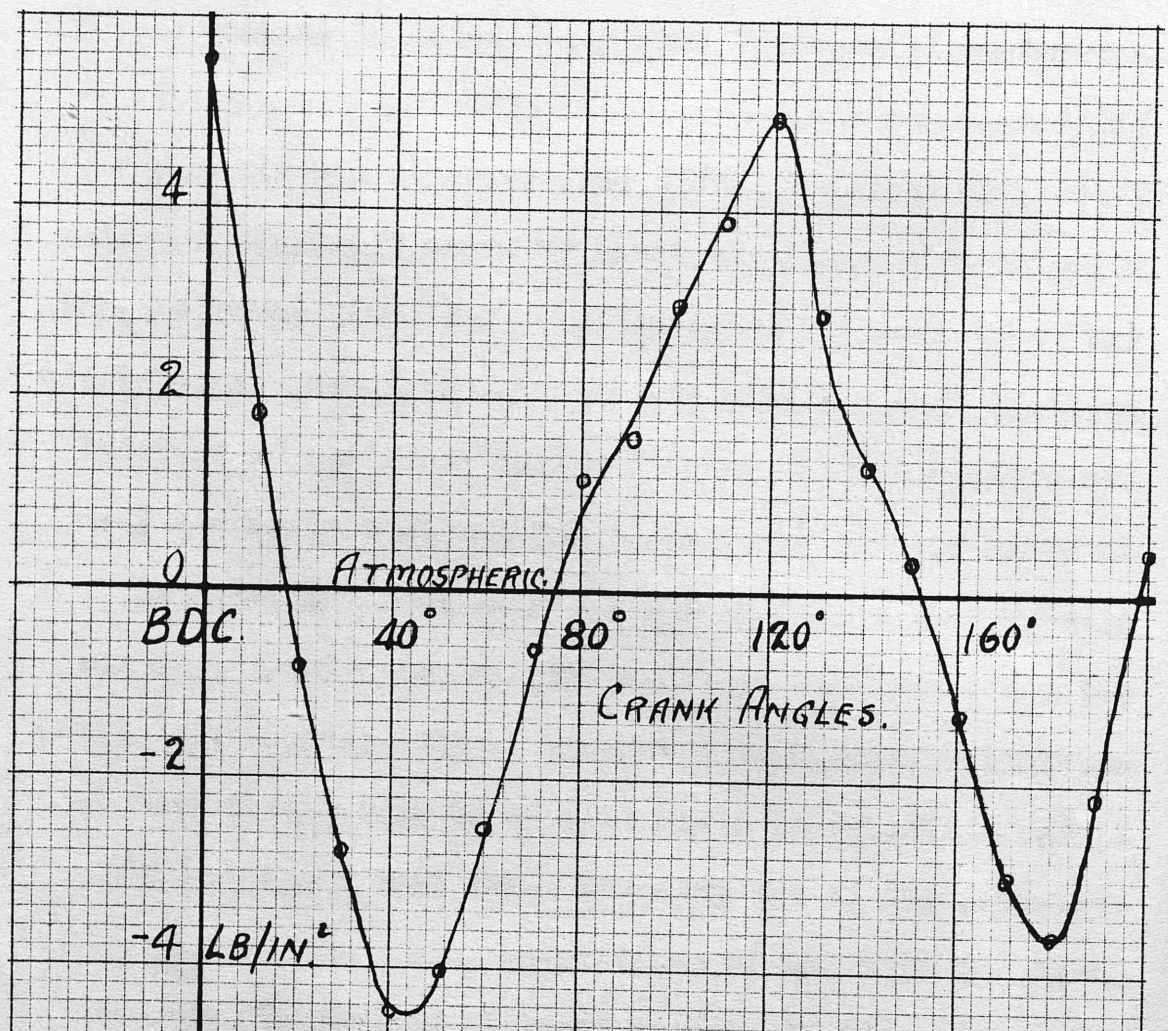
CONSIDER THE CENTRE HOLE TO REMAIN IN POSITION WHILE THE MOVING HOLE PASSES OVER IT FROM POSITION A TO POSITION B.

FIG. 14.

the rubber tube of the face plate connection was held in the operator's mouth so that when the rotor and face plate holes coincided he could blow air through the pipe. From a large scale drawing, see Fig. 14, it can be seen that the holes open and close completely while the faces rotate through an angle of 7·5°. But the gas passages when the holes are 1/3 open or shut are very small indeed, and when tested by blowing it was evident that only over 4° of rotor movement (i.e. 8° of engine revolution) is the opening effective.

To fix the position of the sprocket on the rotor the face plate was held in the B.D.C. position by the stop, and the rotor was moved from the opening to the shutting position, as observed by blowing, till what appeared to be the middle position was found. The sprocket was now tightened up and the engine rotated over B.D.C., and the angles at which the "blow through" began and stopped were observed. If these were not correct the previous operation was repeated. A few of the other positions were also checked, and the sprocket was finally tightened up and marked so that any slip could be seen without the necessity for making a complete investigation. The backlash in the chains made less difference than might have been expected, and a slight allowance was made for the tightening of the driving side when setting the driving sprocket as described above. It is difficult to tell what the effect of the jerk in the chain would be, but probably not more than $\pm 1^1/_2{}^{\circ}$.

For the purposes of the experiments the accuracy of the position of any point need not be very great and if it



CURVE SHOWING VERY RAPID CHANGE OF PRESSURE.

FIG. 15.

is within $\pm 5^{\circ}$ of the true position that should be quite good enough. If Curve No.37, sheet 10, be taken as a sample it is clear from the uniformity of the waves that the points have only a very small error of angular position.

THE ACCURACY OF THE PRESSURE registered is not directly ascertainable. It was found by using the testing device (p. 24) that the indicator gave uniformly correct readings when a static pressure was employed, consequently the pressures which reach the face plate may be supposed to be accurately recorded. But the face plate has the disadvantage of being at a distance from the point at which the pressure is taken off, actually 6½". As the connecting passage is a reasonable size (about $^{3}/_{16}$" bore) it is unlikely that there will be an appreciable loss in pressure, and as the pressure waves have the speed of sound the time lag is negligible.

As mentioned already there is no fear of error due to leak as that can only be very sudden and pronounced and consequently easily seen.

The Indicator as a whole can therefore be considered capable of a high degree of accuracy, although it certainly requires very careful fitting up and handling.

The one factor that limits its applications is the necessity for the maintainance of uniform conditions of speed, and of pressure variation for at least two consecutive minutes. In the present investigation this was obtained sufficiently well to give all the information required. In the case of readings which are on a part of a curve where a rapid change of pressure occurs (e.g.

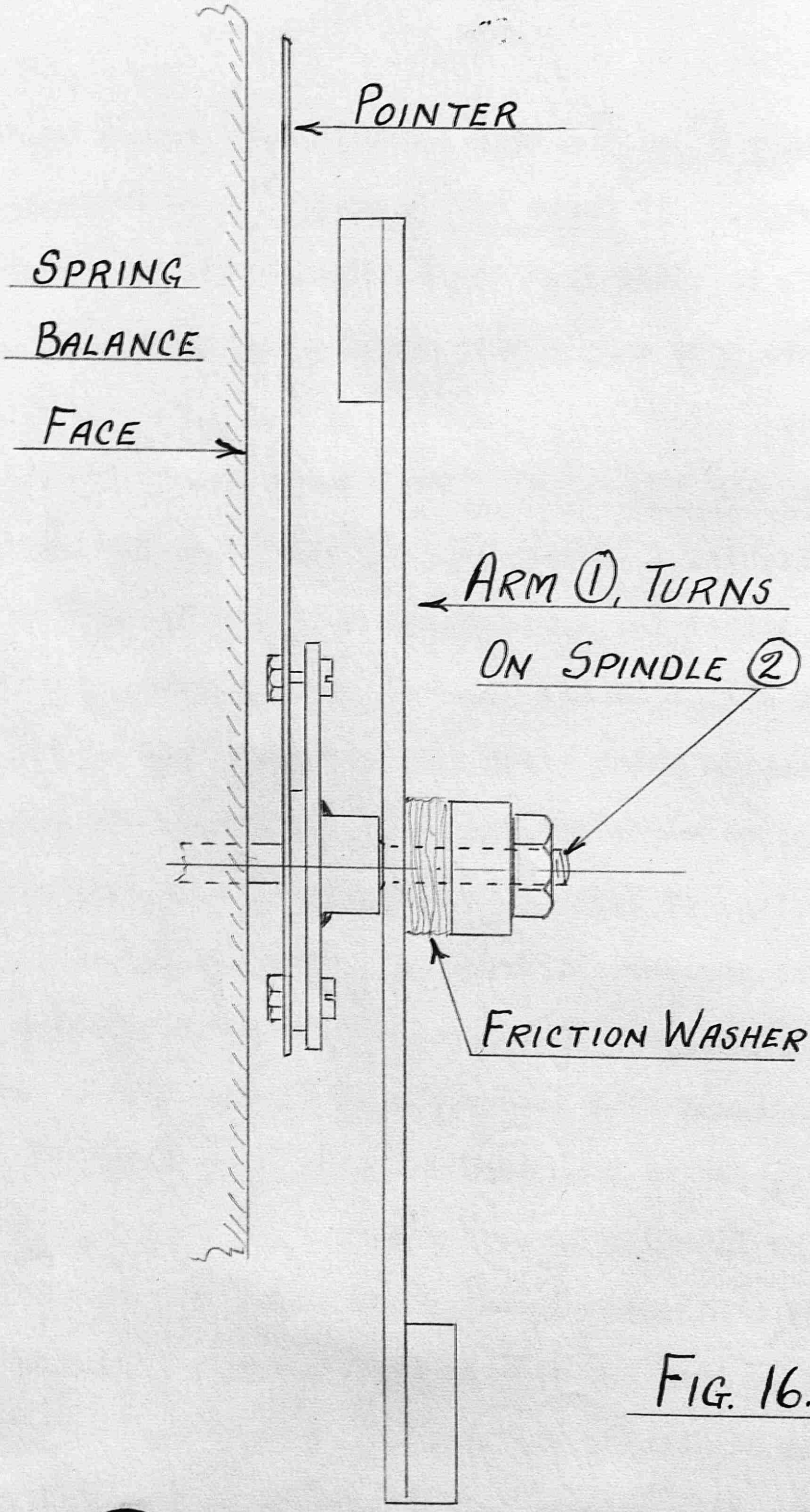

POINTER
SPRING
BALANCE
FACE
ARM ①, TURNS
ON SPINDLE ②
FRICTION WASHER

FIG. 16.

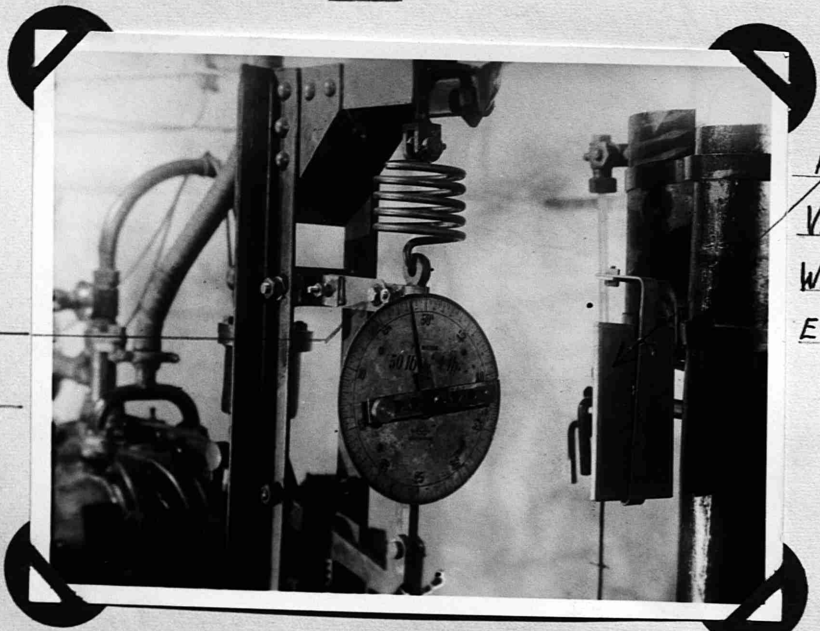

MIRROR FOR
VIEWING BALANCE
WHEN ALTERING
EXHAUST SYSTEM.
DAMPER FOR
TOP SPRING.

Fig. 15) it is evident that a very small change in speed would be accompanied by a large change in pressure. It is found that the mercury level for such a point varies a good deal (i.e. the mercury moves up and down), but this has quite a small effect on the general shape of the curve,

ACCURACY OF BRAKE LOAD READINGS.

The Brake used was intended for absorbing up to 100 H.P. so that great accuracy could not be expected with an engine of 10 H.P. The zero reading was uncertain owing to the stiffness of the water connection but this did not matter as it was only relative Powers that were wanted.

The full-torque load on the Spring Balance was about 20 lb., and the variations due to different exhaust systems were from $^1/_8$ lb. to 1 $^7/_8$ lb. At first the spring balance pointer flickered over $^3/_4$ lb., which made fine readings impossible. As this was partly due to vibrations of the top spring a friction damper was made which took the shake off the spring balance as a whole, but the pointer still flickered too much. Another damper was made for the pointer.

It consists of an arm (1), Fig. 16, - having a relatively large moment of inertia - mounted on a spindle (2) in line with the pointer spindle, and free to turn on it. There is a soft friction washer between the arm and a lock nut on the spindle, so that when the pointer flickers the lock nut rubs on the washer, and the flicker is damped out. The arm being carried on the pointer spindle simply turns with the pointer and in no way influences the readings except to make them much more accurately obtainable.

SECTION III.

EXPERIMENTAL RESULTS AND DISCUSSION.

The exhaust systems tested were (a) different lengths of open pipe, (b) the silencers described later on pages 46-50 on different lengths of pipe.

Pressure curves for all cases were taken with the Multi-Point Indicator; power differences and effect on heating were also noted.

It is not proposed to detail these experiments in the order in which they were performed or to state how and when certain useful facts were observed; the important points will be taken up separately. The first will be the question of the

INFLUENCE OF EXHAUST PRESSURES ON ENGINE POWER.

Engine Power depends on Brake Torque and r.p.m., but for the present purpose the Brake Torque is the more important, and the following figures refer to it only.

When testing various silencers and lengths of pipe at certain constant speeds observations had been made of differences in Brake Torque readings, e.g., at 2000 r.p.m. with an open pipe 3'-4" long the spring balance reading was 19 lb. and when another 2 ft. of pipe was added the reading dropped to $18^1/_8$ lb. This was repeated several times to ensure that the change in the reading was not due to a change in engine conditions - which was not at all unusual. Also when testing a silencer the reading with and without the silencer was noted. It became evident that the maximum power was always got when the pressure at the end of the exhaust stroke was least, and experiments were made to get exact data to show this. A very long pipe was selected to give a pressure much below atmospheric at T.D.C. and was compared with a shorter length of pipe which gave a pressure above atmospheric at T.D.C. Pressure curves were taken for each pipe, and the corresponding Brake Torque readings were noted. This was done also for certain Silencers.

The curves obtained from the Indicator were replotted on a base of piston positions, and the average Effective Pressures for the Exhaust Strokes were found. For Curves see Sheet No.5.

The Brake M.E.P. calculated for this engine working at full throttle within its effective range of speed is 96 lb./in.^{2} As the transmission absorbs a small amount of power it will be convenient to take the true M.E.P. as 100 lb./in.^{2}. This gives a ready means of getting the % effect of the Exhaust Average E.P. on the total M.E.P., i.e., if the Exhaust A.E.P. is 1 lb./in.^{2} over atmospheric it will represent a loss of 1% on the power. This, of course, is not strictly correct but it is near enough to show the general nature of the losses. Also the pressures obtained are not those actually developed in the cylinder but just outside it in the exhaust pipe, and there is at present no means of getting the pressure accurately ~~at present~~ in the cylinder.

EFFECT OF RESIDUAL PRESSURE ON VOLUMETRIC EFFICIENCY.

As already mentioned it was the pressure at the end of the exhaust stroke (referred to here as the Residual Pressure), which seemed to make the greatest difference in power, and this could only be caused by the alteration to the Volumetric Efficiency.

Neglecting the effect of the heating of the fresh charge by the hot gases in the clearance volume and assuming isothermal expansion

$P_1 V_1 = P_2 V_2$ where $P_1 V_1$ are the pressure and volume in the clearance space at T.D.C., and P_2 is atmospheric pressure when the volume V_1 has expanded to V_2. $\therefore$ $V_2 = \frac{P_1 V_1}{P_2}$. Taking the case where P_2 is 15 lb./in.^{2} and P_1 is 16 lb/in.^{2}, and $V_1 = 7{\cdot}01$ in.^{3} $V_2 = 7{\cdot}01 \times \frac{16}{15} = 7{\cdot}477$.

The volume lost to the suction stroke is $\therefore$ = $(7{\cdot}477 - 7{\cdot}01) = 0{\cdot}467$ in^3. The stroke volume is 29·96 in.^{3}, $\therefore$ the loss in volumetric efficiency for one lb. over or under atmospheric pressure is

$$\frac{0{\cdot}467}{29{\cdot}96} \times 100 = 1{\cdot}56\%.$$

If the temperature of the gases in the clearance space is taken at 900° C and allowance is made for the heating effect on the fresh charge it can be shown that the effect on the volumetric efficiency for 1 lb. difference from atmospheric pressure is 2·7%. But brake readings do not show a difference anything like this, and the figure on which the results have been worked is 1·5% per lb. over or under atmospheric pressure.

The results of six tests are shown in table I. System 1 A is compared with 1 B and so on.

In column 4 the Exhaust A.E.P. marked with a negative sign means below atmospheric, and in column (5) for example taking pipe 1 A this pipe is worse off than 1 B by 1·25 lb/in.^{2} above atmospheric + 0·48 under atmospheric, a total of 1·73 which, as already stated, may be taken as a % loss. Column (8) is got by multiplying column (7)

TABLE I. COMPARISON OF THE EFFECTS OF EXHAUST SYSTEMS ON POWER OUTPUT.

1		2	3	4	5	6	7	8	9	10
NUMBER OF CURVE		EXHAUST SYSTEM.	ENGINE SPEED r.p.m.	EXHAUST AVERAGE EFFECTIVE PRESSURE LB/IN².	% DIFFERENCE OF E.A.E.P. BETWEEN 'A' AND 'B'	PRESSURE AT END OF EXHAUST STROKE OVER (+) OR UNDER (−) ATMOSPHERIC. $\frac{LB}{IN^2}$.	DIFFERENCE OF PRESS.^{s} IN COLUMN 6 BETWEEN A & B. LB/IN².	% EFFECT OF THIS (7) DIFFERENCE ON POWER OUTPUT.	NET % DIFFERENCE BETWEEN A AND B 5 ± 8.	ACTUAL % DIFFERENCE GIVEN BY BRAKE READING.
1	A	OPEN PIPE 10'-10" LONG.	2200.	1·25	1·73 LOSS.	−4·5	7·3 GAIN.	10·95 GAIN.	9·22 GAIN.	9·4 GAIN.
	B	" " 5'-2" "		−0·48		+2·8				
2	A	" " 8'-10" "	2500.	2·0	3·125 LOSS.	−4·8	5·8 GAIN	8·7 GAIN	5·57 GAIN	5·0 GAIN.
	B	" " 5'-0" "		−1·125		+1·0				
3	A	" " 8'-10" "	2000.	0·23	0·85 LOSS	−2·0	4·2 GAIN	6·3 GAIN	5·45 GAIN	5·0 GAIN.
	B	" " 5'-3" "		−0·62		+2·2				
4	A	" " 5'-2" "	3500.	0·66	1·37 GAIN	−3·0	6·0 GAIN	9·0 GAIN	10·37 GAIN	7·5 GAIN.
	B	VORTEX STANDARD ON SAME PIPE.		2·03		+3·0				
5	A	CARBJECTOR ON 3'-4" PIPE.	2000	1·2	0·66 LOSS	−0·6	3·0 LOSS	4·5 LOSS	5·16 LOSS	3·75 LOSS.
	B	OPEN PIPE 3'-4" LONG.		0·54		−3·6				
6	A	HOWARTH ON 5'-2" PIPE.	2000	1·43	2·27 LOSS.	−1·0	4·0 GAIN	6·0 GAIN	3·73 GAIN	2·5 GAIN.
	B	OPEN PIPE 5'-2" LONG.		−0·84		+3·0				

by 1·5 giving a figure which is the % difference in volumetric efficiency between A and B. In column (9) taking 1 A and B the total difference between A and B is a gain of 10·95 % for A less a loss of 1·73 % for A = 10·95 - 1·73 = 9·22 %. Column (10) is got from the brake readings, e.g. reading for 1 A is $1^7/_8$ lb. greater than B on an average reading of 20 lb. ∴ gain by A is

$$\frac{1^7/_8}{20} \times 100 = 9{\cdot}4\ \%.$$

It will be seen that the agreement between the actual and calculated differences for the first three is very good considering the rough approximations of the calculation.

The second set are not so good, perhaps because the curves were not all taken at the same time, although the brake readings were taken on the same run.

Making full allowance for the roughness of the methods it is evident that the residual pressure has a much greater effect on power output than has the actual back pressure during the exhaust stroke. Also in case 6 it is notable that while the silencer has a loss of 2·27 % due to back pressure it has a gain of 6 % due to a smaller residual pressure than the open pipe.

It is clear from this that a silencer may increase the power output over that obtained from an open pipe at a certain speed which is unfavourable to the pipe. This is not because the silencer is doing something mysterious or wonderful but because the pipe is the wrong length for this speed.

In case 5 the situation is reversed because the pipe is the right length for the speed, and the silencer is not so good.

Between 39-40

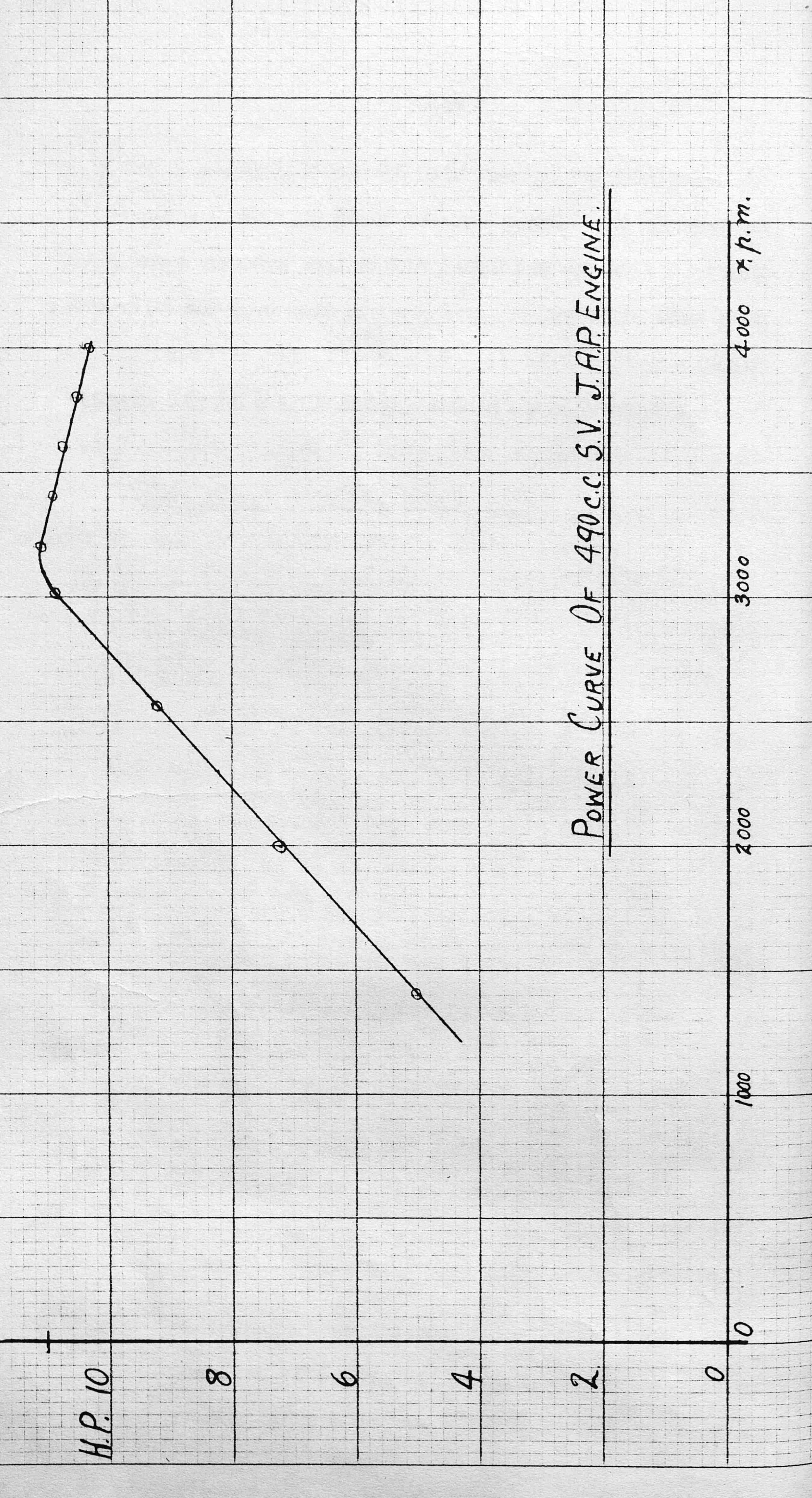

POWER CURVE OF 490 C.C. S.V. J.A.P. ENGINE.
H.P. 10
8
6
4
2
0
0
1000
2000
3000
4000 r.p.m.

FIG. 17.

The effect of baffling or throttling the exhaust as seen in the curves Nos. 11 to 15 on sheets 3 and 4 might in some cases be to prevent the pressure wave rising much over atmospheric, which would increase the volumetric efficiency.

A Power Curve for the engine with a normal length of open pipe is given in Fig. 17.

CRITICAL LENGTH OF PIPE. ~~(pipe is given in Fig. 17.)~~.

In the first series of experiments with the car engine it was observed that adding a silencer to the end of the open pipe did not increase the maximum pressure which generally occurred about B.D.C. This phenomenon was again found with the new engine, and it also appeared to hold for the addition of a long open pipe. But when quite short open pipes were used the maximum pressure was less than with the longer ones, so it was evident that there ought to be a critical length which should give the highest maximum pressure for any given speed. Accordingly, this theory was tested by using a telescoping pipe and varying the length while the engine was running, meanwhile observing the pressure. It was found that there was a certain length below which the pressure fell decidedly. If the length was increased the pressure dropped a little but not so much as for a corresponding shortening of the pipe.

The following explanation is suggested :- Consider a pipe of length greater than the critical length as described above and let there be a moving piston at the left end, see Fig. 18, the right end being open.

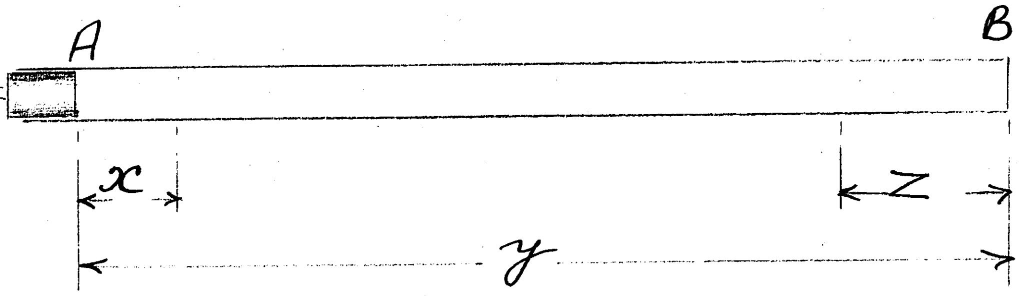


Fig. 18.

If this piston suddenly begins to enter the pipe at a very high speed the air is compressed because of its inertia, and a building up of pressure goes on at the piston face till the piston reaches its top speed after which it may be supposed to slow down again. Let the distance travelled by the piston during the high speed period be 'x'. The air pressure of the piston face began travelling along the pipe with the speed of sound at the instant the piston started, and covered a distance 'y' while the piston did 'x'.

The air pressure at the piston is due to the inertia of the column of air of length y and any length of pipe beyond this will have no effect on the maximum pressure. This maximum pressure is reduced if the pipe length is less than 'y' by, say, a distance 'z' because the resistance due to the inertia of the column in 'z' has been lost.

If the speed of the piston is increased the distance 'y' must decrease since there is less time for the propagation of the pressure along the pipe.

In the case of the engine instead of the piston there is the rapid discharge of gas from the cylinder into the valve pocket as the valve begins to open, and this has practically the same effect as a piston.

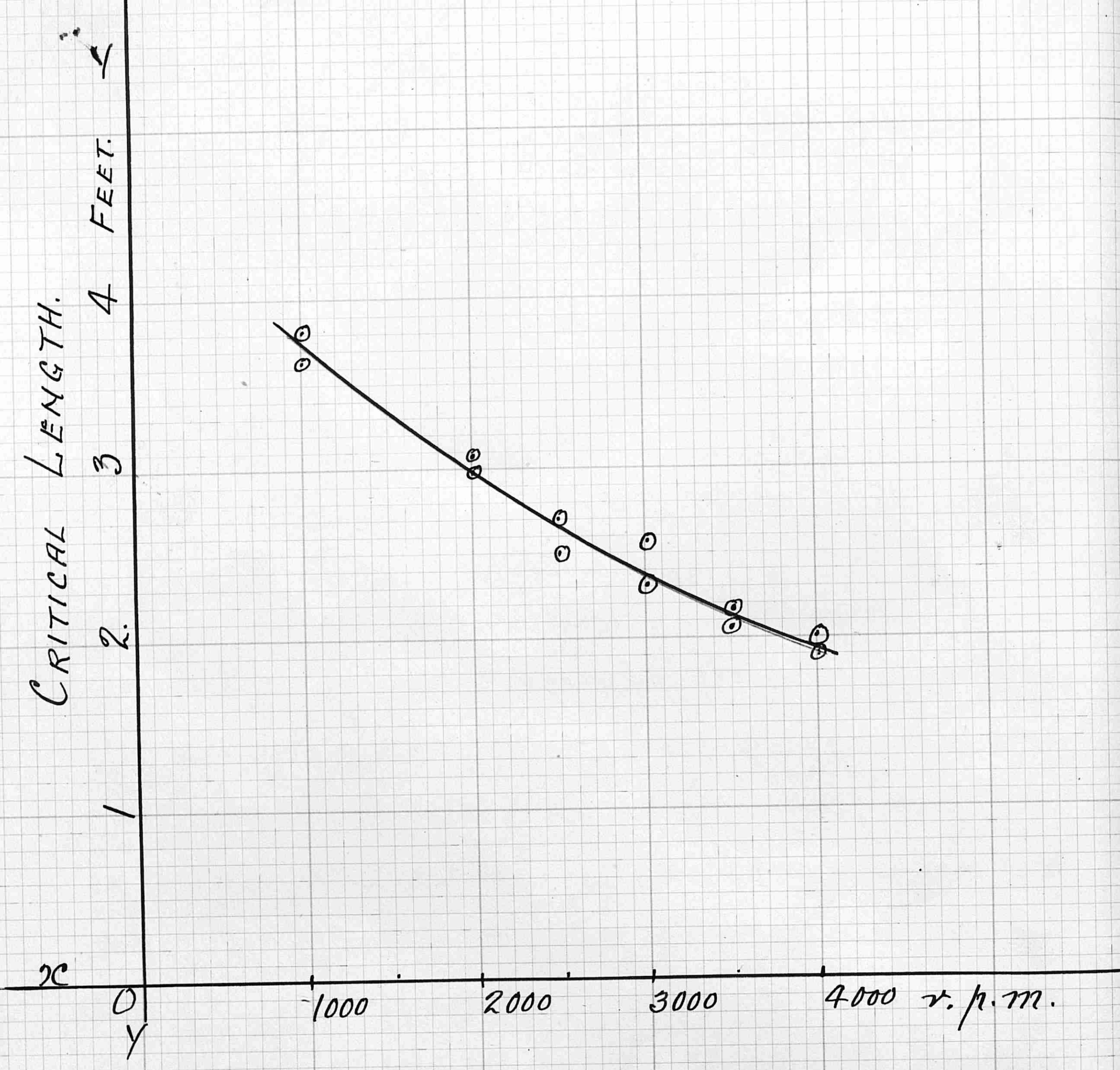


TABLE 2.

ENGINE SPEED	CRITICAL LENGTH.
1000 r.p.m.	3'-8" To 10"
2000	3'-0" To 1"
2500	2'-6" To 9"
3000	2'-4" To 7"
3500	2'-1" To 2"
4000.	1'-11" To 2'-0"

FIG. 19

When the engine speed increases the rate of opening of the valve increases proportionally, and so the critical length should decrease with increased speed.

To determine this the engine was run at speeds of 1000 to 4000 r.p.m. and the critical length found for each speed, see table 2, Fig. 19. The critical length is not easy to find as an inch or two does not make much difference to the pressure.

When the critical lengths are plotted on a base of revolutions the graph might easily be made a straight line which could not be correct as it would give a length of nothing at about 8000 r.p.m. and this is impossible. It is probably a curve which is asymptotic to ~~this~~ the x axis.

When the speed is very low the rate of opening of the valve would be too slow to produce a proper wave motion, so the curve may cease to exist near the y axis. But even if the critical length cannot be gauged accurately to less than $\pm$ 1·5" the result is of great importance and explains certain phenomena indicated by most of the curves.

RESULTS OBTAINED FROM THE CURVES.

Although the primary object of the research was to obtain information which would lead to the design of a more perfect silencer it was considered advisable to go into the behaviour of open pipes before dealing with silencers.

In the first experiment the engine was run under full load at different speeds with an open exhaust pipe of normal size - $1^3/4$" diameter, and 5'-2" long.

The curves obtained are shown on Sheets Nos. 7 and 8.

The key to these curves, <u>Sheet No.6.</u>, should first be studied, as full details are not given on each sheet. Ordinary graph paper was not used because the cross lines tend to obscure the shape of the curves.

<u>CURVE No. 20.</u> for the lowest speed of 1000 r.p.m. is very smooth and almost dies away between each exhaust period. As will be shown in the section dealing with sound these pressure waves are really sound waves and their frequencies correspond to those calculated for an organ pipe having the dimensions of this exhaust pipe, closed at one end and open at the other. This condition obtains only when the exhaust valve is closed, but there is very little difference between the frequency of the wave before and after the closing of the exhaust valve.

The frequency of the fundamental wave is not affected by the speed of the engine, but as the speed rises there is not time for the wave to damp out, and there is consequently a residual wave in the pipe when the next wave begins. This residual wave combines with the new one and alters its character. Also the harmonics of the fundamental wave become pronounced. In some cases such as No.24. at 3000 r.p.m. the fundamental is scarcely recognisable. In general the combination of waves tends to lower the pressure rather than to form nodes of higher pressure.

The maximum pressure occurs a little before B.D.C. for low speeds, and a little after B.D.C. for very high speeds. The intensity of this maximum pressure increases with the speed but* is also modified by the residual wave, e.g. in No.23. the residual wave has a pressure node at

* See also conclusions by Mr Mock, p. 63.

B.D.C. which increases the maximum pressure at that point, whereas in No.24. the reverse is the case. This makes the maximum less for a speed of 3000 r.p.m. than for 2500 r.p.m. and the same effect can be seen for speeds of 3500 and 4000 r.p.m.

RESONANCE.

Although the residual wave has a decided effect on the maximum pressure it is unlikely that resonance caused by the coincidence of engine speed and wave period, would *ever be attained because the harmonics would interfere so much with the fundamental wave. Also the engine speed required for resonance is beyond that found in practice.

SCAVENGING.

As mentioned later, page 65, in connection with Mr. Briggs ~~Mock's~~ paper, there is a widespread opinion that the inertia of the rapidly moving gases in the exhaust pipe immediately after the discharge of the main portion of the exhaust gases, causes a partial vacuum in the cylinder head.

But there is no evidence that the speed of the gases has any appreciable effect on the pressure in the pipe. The first rush of gas into the pipe has an explosive effect and starts a pressure wave which seems to be independent of the speed of the gas.

The scavenging effect of a long pipe depends on whether the part of the wave which occurs at T.D.C. is above or below atmospheric pressure. For a given pipe this effect varies with the speed, e.g. at 2000 r.p.m. - No.22., the pressure is 3 lb/in.^{2} above atmospheric, and

* This opinion is rather modified by the curve for 5000 rpm. which was taken after this part of the work had been completed.

at 3000 r.p.m. No.24 it is 3·5 lb. below atmospheric.

It does not seem possible to have any length of open pipe which will assist scavenging over a wide range of engine speeds.

In motor cycle and car racing in cases where silencers are not compulsory it is sometimes the practice to find by experiment the length of pipe which gives maximum power at the highest engine speeds.

The third Curve No.48, on Sheet No.8., does not belong to this series but is inserted here to show the remarkable similarity between the first portion of it and of the preceding Curve No.26. This will be referred to later, p. 57

In the second experiment the engine was run at a constant speed and full load, and with varying length of the pipe.

The speed chosen was 2000 r.p.m., as the engine then ran very steadily and did not require much attention.

The critical length for this speed as stated in the table on p. 42 , is about 3 ft. Curves Nos. 27 and 28 are from pipes much in excess of this length. The length for No.29 is a little over, while Nos. 30 and 31 are considerably under the critical length. With a very long pipe, No.27, there may be nodes of fairly high ~~and~~ or low pressure, but the curve in general is levelled down by the residual wave and the harmonics. The pipe nearest the critical length, No.29, gives the highest maximum pressure at B.D.C., and the harmonics are not so pronounced as with the longer pipe. As is well known it does not follow that because a curve looks like a regular sine curve, e.g. the latter part of No.30, it has no harmonics, but these, if present, do not alter the fundamental to the same extent as in Nos. 27 and 28.

No.31 for the very short length looked irregular and another curve, No.32, was taken for a higher speed, but the characteristics of the two curves are similar.

The maximum pressure occurs earlier and is lower with the shorter pipes Nos.30 and 31, than with those above the critical length.

Reference is made to the frequencies of the pressure waves for Nos. 28, 29, 30, 31 and 32, in the section on Sound.

Before considering the results of the silencer tests it will be necessary to study the design of the silencers used in the tests.

The Howarth Silencer, Drawing No. 10, is made by the Scott Motor Cycle Co., and won the first prize in a competition held by the A.C.U. for proprietary silencers. The inlet pipe (A) traverses the full length of the silencer and terminates in a nozzle (B) which is intended to act as an ejector. The main volume of the gases passes through two slots in pipe (A) to the coned chamber (C) and returns along the outside of this chamber to a ring of holes in the centre division plate (D). There is another coned chamber (E) from which the gases may pass to the exit through holes in the cone and through the open end. The makers of this silencer have not advanced any theory of its action, except with respect to the ejector the purpose of which is obvious.

It should be borne in mind that the majority of silencers are designed on the supposition that the exhaust gases enter them at a high pressure, and the main object of the silencer is to provide passages or chambers in which the gases can expand before finally reaching the atmosphere. As ~~shown~~ stated later this investigation shows that this assumption is not justified.

The Carbjector silencer was chosen because it has been the most popular proprietory silencer for a number of years and has been used for purposes other than motor cycle engine silencing.

Two examples are shown in Drawing No.10, the first one being taken from the Patent specification and the second is the design as tested. To quote from the specification: ..."The gases...are forced through a thin flexible diaphragm to absorb the energy of the gases. Preferably the diaphragm is a thin flexible circular member of copper centrally perforated to the area of one exhaust valve. The gases pass through the cylindrical pipe by a gradually expanding passage of spiral form. In this way the gases gradually expand after leaving the perforated diaphragm. A portion of the exhaust gases passes down the central tube and creates a suction on leaving the nozzle, thus assisting the removal of the exhaust gases." What particular kind of energy the diaphragm is supposed to absorb and how it is going to dissipate the energy is not mentioned. This diaphragm is not present in the motor cycle models, and as no drawing was available and the silencer could not be taken to pieces, it is ~~not clear~~ Fig. 20 whether the spiral is an expanding one or not. Measurements taken from the ends give the impression that the spiral has the same pitch all the way.

The Vortex Silencer is widely advertised and makes great claims. In order to test the fundamental idea of the design the Vortex Co. was asked to make a special single tube silencer as shown in Fig. (1), Drawing No.11. The following is an extract from the makers' description:- "The absence of baffle-plates in the Vortex prevents the vibration of exhaust gases in the exhaust manifold that occurs with ordinary silencers and causes periods of acute engine oscillation at certain speeds. In the Vortex Silencer the exhaust gases nowhere impinge on any obstruction, consequently their volume is gradually deflated without causing a series of sound waves to strike suddenly on the outside atmosphere. Baffle-plates and muffles have been abandoned, and instead, the exhaust gases are poured in a smooth continuous flow through a succession of S-shaped Vortex cups. These cups are so arranged that they cause a smooth rotary progression of the exhaust gases, thus creating a strong suction which draws every particle of the exploded charge from the cylinder head."

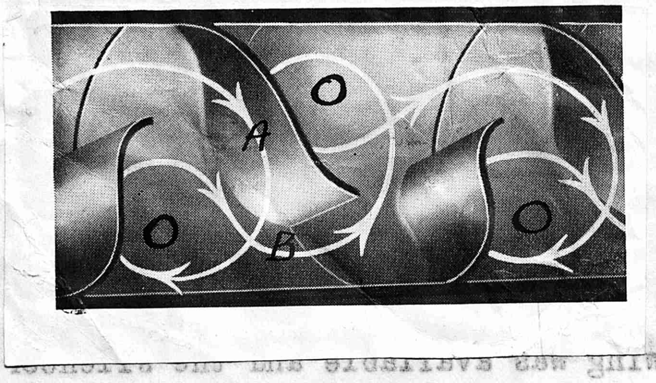


FIG. 20

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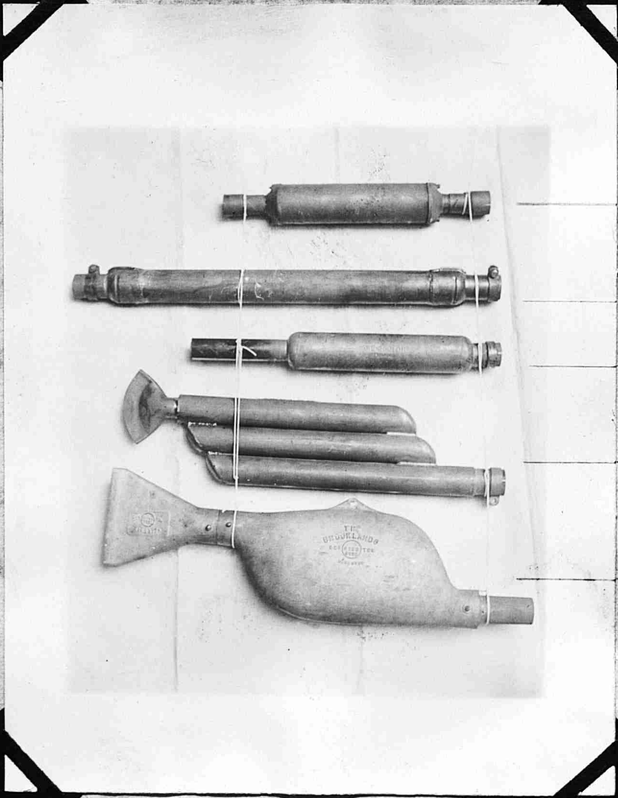
It is quite true that the gases will swirl in the parts marked 'O', Fig. 20, but this can scarcely be called a "vortex", and it clearly cannot assist the forward motion of the main stream. Also there can be no "smooth uninterrupted flow" if the stream 'A' cuts across the stream 'B'. The Vortex silencer, Fig.(2), Drawing No.11, is the latest pattern and was lent for a few days for testing purposes. Instead of a single tube there are three tubes arranged to give parallel flow, and the final exit is by a fishtail.

The Brooklands Standard Silencer. At one time it was customary to race at Brooklands with open exhaust pipes both on motor cycles and cars, but this was so objectionable that a regulation was made to the effect that all engines should have a silencer, the main features of the design of which should be as follows :- "The capacity of the receiver shall be not less than 6 times the volume swept out by the piston of one cylinder of the engine. The inlet pipe shall penetrate into the receiver to a distance of 2" and no more, and the same applies to the exit pipe. The top, bottom and ends of the receiver, in side elevation, shall be straight lines and shall be parallel to one another respectively. The ends shall not be at an angle less than 45 degrees to the top and bottom lines of the receiver. In addition, the extensions of the inlet and exit pipes into the receiver must be parallel to one another and to the top and bottom of the receiver. No part of the exit pipe shall be of greater cross-sectional area than the minimum area of the exhaust port of any one cylinder. The inlet and exit pipes shall not be opposite each other in the receiver, but shall be out of line to the extent of 1½ inches measured at points on the circumference. No device may be employed in the receiver which would tend to produce a straight through flow of gases between inlet and outlet pipes. The exhaust gases must not pass direct from the exit pipe to the atmosphere but must all be finally emitted from a "fishtail" on the end of the exit pipe. The Orifice of such a fishtail shall be of rectangular shape 1¼" X 6½" for motor cycle engines, and the length of the fishtail 6½" The after half of the fishtail may be perforated with any number of holes $^3/_{32}$" diameter."

The silencer used in the tests was a standard model intended for use at Brooklands on engines up to 500 c.c. capacity. There were no small holes bored in the sides of the fishtail which was easily detachable. The photograph gives an idea of the relative sizes of the silencers.



HOWARTH

VORTEX TUBE

CARBJECTOR

VORTEX STANDARD

BROOKLANDS STANDARD

These silencers are of very different types. The Brooklands is simply an open box of large volume with a fishtail exit; in the Howarth the direction of the gases is twice reversed; the Carbjector has a long spiral passage with no reversal of direction or baffles; the Vortex Tube is supposed to give a smooth flow, but it is clear that a good deal of the kinetic energy of the gases will be lost by turbulence in the space behind each ~~wave~~ vane; the Vortex Standard differs from the Vortex Tube in being much shorter, having a larger volume and terminating in a fishtail.

Before the tests on the engine were run a preliminary series of experiments was carried out on the resistance offered by the silencers to a steady flow of air. There was not a sufficient supply of compressed aib to give a speed of flow such as would obtain in working conditions, so the results are only comparative.

The silencers were attached to the discharge pipe of an air reservoir which was fed from an air-compressor. The rate of discharge was regulated by a valve, and a uniform pressure was maintained in the receiver during one test. The resistance to flow through the silencer was measured by a water manometer connected to the pipe leading to the silencer.

In the table the resistances are given for two pressures in the air bottle, 2A being twice the pressure of A. In some cases the silencer was also tested with the air flowing in the direction opposite to the normal.

TABLE III.

Silencer	Manometer Pressure in inlet pipe in inches of water.		
	A	2A	With silencer reversed & pressure as in 2A
Brooklands	1·5	4	
Vortex Standard	8·5	19·5	
Howarth	10	24·5	26
Carbjector	11	25·5	29
Vortex Tube	17	42	
Vortex Tube at a low pressure	12		26 (for pressure as in A).

The Brooklands Silencer has much less back pressure than any of the others. The Howarth and Carbjector are about the same. The Vortex Tube has double the resistance of the Vortex Standard. This is due to the Vortex Tube being much longer and having a smaller cross sectional area than the Vortex Standard.

With the silencers reversed the back pressure of the Howarth and Carbjector increases by about 12%, while that of the Vortex Tube is more than doubled.

The behaviour of the silencers on the engine is, however, quite different to what the above results would lead one to expect.

For purposes of comparison silencers can be divided into two types, (a) those in which the stream of gas meets an obstruction immediately on entering the silencer, e.g., the Howarth, Carbjector, and Vortex; (b) those in which there is chamber of relatively large volume before any obstruction is reached, e.g., the Brooklands Silencer.

EXPERIMENTS WITH THE SILENCERS ON THE ENGINE.

FIRST SERIES: Comparison of different silencers at a constant engine speed of 2000 r.p.m.

In this and in all other experiments the engine was run at full throttle and maximum load.

A standard Brooklands Fishtail was also included in this series. It was simply fixed to the end of the exhaust pipe without being attached to any silencer.

It was difficult to keep conditions absolutely constant for all the tests, but they were never unduly variable.

Observations were taken of the differences in brake loads. The correlation of the exhaust stroke pressure curve with the variation in brake torque has already been dealt with, and the conclusions then arrived at will be made use of in the discussion of the curves.

The temperatures of the cooling water from the engine and the exhaust branch were noted, but as no exhaust system appeared to make any appreciable difference to either of these temperatures this practice was discontinued.

The critical length of the pipe had not at this time been studied. It was fortunate that in this series of tests the pipe to which the silencers were attached was just over the critical length for the speed of 2000 r.p.m. As a consequence of this the maximum initial pressures at 'A' are nearly the same in each case.

Curves Nos.33 and 34, Sheet No.10, for the Howarth and Carbjector silencers are almost identical. In Curves Nos.33 to 36, the second pressure peak 'B' is due to a *pressure wave being reflected from the silencers which are

* See also curves by Dr Watson, Fig 22, p. 62.

EXPERIMENTS WITH THE SILENCERS ON THE ENGINE.

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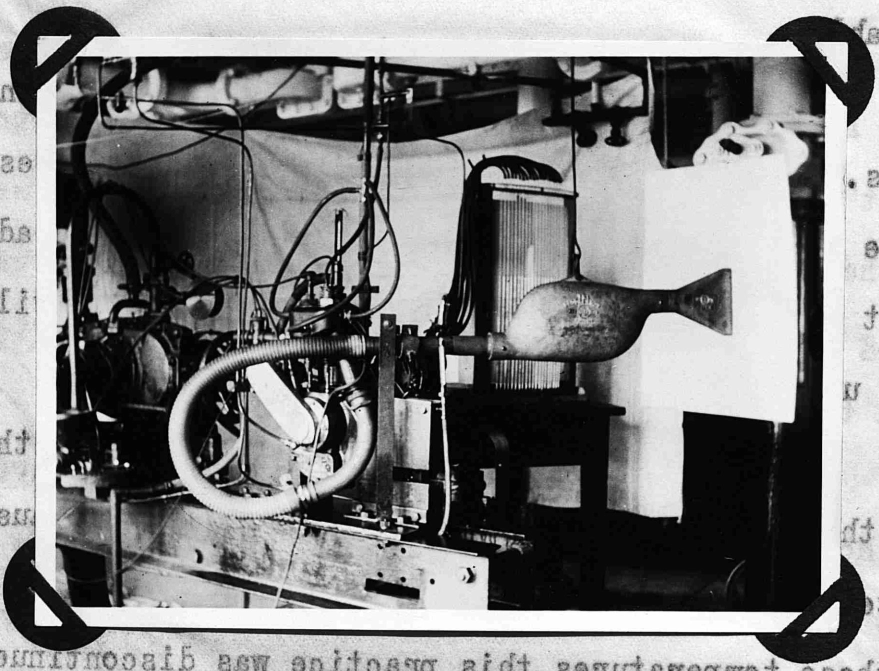

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*... Fig 22. p.62.

of type (a) mentioned on p. 52 .

The Brooklands Silencer is of type (b) which allows the pressure wave to escape from the end of the pipe to which ~~it~~ the silencer is attached and return as a negative wave. There is very little damping in this case.

The Fishtail, Curve No.38, has a very good damping effect.

The Vortex Standard Silencer is really of type (a) but it has a bigger free volume than the first three, and the reflected wave takes longer to form, i.e. the time between 'A' and 'B' is greater than in Curves Nos. 33-4-5.

This reflected wave was further investigated in the

SECOND SERIES OF EXPERIMENTS. A flexible exhaust pipe was bent so as to bring the part where the silencer was attached close to the Indicator, see photograph and sketch on Sheet No.6. A rubber tube connected the offtake from the pipe at a point close to the silencer to the three-way cock on the Indicator, see Fig. 10 ,p. 22.. A curve of pressure at the exhaust stub was taken as before, and then the connection from the end of the pipe just before the silencer was made, and a curve of pressures was taken for this point.

The Curves thus obtained are given on Sheet No.11. Considering Curves Nos.42A and B, and 43A and B, the maximum pressure just before the silencer is greater than at the exhaust stub. This may be due to the fact that in addition to the normal wave pressure there is a resistance in the silencer to the first rush of gas.

The flexible pipe was tested without a silencer. The offtake was 4 ins. from the open end, and as this is much below the critical length for this speed the maximum pressure is less than at the exhaust stub, see Curves Nos. 41 A and B. A pipe just over the critical length was added to the end of the flexible pipe so that the full pressure of the wave might be retained. Curves Nos. 39 A and B show that this pressure is retained. The time taken for the pressure node (1) Curve A, to reappear at (2) in Curve B, gives the speed of the wave along the pipe as 1400 ft./sec. - see later page 70. When Curves A and B are superimposed with the node (2) advanced to the position of (1), Curve No.40, the period of the fundamental waves are seen to be the same, but the harmonics occurring at different times in A and B prevent the curves coinciding completely. Assuming the same wave speed for Curves Nos. 41 A and B, it is found that the node (2) of B occurs much too early making the wave speed very high. At the point where Curve B is taken the wave is emerging from the pipe and the resistance is consequently very much less than that of another length of pipe and this may account for the apparent increase in speed. Similar considerations apply to Curves Nos. 44 A and B.

When a silencer is attached, Curves Nos. 42 and 43, the time between nodes (1) and (2) is slightly greater than that corresponding to the speed of 1400 ft./sec., due probably to the greater resistance of the silencer.

The successive reflections of the wave from the silencer to the engine and back are clearly shown; node (2) reappears at (3), (3) at (4), (4) at (5), and (5) at (6). A large proportion of the energy is lost at each reflection being entirely dissipated before the end of the cycle.

A method of finding the effect of the silencer on the pressure wave is shown in Fig. 21. The silencer was fitted up in the same way as in the experiments just described, and a long tail pipe was attached with an offtake to the Indicator just after the silencer. Curves Nos. 80 and 81, sheet 16, were taken before and after the silencer. The pressure is reduced by half in going through the silencer and the crest of the wave is much rounded.

The other silencers might have been tested in the same way but a better method of comparing the silencing effects was found and is described on pages 88-9.

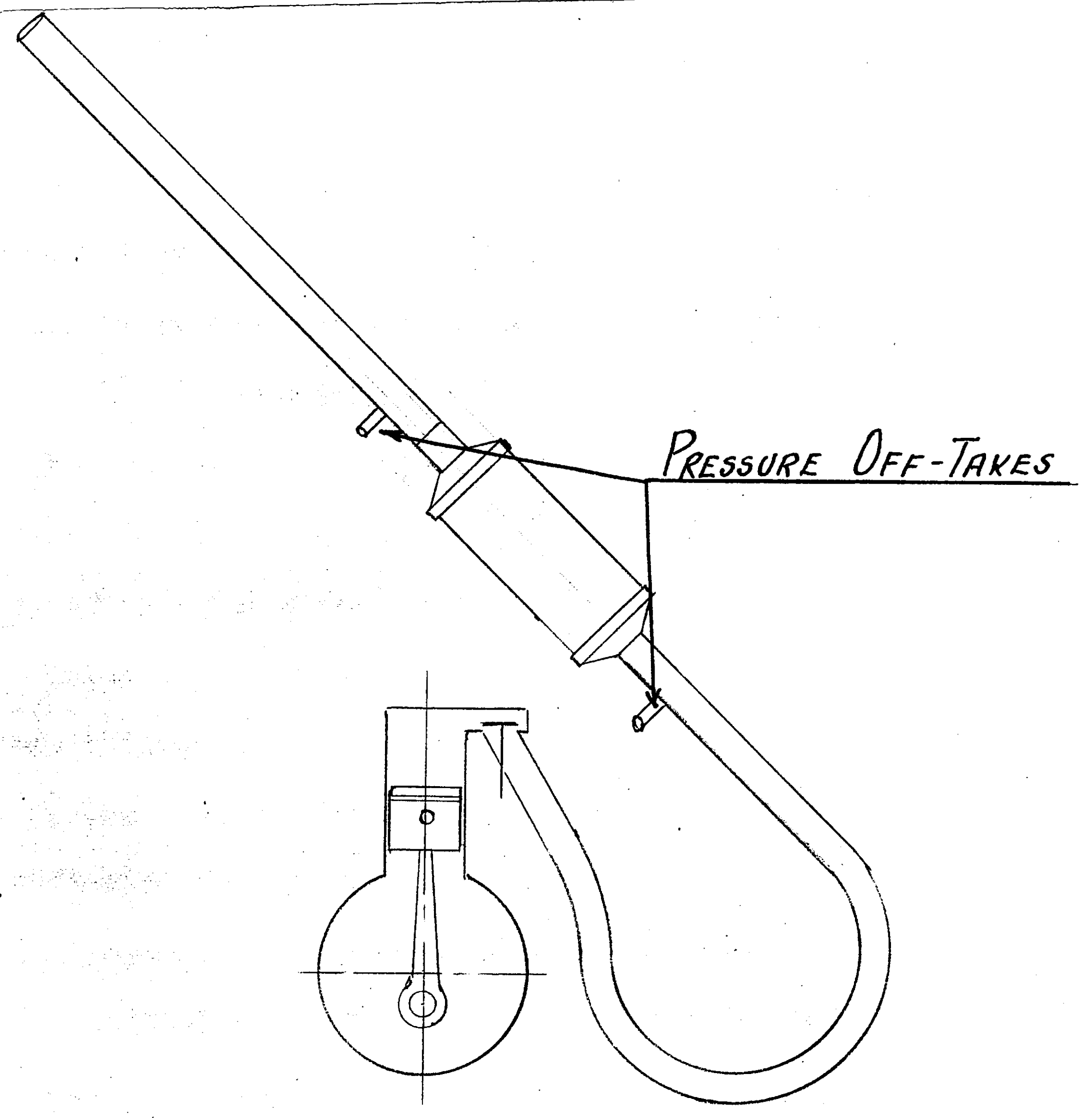

FIG. 21.

THIRD SERIES OF EXPERIMENTS: In this series the silencers were tested at different engine speeds. The same pipe was used throughout and was above the critical length for all the speeds. See Curves Nos.45 to 58 on Sheets 12 and 13. All pressures were taken at the stub. In all cases the initial maximum pressure increases and takes place later as the speed rises. The two facts to be considered are (a) the average back pressure during the exhaust stroke, and (b) the residual pressure at the end of the stroke. The T.D.C. is marked on all the curves.

At 2000 r.p.m. there is not much objection to any of the silencers except that the Brooklands has a node of pressure at T.D.C. At 3000 r.p.m. and above the reflected wave becomes very prominent and causes considerable back pressure.

The residual pressure for the Howarth is just beginning to be above atmospheric at 4000 r.p.m., Curve No.48, after which speed it would increase rapidly.

Referring to the comparison already made on p. 45. between this curve No.48 and the curve for the open pipe at the same speed, sheet 8, it will be seen that the reflected wave in curve No.48 happens to be similar to that produced in the open pipe by the combination of the fundamental wave, the residual wave and the harmonics.

The reflected wave in the Vortex Standard Silencer at 4000 r.p.m., Curve No.51, is giving a high residual pressure which would increase with the speed. The Carbjector has much the same effect as the Howarth. The Vortex Tube Silencer, Curve No.55, is also liable to give high residual pressure at high speeds. The effect of the Brooklands is similar to that of the open pipe in this respect that its pressure at T.D.C. is above or below atmospheric depending on the engine speed. Between 2500 and 3000 r.p.m. the residual pressure is well below atmospheric, and above 3400 r.p.m. the residual pressure will be considerably above.

FOURTH SERIES OF EXPERIMENTS: Silencers on different lengths of pipe - Curves Nos.59 to 72, Sheets Nos. 14 and 15. In this series the engine speed was kept constant during a test, and the length of the pipe between the engine and the silencer was varied from above the critical length to the shortest possible length. All pressures were taken at the stub. The distance between the first two pressure nodes increases with the length owing to the longer time required for the wave to be reflected, see Curves Nos.59-60, and 67-68, and 70-71. In the case of the Vortex Standard Silencer, Curves Nos. 67-68, this has a bad effect at high speeds. The Brooklands Silencer on a very long pipe at 3500 r.p.m., Curve No.69, has a node of pressure 4½ lb. below atmospheric at T.D.C., and there would also be a node of low pressure for a length between 3'-4" and 1'-7", see Curves Nos.70 and 71.

In order to see if it were possible to get good silencing and retain this advantage of a node of low pressure at T.D.C., the fishtail was taken off the Brooklands, and the Howarth was put on instead. Curve No.72. for this arrangement appears to give the desired effect, but there is so much of the reflected wave above the atmospheric line that the average back pressure is raised, and conditions on the whole are not so good as might have been expected. The initial maximum pressure increases for silencers of type (a) - Howarth, etc., when the pipe is below the critical length, and decreases for that condition for type (b).

The increase for type (a) is due to the resistance in the silencer to the flow of gas being additional to the inertia force of the column of gas from the beginning of the exhaust pipe to the end of the silencer.

The decrease for the Brooklands, type (b), is explained by the relative shortness of the column of gas, and the absence of any obstruction in the first part of the silencer. Curve No.73 compared with Curve No.71 shows this difference clearly.

FIFTH SERIES OF EXPERIMENTS. This is not a connected series, but consists of tests of silencers reversed, silencers in series, and a silencer with a tail pipe.

Curves Nos.74 A and B, sheet No.16, show the effect of reversing the Howarth Silencer. See also Curves Nos. 9 A and B, sheet No.3, for the car engine. In both cases the initial maximum pressure is less when the silencer is reversed. This is because the silencer in the normal position is of type (a) and when reversed is tending more to type (b). It is better in the normal position because the higher initial pressure does not matter, and the pressure decreases more quickly than when reversed. The Vortex Tube Silencer was also tested in the reverse direction. The pressure was taken at a point on the pipe 4" from the silencer, see Curve No.79. The first pressure wave is not altered but the second one is doubled. The main point in reversing the silencers is to show that their performance cannot be predicted from considerations based on continuous gas flow.

SILENCERS IN SERIES.

It is sometimes the practice to use two silencers in series. The result of using the Howarth and Carbjector silencers in series is shown in Curves Nos.75 and 76.

This had already been done on the car engine, see Curve No.10, Sheet No.3. The initial maximum pressure for the Howarth alone, Curves Nos.75 and 76, is slightly greater than when the second silencer is added, but the curves were not taken on the same run so the comparison is not quite valid. At the lower speed, Curve No.75, the two silencers give very little more back pressure than one alone, but at the higher speed, Curve No.76, the back pressure and the residual pressure are both high.

Silencers in parallel have not yet been tested. It is the case that two silencers, if as efficient as the Howarth, give as good silencing when used in parallel as in series, and the use of twin port exhaust systems with two silencers is becoming popular.

ADDITION OF A TAIL PIPE.

The Howarth was attached to a short pipe, 1'-7" long, and had a tail pipe 3'-6" long, open at the end. The first part of the curve obtained for this condition, No.77, is very similar to the same part of the curve for the silencers in series, No.76. This shows that the back pressure given by the silencers in series is mostly an inertia effect. The frequency of the waves in Curve No.77 is very low, which would be advantageous in allowing the pressure at T.D.C. to remain below atmospheric for a wide range of speed up to 3000 r.p.m. Above this speed the back pressure and the residual pressure would both be high. The addition of a fishtail to the end of the tail pipe adds a little to the residual pressure and damps the subsequent wave.

An indefinite number of experiments might be performed with these and other silencers, but enough has been done to cover the important points. ~~These points are dealt with further in the General Discussion.~~

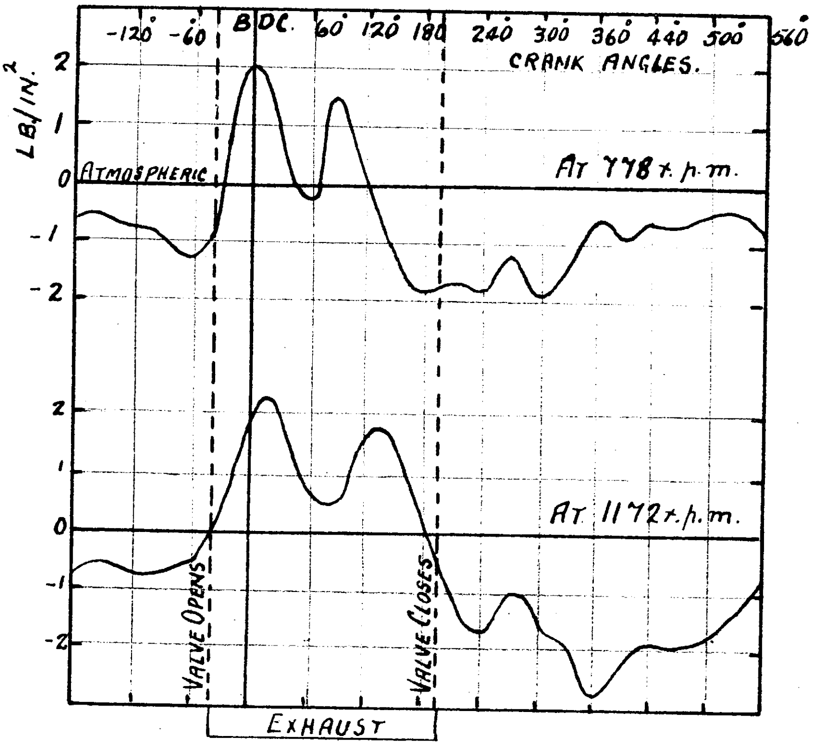
-120° -60 B.D.C. 60° 120° 180 240 300 360 440 500 560
CRANK ANGLES.
LB/IN.²
2
1
0
-1
-2
ATMOSPHERIC
AT 778 r.p.m.
2
1
0
-1
-2
AT 1172 r.p.m.
VALVE OPENS
VALVE CLOSES
EXHAUST

FIG. 22.

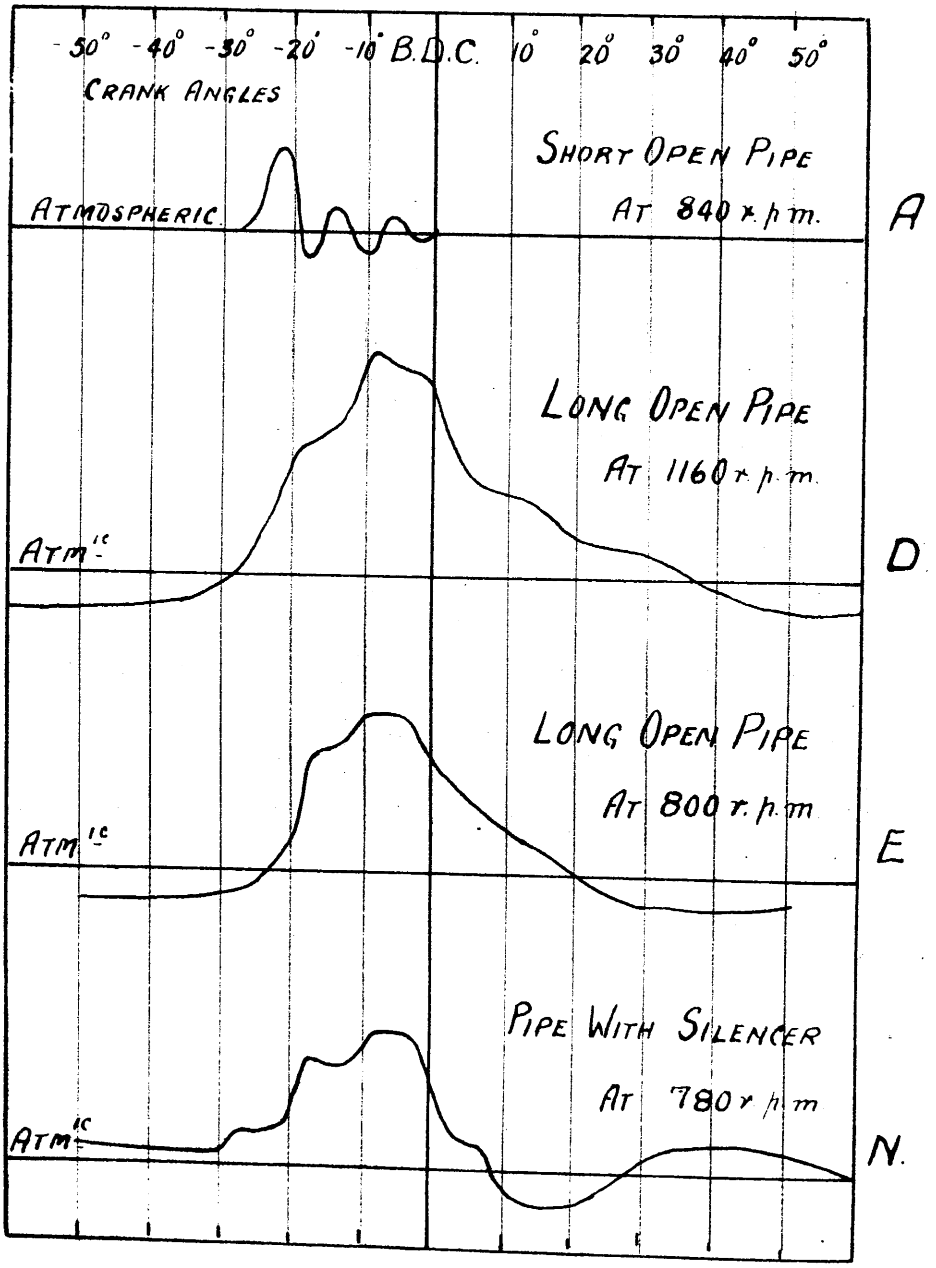
-50° -40° -30° -20° -10° B.D.C. 10° 20° 30° 40° 50°
CRANK ANGLES
SHORT OPEN PIPE
AT 840 r.p.m.
ATMOSPHERIC
A
LONG OPEN PIPE
AT 1160 r.p.m.
ATM^IC
D
LONG OPEN PIPE
AT 800 r.p.m
ATM^IC
E
PIPE WITH SILENCER
AT 780 r.p.m
ATM^IC
N.

FIG. 23.

EXPERIMENTAL WORK BY OTHER INVESTIGATORS.

Diagrams of exhaust pressures at fairly high speeds have been obtained by other investigators and are given in the following papers:-

*(1) Tests of a Daimler Knight Engine, by Dr. Watson.

The exhaust tests in this case were only supplementary to the main engine tests and were not considered to be of any special importance. The diagrams are, however, of interest in that they show the phenomenon of the pressure wave being reflected from the silencer, see page 55. A copy of the diagrams taken at the lowest and highest speeds is given in Fig. 22.

The pressures are very much lower than those obtained by the writer from the J.A.P. Engine but the general character of the waves is similar.

**(2) The Exhaust Discharge and Proper Manifold Design, by Mr. F.C. Mock.

Mr. Mock states that the diagrams shown in his paper were obtained hurriedly with an optical indicator and are only intended to show the nature of the pressure variation. Some of these diagrams have been enlarged, see Fig. 23., but true scales of pressure and angular position cannot be given as they are not clear on the original diagrams. Also there is not enough data of engine, exhaust valve and pipe sizes to enable a proper comparison to be made between these curves and those obtained by the writer. With reference to

* See The Autocar, Nov. 9th, 1912.
** See the Society of Automobile Engineers Bulletin, N.Y. 1913.

EXPERIMENTAL WORK BY OTHER INVESTIGATORS.

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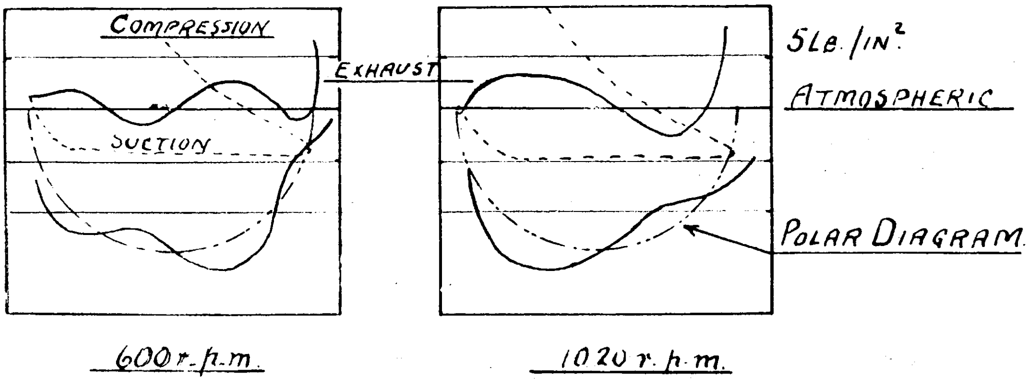

FIG 24.

enable a proper comparison to be made between these curves and those obtained by the writer. With reference to

* See The Autocar, Nov. 8th, 1913.
** See the Society of Automobile Engineers Bulletin, N.Y. 1913.

curve 'A', Fig. 23., he states "The wave motion is very evident and the period is very close to that of a closed end organ pipe of the same length". This curve is similar to the writer's curve for a short open pipe No.31, sheet 9.

In connection with the initial maximum pressure his conclusion - "It is noticeable that the shock at the upper end of the pipe becomes greater as the speed increases", is the same as that of the writer's, see pages 41 and 43

*(3) Inlet and Exhaust Phenomena, by Mr. K.J. de Juhász.

The diagrams accompanying Mr. de Juhász's article are specially interesting because they show conditions inside the cylinder and prove that the wave pressures in the pipe are transferred to the gas in the cylinder. The diagrams were taken with an ordinary indicator using the de Juhász device** which enables an indicator to be run at a hundredth of the engine speed. This arrangement appears to give excellent results up to quite high speeds. Unfortunately the speeds used in these particular experiments were not very high - the highest being 1100 r.p.m., but this is enough to show the existence of the pressure wave phenomena. Copies of two of the diagrams are shown in Fig. 24. Mr. de Juhász was mainly concerned with pumping losses during the inlet stroke, and with the reduction of efficiency due to this cause when running at light loads.

After dealing with acoustic phenomena which are referred to later, he proceeds to show how the residual pressure in the cylinder head affects the volumetric efficiency. He concludes that "the difference as manifested

* See The Automobile Engineer, Oct. 1926.
** See The Automobile Engineer, Sept. 1925.

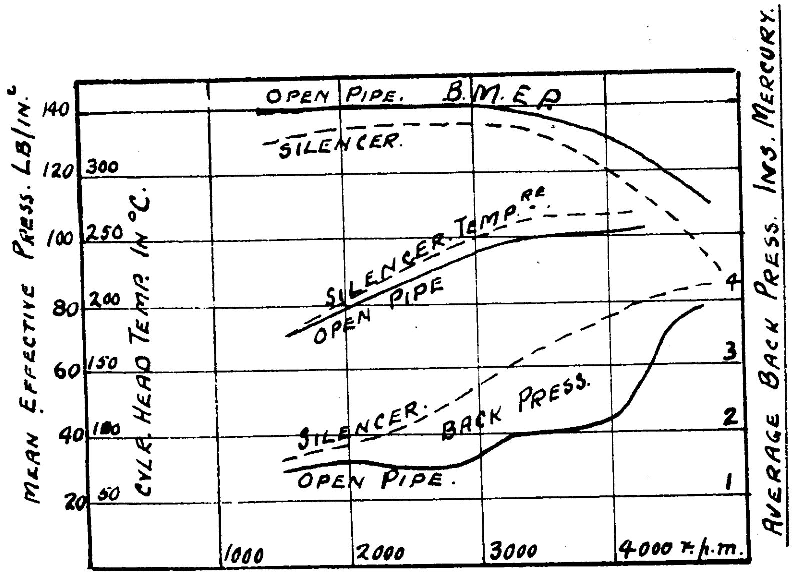
OPEN PIPE.
B.M.E.P.
SILENCER.
SILENCER. TEMPRE.
OPEN PIPE
SILENCER.
BACK PRESS.
OPEN PIPE.
MEAN EFFECTIVE PRESS. LB/IN.²
CYLR. HEAD TEMP. IN °C.
AVERAGE BACK PRESS. INS. MERCURY.
140
120
100
80
60
40
20
300
250
200
150
100
50
4
3
2
1
1000
2000
3000
4000 r.p.m.

FIG. 25.

curve 'A', Fig.23., he states "The wave motion is very evident and the period is very close to that of a closed end organ pipe of the same length". This curve is similar to the writer's curve for a short open pipe No.31, sheet 9.

In connection with the initial maximum pressure his conclusion - "It is noticeable that the shock at the upper end of the pipe becomes greater as the speed increases", is the same as that of the writer's, see pages 41 and 43

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After dealing with acoustic phenomena which are referred to later, he proceeds to show how the residual pressure in the cylinder head affects the volumetric efficiency. He concludes that "the difference as manifested

* See The Automobile Engineer, Oct. 1926.
** See The Automobile Engineer, Sept. 1925.

in these experiments was so slight as to be well within the limits of the test errors. It was therefore impossible to test the actualities of such differences experimentally". Also he concludes that the effect on the pressure in the cylinder during the exhaust stroke of the pressure fluctuations in the exhaust pipe is not sufficient to appreciably affect the power output. In the writer's experiments the pressure differences in the exhaust were of such magnitude that they did appreciably affect the power, see page 36.

*(4) The Elimination of Noise in the Motor Cycle, by Mr. Harold Briggs.

This paper deals with both exhaust and mechanical noises.

Engine tests were made with various exhaust systems, and curves are given showing B.H.P., B.M.E.P., Cylinder Head Temperature, and average back pressure in the exhaust pipe near the valve.

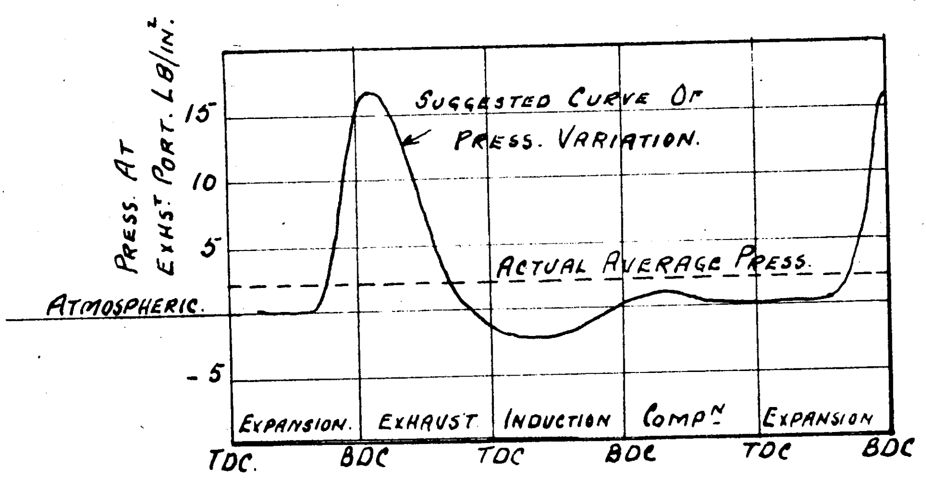
A copy of a set of curves is given in Fig.25. for a 500 c.c. air cooled overhead valve engine.

The silencer was of the carbjector type. There is a slight reduction of power due to the silencer, and an increase in the average back pressure and cylinder head temperature.

In the writer's experiments no exhaust system made any difference to the jacket water temperature. This was probably due to the fact that the cooling was so ample that slight variations in cylinder temperature would have no appreciable effect.

It is, however, a general compkint that silencers do

* The Institute of Automobile Engineers 1926.



PRESS. AT EXHST. PORT. LB/IN.²
15
10
5
-5
SUGGESTED CURVE OF PRESS. VARIATION.
ACTUAL AVERAGE PRESS.
ATMOSPHERIC.
EXPANSION.
EXHAUST
INDUCTION
COMPN.
EXPANSION
TDC.
BDC
TDC
BDC
TDC
BDC

FIG. 26.

cause heating, though it is likely that in many cases the heating is not caused solely by the silencer.

With reference to the supposed 'extractor' effect mentioned on page 44, Briggs states:- "When the exhaust valve opens, the exhaust gas surges down the pipe, and the pressure in the system, after having reached a maximum, is reduced owing to the inertia of the column, which creates an extractor effect". He gives a curve, Fig.26, showing what he supposes to be the pressure variation in the pipe. This curve while bearing a slight resemblance to some of those taken with the writer's Indicator, is quite unlike them in detail.

The opinion that the high velocity exhaust gases have an extractor effect appears to be common among motor engineers.

It is true that the inertia of a very long column of exhaust gas was at one time employed to assist the scavenging of slow speed gas engines. In this case the wave motion was a minimum and the extractor effect a maximum. In high speed engines the effects are reversed, and the wave motion is so intense that even the sudden stoppage of motion of the gas in the pipe due to the closing of the valve has no effect on it.

Briggs does, however, realise that there is wave motion and he states, "The pulsation may give a high or low pressure at the exhaust port at the moment of valve closing. A low pressure means good scavenging, and a purer inlet charge. A high pressure means dilution of the new charge by exhaust gas. This effect is well shown in the torque curve Fig.27.

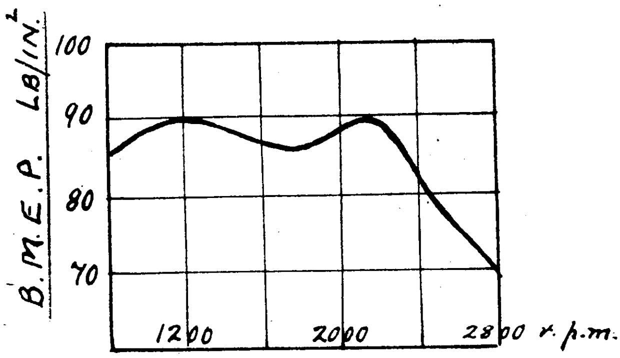


Fig. 2.7.

At 1,100 r.p.m. the column inertia favours volumetric efficiency. An increase in the revolution speed then lowers the torque, which again reaches a maximum at 2,200 r.p.m. The second peak occurs at twice the revolutions of the first, and the same characteristic has been met on a variety of engines."

While the above is partly in agreement with the writer's conclusions it might be pointed out that the most favourable condition (lowest pressure at T.D.C. on the exhaust stroke) does not always recur at multiples of the engine speed. An examination of the writer's curves on sheet 7, shows that the above condition would occur at about 1300 r.p.m. and not again till over 3000 r.p.m. In the discussion on the paper Briggs states with reference to the average back pressures:- "I have quoted these average figures as a matter of interest, and I find that they are of value in comparing different silencers, but it is no use using this method in comparing an open exhaust pipe with a silencer. If two silencers are tried on one engine, the one with the higher back pressure will give a lower power reading". A similar conclusion is arrived at by Mock whose work has already been mentioned "A great number of the types the writer has tested, show the same power loss, for the same degree of silencing, or rather, a power loss in proportion to the degree of silencing".

The same statement was made in a report on the result of the Triumph Motor Cycle Company's Silencer Competition*, "Almost without exception through all the great variety of types, shapes, sizes, and arrangements which have been tested,

* See "The Motor Cycle, Aug. 18th, 1927.

the silence obtained is almost in proportion to the power lost". The above opinions are those most commonly accepted but a different conclusion is given in an *R.A.F. Report on silencing experiments:- "An increase in noise reduction is not necessarily accompanied by an increase in back pressure. It may be accompanied by a decrease".

The connection between back pressure and sound reduction is dealt with by the writer on pages 114-115.

* See T.3. Scientific Research, Report No.9A. Nov. 1917.

SECTION IV.

VELOCITY OF SOUND IN GASES AT HIGH TEMPERATURES.

The expression for the velocity of sound in a gas is

$$V_s = \sqrt{g.\frac{\overline{R}}{m} \times \frac{C_{pm}}{C_{vm}} \times T.} \quad ft/sec. \text{ where}$$

$g = 32.2\ ft/sec^2$; $\overline{R}$ = the universal gas constant which = 1544 for Fah., and 2780 for Cent. degrees.

C_{pm} = the molecular specific heat of the gas at const. pressure,

C_{vm} = " " " " " " " " volume,

m = the molecular weight of the gas,

T = the absolute temperature.

For gas mixtures consisting of w_1, w_2 --- by weight, and v_1, v_2, --- by volume; m_1, and m_2 --- the molecular weights; C_{v_1}, C_{v_2} --- the molecular specific heats; the apparent molecular weight ~~of~~ the mixture is given by

$$M_m = \frac{\sum w}{\sum \frac{w}{m}} \text{ ---- taking the proportions of gases by weight,}$$

$$\text{or } \frac{\sum m}{V_m} \text{ --- " " " " " " volume.}$$

The molecular Specific heat, $C_v = A + S.T + U.T^2$

For a gas mixture of volumes v_1, v_2, ----

$$C_{vm} = \frac{\sum (v_\times A)}{V_m} + \frac{\sum (v_\times S).T}{V_m} + \frac{\sum (v_\times U)_\times T^2}{V_m}.$$

$$C_{vm} = C_{vm} + 1.987.$$

The exhaust gases of a petrol engine using an efficient mixture of petrol & air consist mainly of N_2, CO_2, and H_2O, in the following proportions by volume:- N_2, 0.7382; CO_2, 0.1270; H_2O, 0.1348.

The molecular weights are N_2, 28; CO_2, 44; H_2O, 18.

The molecular weight of the mixture

$$= \frac{\sum m v}{V_m}.$$

$$\therefore M_m = 28 \times 0.7382 + 44 \times 0.1270 + 18 \times 0.1348$$

$$= 20.67 + 5.588 + 2.4264$$

$$= \underline{\underline{28.684}}.$$

The Constants for the specific heat equation are given by Partington and Shilling as:-

Gas	A	S	U
N_2	4·9467	0	$0{\cdot}31 \times 10^{-6}$
CO_2	5·396	$5{\cdot}057 \times 10^{-3}$	$-1{\cdot}02 \times 10^{-6}$
H_2O	7·249	$-2{\cdot}468 \times 10^{-3}$	$2{\cdot}34 \times 10^{-6}$

The average temperature of the gas in the exhaust pipe was found by means of an electric pyrometer to be 850°C at 1000 r.p.m., and 900° to 950°C at 2000 r.p.m.

Taking the temperature as 900°C = 2115° Fah. absolute,

$$\text{the value of } C_{vm} \text{ is} = 5{\cdot}314 + {\cdot}6528 + 1{\cdot}8337$$
$$= 7{\cdot}8005$$
$$C_{pm} = 7{\cdot}8005 + 1{\cdot}987 = 9{\cdot}7875$$
$$= \frac{C_{pm}}{C_{vm}} = \frac{9{\cdot}7875}{7{\cdot}8005} = \underline{\underline{1{\cdot}255}}$$

$$\therefore V_s = \sqrt{\frac{32{\cdot}2 \times 1544 \times 1{\cdot}255 \times 2115}{28{\cdot}684}} = \underline{\underline{2095 \text{ ft./sec.}}}$$

In a *paper by Professor Dixon, Drs. Campbell and Parker, on the velocity of sound in gases at high temperatures, the value of the velocity in nitrogen at 900°C was shown to be 2270 ft./sec. calculated in the above manner. The speed of 2095 ft./sec. obtained for the foregoing exhaust gas mixture therefore appears reasonable. By experiment on the velocity of sound in nitrogen in tubes at the same temperature it was found that the velocity was 2200 ft./sec. This smaller value was reckoned to be due partly to friction in the tubes and partly to the exchange of heat between the gas and the walls of the tube. In general the experimental values were found to be less than the calculated values, and it is to be expected that the actual velocity in the exhaust pipe will be less than the calculated.

* Proceedings of the Royal Society, 1922.

VELOCITY OF SOUND DEDUCED FROM PRESSURE CURVES.

If the wave motion shown by the pressure curves corresponds to that in an organ pipe open at one end and closed at the other (the condition for an open exhaust pipe after the valve is shut) the wave length should be four times the length of the pipe, and the frequency multiplied by the wave length is equal to the velocity of sound.

The frequency for a pipe 5'-2" long, as shown by Curves Nos.20-24, Sheet 7, is about 64 waves/sec. From this value the velocity of sound V_s = 4 (5'-2") x 64 = 1322 ft./sec. Also from Curve No.3, Sheet 2, the frequency is 50 waves/sec. for a pipe 6'-9" long giving V_s = 4 (6'-9") x 50 = 1350 ft./sec.

A rough approximation of the velocity of the wave can be obtained from Curves 39 A and B, Sheet 11. In this case a long pipe was tapped at two points 3'-6" apart so that the time taken for the crest of the first pressure wave to travel 3'-6" was obtained. This was found to be 0·0025 sec. giving the speed of the wave as $\frac{3'\text{-}6}{0\cdot 0025}$ = 1400 ft./sec. This velocity is greater than the true wave velocity owing to the exhaust gas having at this period a fairly high speed in the pipe.

These velocities are very much less than that calculated for a temperature of 900°C, and correspond to a temperature of about 400°C. There may be two reasons for this (a) that the velocity of the high pressure waves recorded by the curves is less than that of normal sound waves, (b) that the actual temperature of the gases at the time during which the waves were recorded was much less than the average temperature as measured.

The temperature of the gases in the pipe after the exhaust valve has closed will certainly be less than the average temperature. This could be ascertained experimentally by making a device for inserting the pyrometer in the pipe during this period of the cycle and removing it during the exhaust stroke. Some idea of the temperature can, however, be got by calculating the highest temperature at the beginning of the exhaust discharge. Knowing the upper limit and the average value the lower value can be got approximately.

Let P_r, V_r, T_r, be the pressure (gauge), volume and temperature of the gas in the cylinder at the point of release on the expansion stroke; and P_1, V_1, T_1, the pressure, volume and temperature near the beginning of the compression stroke. Taking a point where $V_1 = V_r$; P_1 as 15 lb./in.^{2} ; T_1 as 350°C; and P_r as 75 lb./in.^{2} (this value was got from an indicator diagram of a similar engine, see R.A.F. Report, 1922); $T_r = T_1 \times \frac{P_r}{P_1} = 350 \times \frac{75}{15} = 1750°C$.

If T_p = the temperature in the exhaust pipe where the pressure is 5 lb./in.^{2} over atmospheric,

$$\frac{T_r}{T_p} = \left(\frac{P_r}{P_p}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{75}{20}\right)^{\frac{0.255}{1.255}} = 1.308$$

$$\therefore \quad T_p = \frac{1750}{1.308} = 1340°C,$$

the average temperature $900°C = \dfrac{1340 + \text{lower temp.}}{2.}$

the lower temperature = 460°C.

This method would be correct only if the high and low temperatures existed over the same length of time, which is not the case. The high temperature period is much the shorter of the two, consequently the temperature during the longer period must be greater than that already calculated, namely 460°C.

The average temperature in a short pipe might be expected to be higher than in a long one and should show a difference in the wave velocity. This is not so in the case of pipes 3'-4", and 1'-7" long, giving Curves Nos. 29 and 30, Sheet 9. The frequencies of the waves as shown by these curves are 107 and 212, giving wave velocities of 1330 and 1340 ft./sec., respectively.

These low velocities cannot apparently be altogether accounted for by low temperatures, and the general evidence tends to support the conclusion that high pressure sound waves in pipes travel with a velocity lower than that of normal sound waves under the same temperature conditions. *A few experiments by Mr. A.L. Foley on the photographing of sound waves in pipes led him to conclude that, "The speed of sound in tubes can not be represented by the Helmholtz equation or by any other equation which is independent of intensity".

A brief consideration of the standard theory of the velocity of sound shows why high pressure waves cannot be treated in quite the same way as normal ones:- According to Lamb.

"Let Δ = the ratio of $\frac{\text{increment of volume}}{\text{original volume}}$

$$= \frac{v - v_0}{v_0}$$

$$\text{or } v = v_0(1 + \Delta) \qquad (1)$$

* See, The Speed of Sound Pulses in Pipes, Physical Review, 1919.

** See, Lamb's Dynamical Theory of Sound.

Let S = the condensation = $\dfrac{\text{increment of density}}{\text{original density}}$

$$= \frac{\rho - \rho_0}{\rho_0} \quad \text{or} \quad \rho = \rho_0(1+S) \text{ ---- } (2)$$

Since $v = \dfrac{1}{\rho}$ $\therefore$ $(1+S)(1+\Delta) = 1$ ···· (3)

"Considering the motion of air in a pipe the stratum originally bounded by the planes x and $x+\delta x$ at t_0 is at time t bounded by $(x+\xi)$ and $(x+\xi+\delta x+\delta\xi)$, and its thickness is $\therefore$ changed from δx to $(\delta x+\delta\xi)$ or $\left(1+\dfrac{\delta\xi}{\delta x}\right)\delta x$, and the dilatation is therefore

$$\Delta = \frac{\delta\xi}{\delta x} \text{ ---- } (4)$$

Hence in the case of infinitely small disturbances we have by (3)

$$S = -\Delta = -\frac{\delta\xi}{\delta x} \text{ ------ } (5)$$

"In forming equations of motion we assume that the pressure varies with the density, $\therefore$ for small values of S $p = p_0 + KS$ --- (6)

The acceleration of momentum of unit area of stratum originally bounded by planes x and $(x+\delta x)$ is $\rho_0\,\delta x\,\ddot{\xi} = -\delta p$ where δp represents the excess of pressure on the anterior face.

Hence by (5) and (6) $\dfrac{\delta^2\xi}{\delta t^2} = \dfrac{C^2\delta^2\xi}{\delta x^2}$ ----- (7).

where $C = \sqrt{\dfrac{K}{\rho_0}}$

But for adiabatic expansion

$$\frac{p}{p_0} = \left(\frac{\rho}{\rho_0}\right)^\gamma$$

This for small values of S becomes

$$p = p_0(1+\gamma S) \quad \therefore \quad K = \gamma p_0$$

$\therefore$ the velocity of sound = $C = \sqrt{\dfrac{\gamma p_0}{\rho_0}}$. "

It is to be noted that in the above treatment δ and Δ have been assumed to be so small that the product $\delta\Delta$ as in equation (3) may be neglected, i.e. the true value of from (3) is $-\Delta - \delta\Delta$, and in equation (5) δ is made $= -\Delta$

The assumption is justified in the general case of sound waves in which the compression is relatively small, but it is not justified when the compression is as great as in the case of waves in an exhaust pipe.

As far as the writer is aware a full theoretical treatment of the subject has not yet been given.

In Mr. de Juhász's experiments (see page 63) the exhaust pressures were taken inside the cylinder, and with regard to the wave frequencies he states "The cylinder and the attached exhaust pipe may be regarded as an acoustic pipe closed at one end, the effective length of which varies with the motion of the piston. The mean volume of the cylinder is the compression volume + half the swept volume, or 44 in.^{3} This corresponds in volume to an exhaust pipe 23 ins. long, and this figure has been used in the following calculations."

The above method of correcting the pipe length for the cylinder volume does not seem to the writer to be quite accurate especially when the pipe is short. The writer's diagrams show the periodic motion in the pipe during the exhaust stroke and after the valve is shut. Applying the correction for the cylinder volume during the exhaust stroke in the cases of Curves No.20, Sheet 7, Nos. 29 and 30, Sheet 9, the result shows that this correction is too small for the 5'-2" pipe and too great for the 3'-4" and 1'-7" pipes. The length of pipe used by de Juhász was 5'-7" for which the correction was probably near enough, and the agreement between the calculated values of the periodic time and those obtained from the diagrams is fairly good. His values also show that the frequency increases with the temperature. He concludes as follows, "These experiments substantially corroborate the general application of the laws of acoustics to exhaust phenomena."

Referring to the diagrams Fig 22. on page 62 taken by Dr. Watson, Mr. Mock in his paper (already mentioned), states "The shorter waves have exactly the period of a sound wave in a two-foot closed-end pipe, while the large wave motion, on which the smaller waves are superimposed, bears probably the period of the whole system." Mr. Mock also states that the frequency of the wave shown at A on Fig. 23, corresponds to the calculated frequency for the pipe, but in neither case did he know the exhaust gas temperature so his deductions are to be taken with reserve.

It would appear, therefore, that the wave motion in an exhaust pipe approximates to that of ordinary acoustic phenomena, but the question of the velocity of high pressure sound pulses in pipes requires a more thorough investigation before any reliable statement can be made about it.

ANALYSIS OF EXHAUST NOISE.

As shown later the silencing of a note depends to some extent on its wave length. It is therefore necessary to know something of the separate sounds that make up an exhaust noise.

In the case of a single cylinder engine with an open exhaust pipe the fundamental note of the pipe can be heard quite plainly at small throttle openings. With a little more throttle the note rises to the overtones of the pipe, and a harsh element of much higher pitch begins to appear. This latter can be eliminated to some extent by making the exhaust pipe of soft metal. Before silencing regulations came into force motor cyclists sometimes fitted large diameter open pipes of annealed copper which gave a mellow, and a really pleasant, musical note when the engine was running light. With half or full throttle the harsh element increases so much that the musical effect is quite lost, and the total sound can be resolved into two distinct
* components. These are a heavy 'impact', or 'bang', and a harsh, high pitched, 'hiss'. The presence of intense short waves can be clearly shown by holding a plate of hard material across the axis of the pipe at the opening - fig (28).

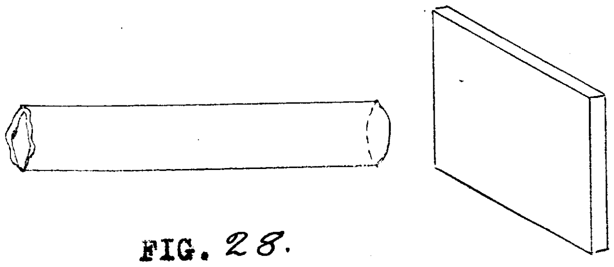


FIG. 28.

The space between the plate and the pipe forms a resonator for the short waves which are thus intensified and make a deafening noise. This effect is a maximum when the distance between the plate and the pipe is about 3 ins., and ceases when the distance is less than 1¼ins. or/

* See also Appendix

or more than 9 ins.

These two elements in exhaust noise - the 'bang' and 'hiss' - do not appear to be generally recognised. The writer has found only two references to them, one in an article* on the Maxim Silencer, - "To the ear, there are two kinds of noises - intermittent as in the case of the internal combustion engine or a reciprocating compressor and steady as in the case of rotary blowers or continuous air or steam discharges. Then there is the entirely different sensation of pulsation with little or no noise; reciprocating compressors often produce this sensation. These pulsations, from the standpoint of the silencer, come under the same category as noises, in that they are subject to the same law of behaviour as ordinary audible sounds. Each noise has particular variations and the psychological effects differ. For instance, a gas or oil engine has two distinct noise elements, the thumping, booming air shock coming from the sudden release of a charge of high pressure gas and the rushing ~~an~~ or high velocity noise caused by high exhaust gas velocity at any point in the exhaust system. These two vary with the particular type of engine."

The other reference is by Mr. Mock in the paper already quoted, page (62), who states regarding exhaust pressure effects:- "It is quite evident from the foregoing what the duty of the muffler must be and what conditions it is called upon to meet. Nevertheless the question may be raised whether the wave shown on the diagrams is the wave which, spreading from the exhaust pipe/

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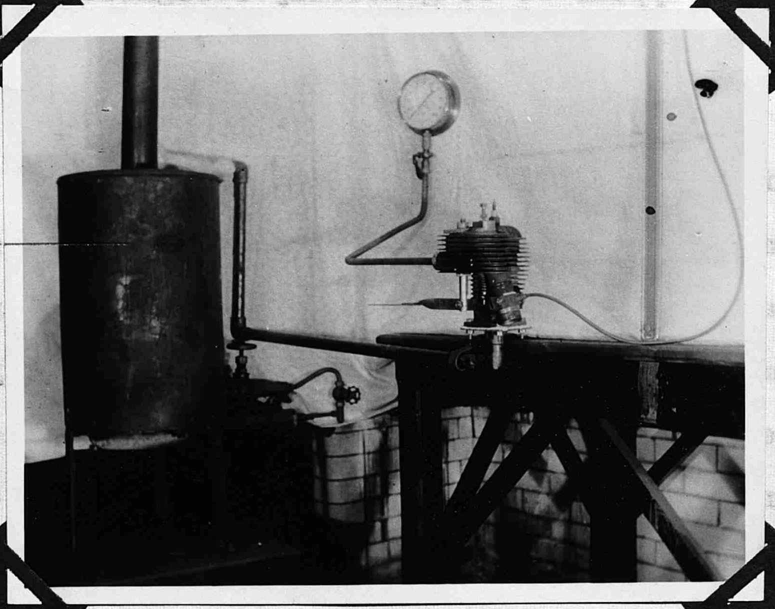
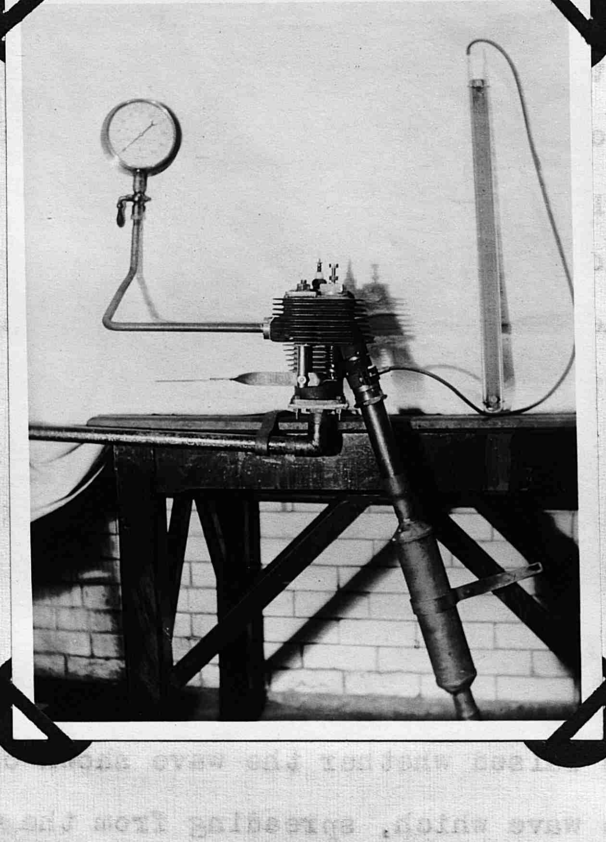
* Power plant Engineering, Nov. 1st. 1926.

AIR HEATER





pipe outlet, affects the auditory organs, or whether it is any other sort of wave, possibly of more common size, superimposed upon the first one. I find that a sharp noise, like the ticking of a watch, is very distinctly heard through a muffler which would silence the exhaust very completely. It sounded more distinct as if heard through a length of pipe. This is also true of a sustained vibration, such as a musical note. A succession of modulated tones, as from speaking, is made slightly indistinct and hollow. From these considerations, and from the very great intensity of the exhaust sound, as evidenced from the distance which it carries, and the vigor with which it is reflected, I conclude that the discharge pulse is the cause of the impression our ears receive". He evidently considers the 'hiss' to be of relatively small importance, but the writer's observations do not bear this out.

As the 'bang' and the 'hiss' occur simultaneously they cannot be examined properly under working conditions, and the only way to deal with them is to produce them separately. This was done as follows:-

INVESTIGATION OF 'HISS'.

An old motor cycle engine cylinder of the same capacity as the one on the J.A.P. engine was obtained and fitted up as shown in Drawing No.12, and in the photographs. The inlet valve and guide were removed and a bolt (1) was inserted to carry the fulcrum pin of a lever (2) by means of which the exhaust valve could be easily raised. A weak valve spring was used. The lift of the exhaust valve could be regulated by a stop (3). An offtake (4) from the exhaust stub led to a manometer for registering pressures in the pipe. It could also be used for "listening in" to the noise in the pipe. Compressed air was/

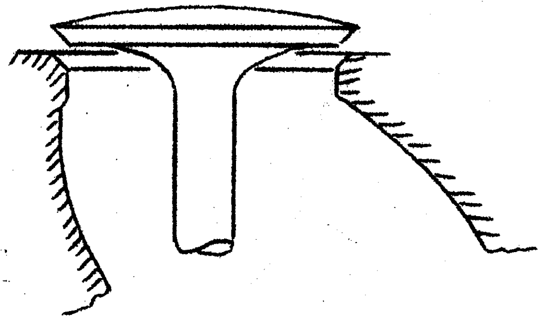


FIG. 29.

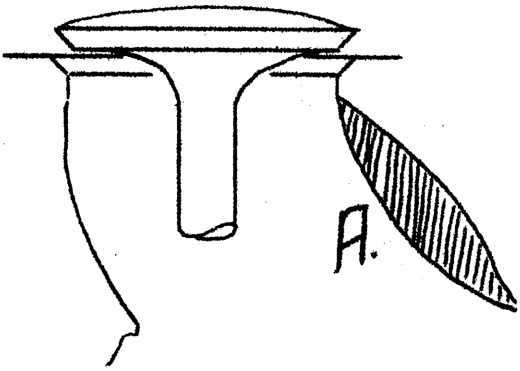


FIG. 30.

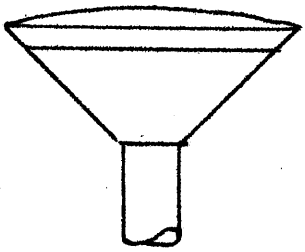


FIG. 31.

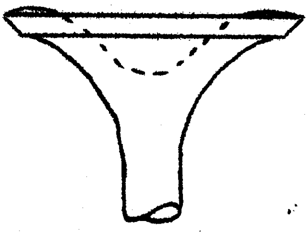


FIG. 32.

was supplied from an air reservoir through an air heater. A pressure gauge was connected to the cylinder through the inlet stud. A continuous noise could be produced by holding the exhaust valve partly open and allowing the compressed air to pass through at constant pressure, or the valve could be opened suddenly in which case the pressure in the cylinder dropped at once and there was just one loud hiss. It was not possible to open the valve by hand quickly enough to produce the 'bang' that takes place on the engine. The 'hiss' could therefore be studied by itself. The effect of heating the air before passing it through the cylinder was tried but did not make much difference. It was a rather troublesome scheme, so the air was used cold.

The question now is - what kind of sound waves make up a hiss? It is evident that there are waves of differing frequencies which do not combine to form a musical note; also the average frequency is very high. It was found that the hiss was present only when the exhaust valve was partly open, and almost ceased when the rise of the valve was 3/16", the total rise being 5/16". The intensity of the noise was a maximum at a rise of about 1/8". It then appeared to consist of a mixture of very shrill whistles which are very fatiguing to the ear, rendering the writer almost deaf for hours after an experiment.

An examination of the air passage when the valve is 1/8" open shows the possibilty of whistle being produced. Taking a small part as in Fig 29, the opening is a nozzle from which the air expands into a chamber which can form a resonator for the high frequency waves produced at the nozzle.

Some experiments were made in varying the shape of the valve/

valve pocket and valve, e.g. packing the pochet at 'A' as indicated by the section lines, Fig.30, gave a slightly decreased noise. The valve as in Fig.31 increased the hissing period; the 'hiss' did not cease after the valve was raised 3/16 ins., as in the case of the standard valve, but continued till the valve was full open. Valves are not made of this exact shape, but there is a type very similar called the "tulip" valve, Fig.32, which would possibly increase the period of the hiss.

No matter how smooth the valve passage or nozzle opening is made there is always a loud 'hiss' when high pressure gas is escaping; but this noise is greatly increased if the shape of the passage forms a resonator for even a small part of the sound. Consequently while it is impossible to prevent a fairly loud 'hiss' it might be possible so to shape the valve pocket that resonance would not occur. A considerable amount of experimental work might be necessary before the best shape could be determined.

INVESTIGATION OF 'BANG'.

'Bang' is the term generally used to signify a single sound pulse such as is given by the sudden shutting of a door, the striking of a wooden table with a hammers, the firing of a rifle, etc. It differs from a 'booming' sound which has a continuing quality and may be a musical note, and from a 'crash', e.g. the fall of a tree, in which case there is a series of noises extending over an appreciable interval of time. The cause of the 'bang' in the case of the engine exhaust is most easily understood by considering the analogus case of wave motion in a helical spring; Fig.35.

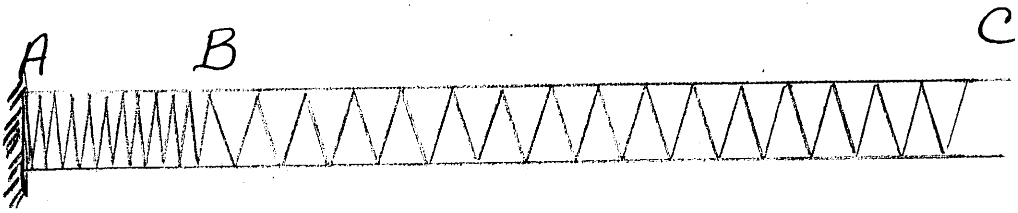


FIG. 33.

The part AB is compressed and BC is free or it may be under a compression less than that in AB. Suppose the spring at B to be suddenly released. The end B is accelerated to the right until the parts immediately to the right and left of B are in a state of equal compression, e.g.

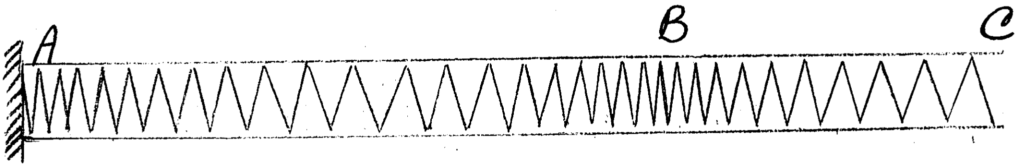


FIG. 34

The part at B is then the crest of a compression wave which proceeds towards C.

One important fact to note is that no matter how small may be the difference in compression between AB and BC, the sudden release of B will produce wave motion.

If instead of a spring there is a tube of compressed gas with a valve at B a wave motion will be set up in the gas in exactly the same way, provided that the valve is opened with sufficient rapidity. There is no necessity for the difference in gas pressure between AB and BC to be great enough to give a gas velocity along the pipe equal to that of sound.

In the engine the part AB is replaced by the cylinder, the valve at B is usually smaller in area than the pipe BC, and the raising of the valve is not instantaneous. But even when the engine is running quite slowly the valve leaves its seat so suddenly that the first period of its rise can be thought of as instantaneous. A wave is therefore formed at any speed or pressure, though at/

at low speeds and pressures the wave intensity is negligible.

If the engine speed is constant the wave intensity will increase with the difference between the pressure in the cylinder and the pipe until this difference is sufficient to give the critical gas velocity at the valve. The wave pressure clearly cannot rise above its value at this point no matter how high the cylinder pressure may be.

If the pressure in the cylinder remains constant (above that giving the critical gas velocity) and if the engine speed increases the wave intensity can increase to a maximum. This occurs when the speed of the valve lift is such that the gas velocity through the opening is still equal to its critical value when the valve is full open.

The curve of intensity on a base of speed, Fig.39. page, shows that this limit would be reached for the J.A.P. engine at between 4000 and 5000 r.p.m. It is actually reached before 4000 r.p.m. owing to the decrease in volumetric efficiency.

The best method of producing a really loud 'bang' similar to that on the engine was found to be that of 'firing' a pop-gun into the end of a long exhaust pipe, e.g. Fig.35.

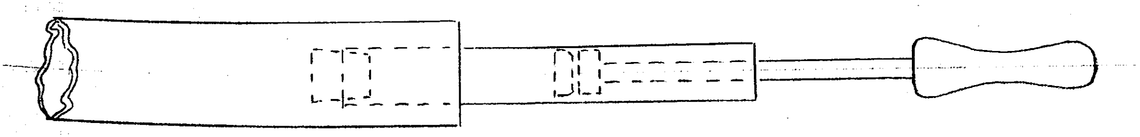
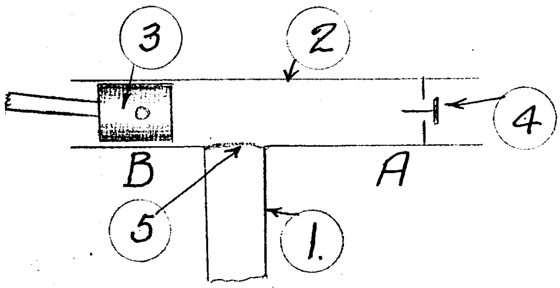


FIG.35

The pressure of the air in the pop-gun is not high but the cork leaves the muzzle so quickly that the impact of the compressed air on the air in the pipe produces a sharp crack.

As this arrangement was inconvenient, another method/

method was adopted - that of suddenly decreasing the pressure at the end of the pipe. This can be done in the following way, Fig.36. The pipe (1) is joined to the cylinder (2) in which a piston (3) moves rapidly backwards and forwards from position 'A' to position 'B'. When the piston moves towards



'A' the air is expelled through the non-return valve (4), so that on returning towards 'B' it creates a partial vacuum at 'A'. When the port (5) to the pipe (1) is uncovered air enters from pipe (1), and the sudden decrease in pressure followed by the impact of the air rushing in starts the column of air in the pipe vibrating. If the cylinder is small the movement of the air does not make an appreciable 'hiss'. The device used in the experiments is shown on Drawing No.13. It was made largely from parts of apparatus that had been used for other purposes. The crankshaft was driven by a belt from a motor at any desired speed. In order to have sudden opening of the ports in the cylinder the piston was arranged to uncover them when at its highest speed.

A series of 'bangs' similar to that of a small two-stroke engine running light was obtained. These 'bangs' were not so loud as in the case of the J.A.P. engine at full throttle but were loud enough to give an idea of the comparative effect of the various silencers.

The behaviour of the silencers on the separate 'hiss' and 'bang' tests is discussed later on pages 111 -

POPULAR IDEAS OF EXHAUST NOISE.

A great deal has been written in the motoring press on the subject of exhaust noise and silencing.

One/

One of the most widely held beliefs is that the noise is produced at the outlet of the pipe by the sudden expansion of high temperature, high pressure, gas into the atmosphere in much the same way as the report of a gun. In the case of the gun the gas in the barrel is at an extremely high pressure and expands rapidly into the atmosphere generating an intense sound wave over a comparatively large area. It is the latter fact which accounts for the loudness of the report of a rifle having such a small bore as 0.22 ins.

The difference between the two cases, then, is that the gun report is produced outside the barrel and the exhaust noise is produced inside the exhaust pipe. This does not prevent the 'bang' of the exhaust being similar in character to the report of a gun, but it does make a difference to the way in which the two noises are silenced.

In the Maxim gun-silencer there is first a chamber in which the very high pressure gases can expand nearly to atmospheric pressure after which they pass to the atmosphere through spiral passages. The action of the silencer is largely to delay the expansion of the high pressure gases. This does no harm to the gun, but any delay in the expansion of exhaust gases is fatal to the working of an engine.

The designers of most patent silencers make some statement to the effect that their silencers allow gradual expansion, e.g. "The long circuitous route by which the exhaust gases are made to travel through the Vortex Silencer allows them more time for cooling and expansion, so that when they are at last transmitted to the outside atmosphere, it is at practically normal atmospheric/

atmospheric pressure. There is no noisy impact, no resistance."

The designers' efforts appear in many cases to be expended in a wrong direction. They attempt to provide for gradual expansion of the gas, when the expansion has already taken place at the exhaust valve and caused the sound that has to be retained or destroyed.

At first sight it appears strange that a silencer designed on demonstrably wrong principles should have any silencing effect at all. The reasons are explained in a later discussion of the behaviour of the silencers which were tested.

EXPERIMENTS ON NOISE INTENSITY.

In the foregoing analysis the nature of the Exhaust Noise was dealt with, but it is advantageous to have a more quantitative knowledge of the intensity of the noise.

The curves of the pressure inside the exhaust pipe do not give a true idea of the intensity of the waves when they have escaped into the atmosphere and become reduced to the pressure at which they strike the ear.

The following experiments were made in order to examine the sound waves at the lower pressures:-

As the multi-point Indicator did not work well with very low pressures it was necessary to have the sound waves at a slightly higher pressure than they would have been in free air. The end of the exhaust pipe was therefore attached to a long cylinder of light sheet steel from the middle of which a short pipe led to the Indicator, Fig. 37.

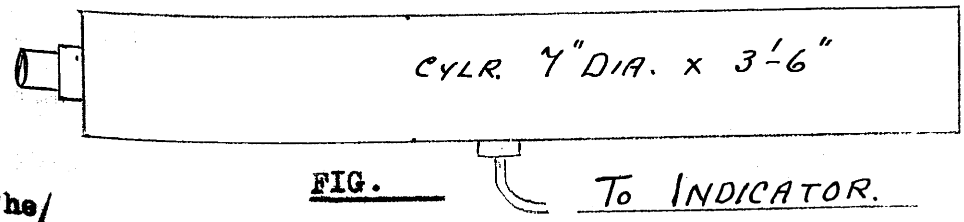
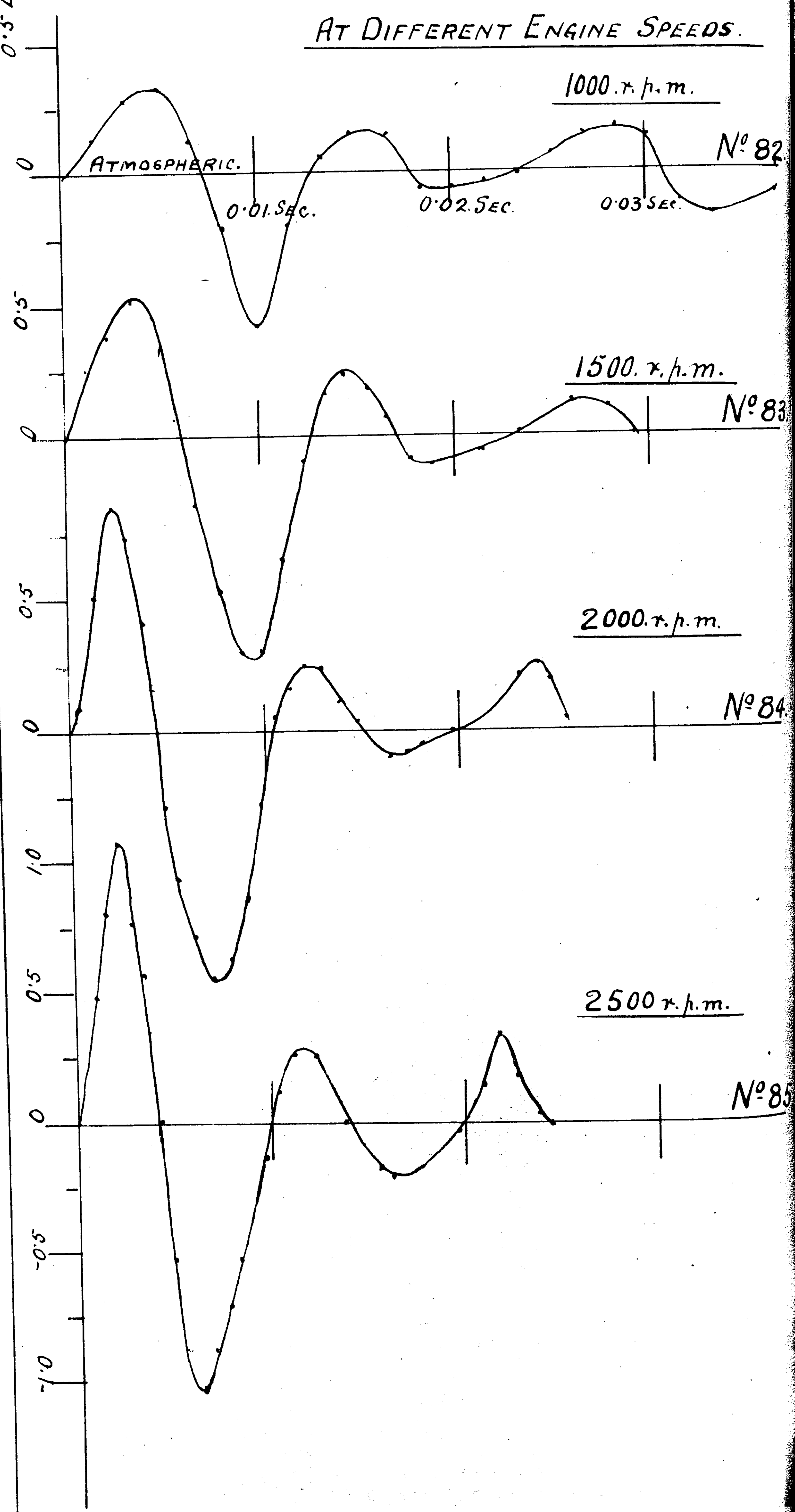


FIG.

The/

FORM OF SOUND WAVES FROM AN OPEN PIPE AT DIFFERENT ENGINE SPEEDS.



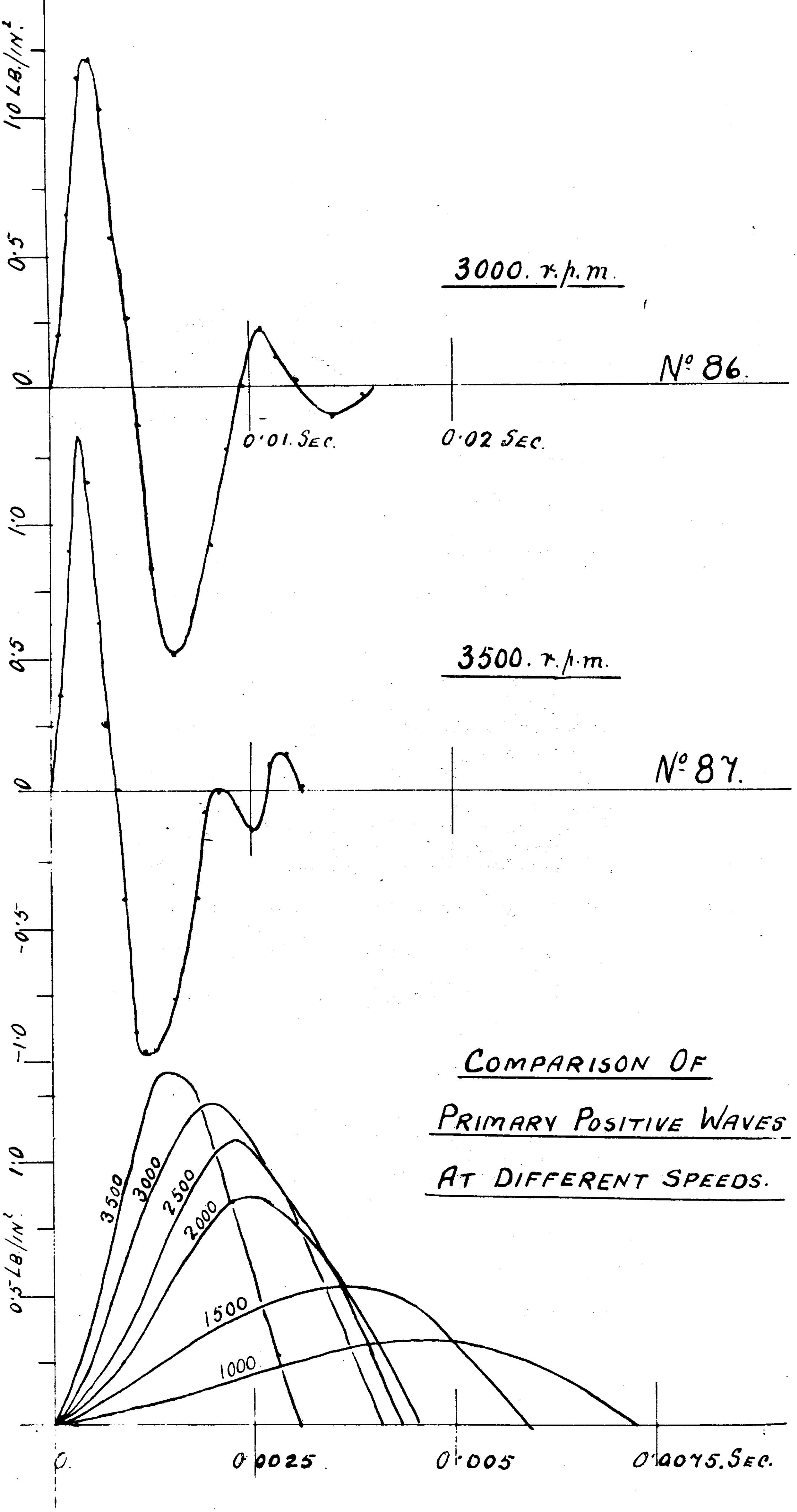

1.0 LB./IN.²
0.5
0
3000. r.p.m.
Nº 86.
0·01. Sec.
0·02 Sec.
1.0
0.5
3500. r.p.m.
0
Nº 87.
-0.5
-1.0
Comparison Of
Primary Positive Waves
At Different Speeds.
1.0
0.5 LB/IN.²
3500
3000
2500
2000
1500
1000
0.
0·0025
0·005
0·0075. Sec.

Fig. 38.

The period of the sound waves is modified to some extent by the length of this cylinder but this does not affect the important features of the results.

FIRST SERIES OF EXPERIMENTS: (Pipe to Cylinder).

The object of this series was to find how the intensity of the sound varied with engine speed, other factors being constant. For the purpose of comparison all the curves are plotted on a base of time to the same scale, and this applies to the other series also.

Curves Nos.82 to 87, are for speeds of 1000 to 3500 r.p.m.

The points to be noted as the speed increases are as follows:- (a) the positive and negative parts of the primary waves increase in intensity, (b) decrease in time, (c) the remaining waves become relatively unimportant. This last point (c) shows why the noise is heard as one 'bang' per cycle instead of as a more or less continuous moise which might be expected from the curves of pressure taken inside the pipe where the waves do not die out.

In Fig.38 the primary positive waves are plotted to a more convenient time scale. It would be interesting to know whether the ear estimates the loudness of a sound purely by the maximum pressure of a wave or by a combination of the maximum pressure with the time taken to attain it. It is probable that the time factor determines the "sharpness" of a sound. It was just possible to put one's ear to the cylinder and stand the full pressure of the sound at 1000 r.p.m., but at higher speeds this could not be done.

Curve ABCDEF, Fig.39 shows the variation of intensity of the primary positive wave on a base of engine speed, and curve A'B'C'D'E'F' shows the variation of/

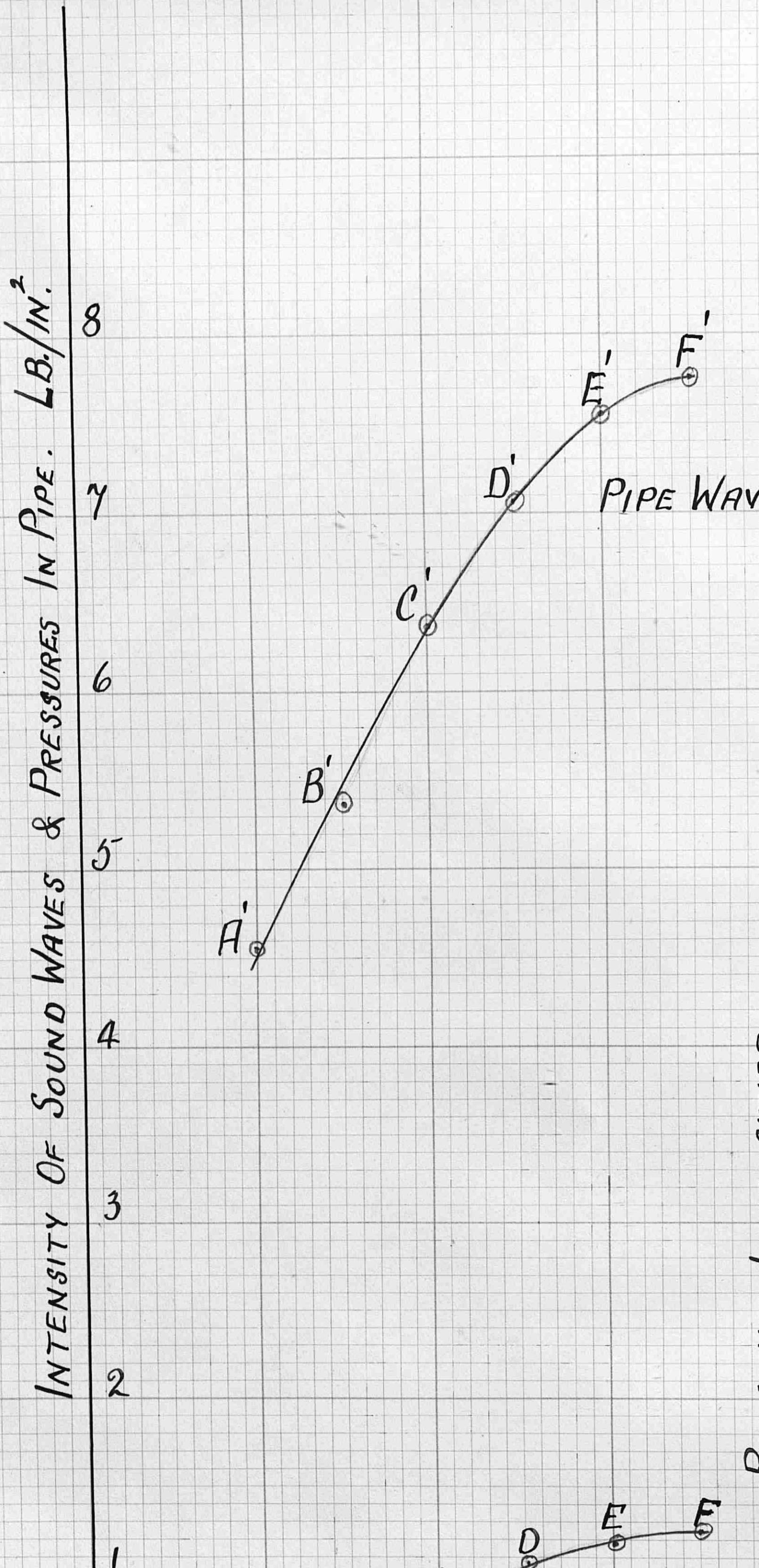

Intensity Of Sound Waves & Pressures In Pipe. LB./IN.²
Pipe Wave Pressures.
Sound Wave Pressures.
A' B' C' D' E' F'
A B C D E F
0 1 2 3 4 5 6 7 8
0 1000 2000 3000
revs. per min.

Fig. 39.

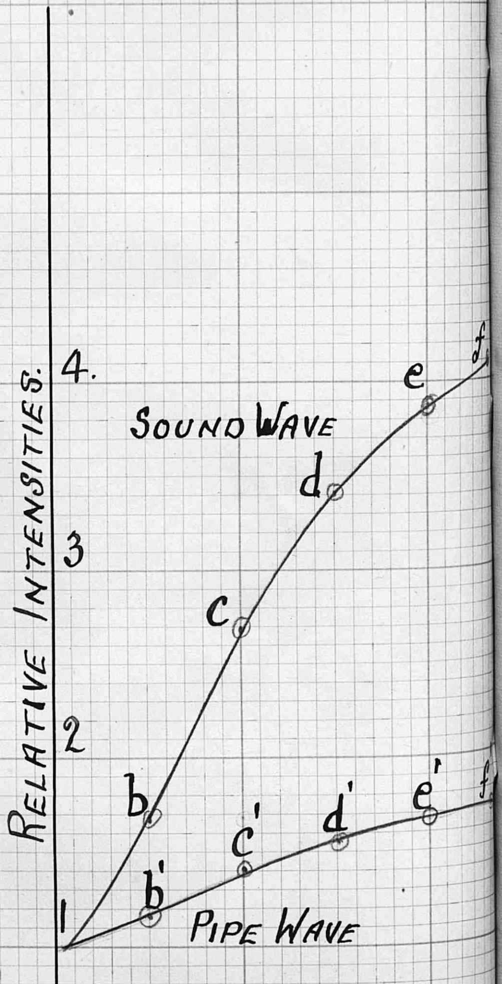

Relative Intensities.
Sound Wave
Pipe Wave
b c d e f
b' c' d' e' f'
1 2 3 4.
1000 2000 3000
revs. per min

Fig. 40.

of the initial maximum pressure in the pipe for corresponding speeds. It might be expected that the intensity of the sound waves would increase in the same ratio as the pressures in the exhaust pipe, but this does not seem to be the case. Curve bcdef, Fig.40 is a curve of relative intensities, i.e. the ordinate of 'b' is got by dividing the ordinate at B, Fig.39. by the ordinate at A; the ordinate at 'c' is the ordinate at C divided by that at A, and so on. Curve b'c'd'e'f' is obtained from curve A'B'C'D'E'F' in the same way.

The relative intensity of the sound waves increases much more rapidly with engine speed than that of the waves in the pipe at the engine end.

The writer has not, so far, been able to account for this phenomenon.

One thing both curves have in common is that they seem to be reaching a limiting value.

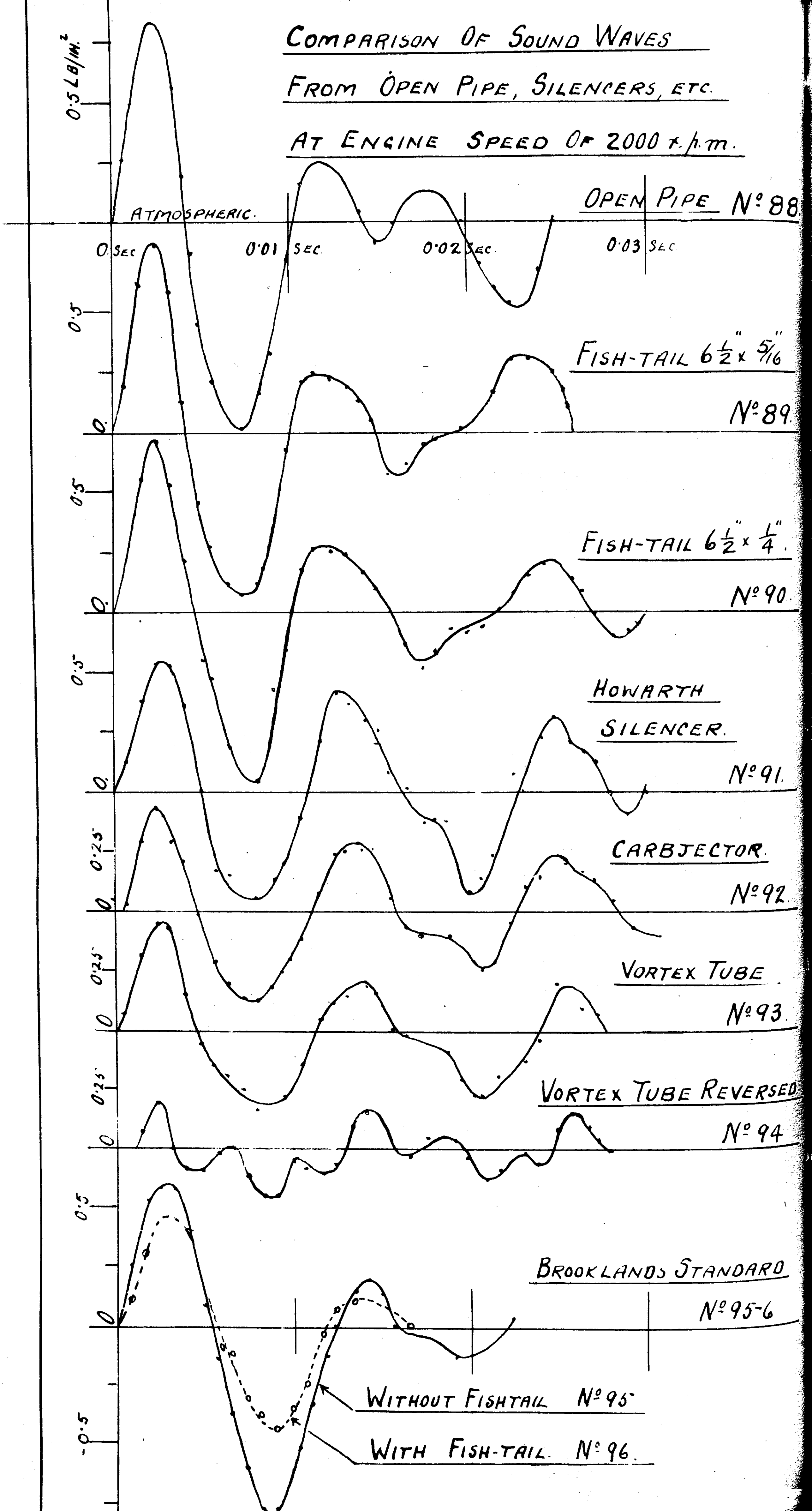
This should be the case for the pressures in the pipe because these are determined by the rate of flow of the gases through the valve port. This rate cannot be greater than the critical velocity so that the pressure in the pipe reaches a limit when the engine speed and consequently the rate of opening of the valve becomes very high.

The intensity of the noise at 3500 r.p.m. as shown by Fig.38 is four times that at 1000 r.p.m. but it would be impossible to make any estimate of this relative intensity simply by listening to the noise as the engine speeds up.

SECOND SERIES OF EXPERIMENTS (Silencers to Cylinder).

In this series various silencers were tested. (For details of the silencers see drawings Nos.10 and 11).

Curve/



COMPARISON OF SOUND WAVES
FROM OPEN PIPE, SILENCERS, ETC.
AT ENGINE SPEED OF 2000 r.p.m.
0·5 LB/IN.²
ATMOSPHERIC.
0 SEC
0·01 SEC.
0·02 SEC.
0·03 SEC.
OPEN PIPE Nº 88.
0·5
0.
FISH-TAIL 6½" x 5/16"
Nº 89.
0·5
0.
FISH-TAIL 6½" x ¼".
Nº 90.
0·5
0.
HOWARTH
SILENCER.
Nº 91.
0·25
0.
CARBJECTOR.
Nº 92.
0·25
0
VORTEX TUBE.
Nº 93.
0·25
0
VORTEX TUBE REVERSED
Nº 94
0·5
0
-0·5
BROOKLANDS STANDARD
Nº 95-6
WITHOUT FISHTAIL Nº 95
WITH FISH-TAIL Nº 96.

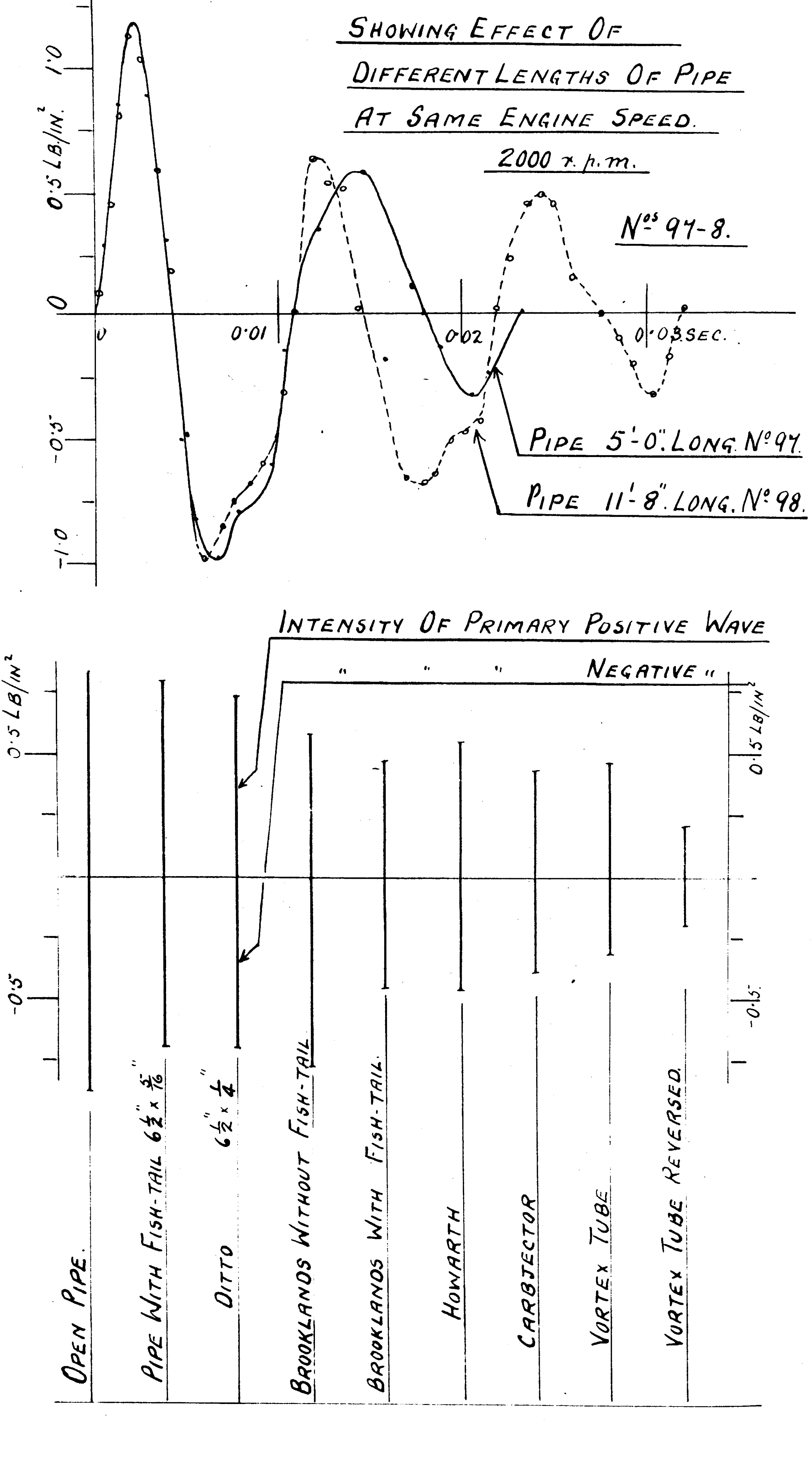

Showing Effect Of
Different Lengths Of Pipe
At Same Engine Speed.
2000 r.p.m.
Nos 97-8.
0·5 LB/IN²
1·0
0
-0·5
-1·0
0
0·01
0·02
0·03 SEC.
Pipe 5'-0". Long. No 97.
Pipe 11'-8". Long. No 98.
Intensity Of Primary Positive Wave
" " " Negative "
0·5 LB/IN²
-0·5
Open Pipe.
Pipe With Fish-Tail 6½" × 5/16"
Ditto 6½" × ¼"
Brooklands Without Fish-Tail
Brooklands With Fish-Tail.
Howarth
Carbjector
Vortex Tube
Vortex Tube Reversed.

Fig. 43.

Curve No.88 for an open pipe was first obtained as in the previous experiment. The fish-tails had to be put inside the large cylinder, i.e. Fig.41

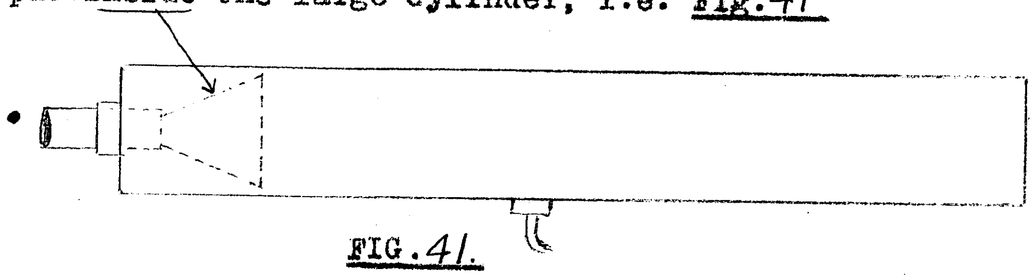


FIG.41.

The silencers were coupled to the end of the pipe and the outlet inserted in the end of the cylinder, Fig. .

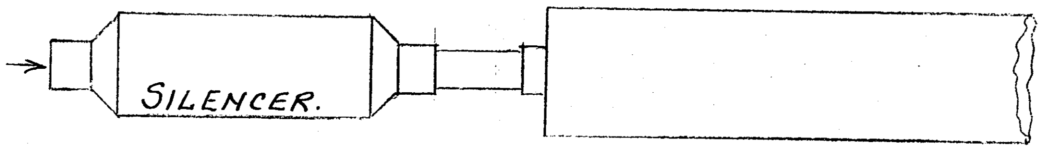


FIG.42.

The first notable fact about these curves Nos.88 to 96 is that although the amplitude varies considerably the period remains nearly constant. This result is different from that obtained from previous curves for free vibration in an open pipe in which case the period increased as the amplitude decreased.

The reason for the constancy of the period is that the period of the wave in the pipe is constant and though a silencer prevents the full pressure of the wave from getting out it does not alter the time taken for the passage of the wave.

A comparison of the intensities of the primary waves for the different silencers is given in Fig.43. The fish-tails attached directly to the pipe make very little difference.

When fish-tails began to be used (about 1922) the writer made some experiments on the size of outlet and found that a properly shaped fish-tail with an outlet equal in area to the bore of the pipe had very little effect on the noise. Only when the outlet area was rather/

rather less than that of the pipe was there appreciable silencing. Any kind of fish-tail or flattened end of a pipe does, however, have the effect of stopping the 'booming' quality of the sound even when the intensity is not much reduced.

A fish-tail on the outlet of a large volume silencer such as the Brooklands reduces the sound to a very pronounced degree. This is probably because the wave pressure is reduced in the silencer and the fish-tail can therefore reflect the waves more easily.

It is often stated that a long pipe gives a quieter exhaust than a short one, so two lengths of pipe were tested at the same engine speed, see curves Nos. 97-8. The primary waves are identical and the succeeding waves are more pronounced in the case of the longer pipe. Increase of pipe length, therefore, within the limits as tested does not reduce the wave intensity.

It is, however, the case that a long exhaust pipe of small diameter as fitted to some cars does reduce the noise. The deposit of soot and oil inside a small pipe has a considerable damping action, especially on the 'hiss'. This damping would not occur in the large diameter pipe used in the writer's experiments.

ACTUAL AND APPARENT INTENSITIES:-

The actual intensity of the explosive 'bang' is shown by the Indicator curves but the high frequency notes of the 'hiss' are not recorded. Direct measurement of the 'hiss' is not possible with the present apparatus and the estimation of its intensity must be made by the ear. This is very unsatisfactory as so much depends on the personal element, but the evidence is quite strong and/

and the following deductions may be made:-

If two different silencers give the same curves for 'bang', and if the noise as apparent to the ear is greater in the case of one than the other the reason for this difference must be that the noisier silencer is allowing more of the 'hiss' to get through. Of the four silencers tested the Howerth shows more 'bang' and yet is distinctly quieter than any of the others. The addition of a fish-tail to the Brooklands decreases the apparent noise to a much greater extent than is shown by the curves. The Vortex Tube when reversed almost eliminates the 'bang' and yet is apparently quite as noisy as when fitted in the normal way. A method of reducing the 'hiss' without affecting the 'bang' was to put a thick felt liner inside the test cylinder (of Fig 37). This made only a slight difference to the Indicator Curve but it reduced the total noise considerably. The conclusions are, therefore, that the 'hiss' accounts for a large proportion of the total exhaust noise, and that silencers which have the same capacity to reflect long, high pressure, waves may have very different capacities for reflecting short waves.

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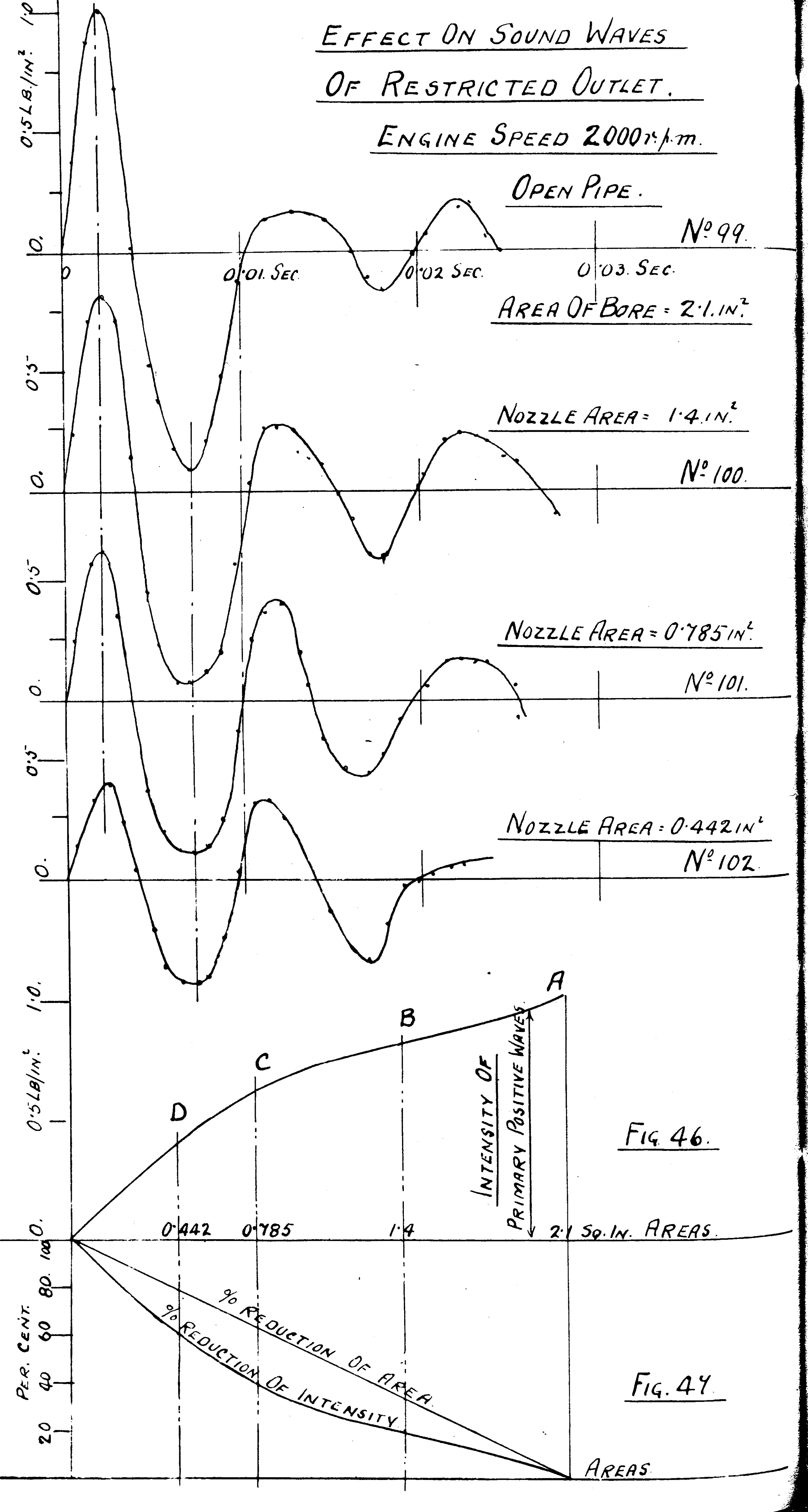
METHODS OF SUPPRESSION OF SOUND.

Before criticising the silencers in detail it will be necessary to discuss the various ways in which sound waves may be suppressed or destroyed.

The methods of production of sound have already been stated, and the condition is that high frequency and low frequency waves are travelling along a pipe together/



EFFECT ON SOUND WAVES
OF RESTRICTED OUTLET.
ENGINE SPEED 2000 r.p.m.
OPEN PIPE.
No 99.
0'01. SEC
0'02 SEC.
0'03. SEC.
AREA OF BORE = 2·1. IN.²
NOZZLE AREA = 1·4. IN.²
No 100.
NOZZLE AREA = 0·785 IN.²
No 101.
NOZZLE AREA = 0·442 IN.²
No 102.
0·5 LB/IN.²
A
B
C
D
INTENSITY OF PRIMARY POSITIVE WAVES
0·442
0·785
1·4
2·1 SQ. IN. AREAS
PER. CENT.
% REDUCTION OF AREA
% REDUCTION OF INTENSITY
AREAS

FIG. 46.

FIG. 47

together with high speed exhaust gas. The flow of the gas must not be unduly impeded, so that the outlet area cannot be decreased sufficiently to keep back a high proportion of the sound. But in most silencers the outlet area is reduced to some extent, and the effect of this reduction was investigated in the following experiments:-

EFFECT OF REDUCED AREA OF OUTLET.

Three nozzles of different diameters were made to fit the end of the exhaust pipe, Fig.45

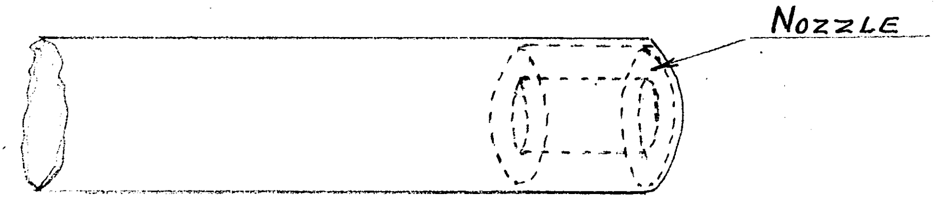


FIG.45.

Nozzle	Open Pipe	(1)	(2)	(3)
Bore of Nozzle	$1\frac{5}{8}$"	$1\frac{5}{16}$"	1"	$\frac{3}{4}$"
Cross Sectional Area	2.1 in^2	1.4 in^2	0.785 in^2	0.44 in^2

Curves Nos.99-102, show that the restricted outlet reduces the amplitude but does not alter the period of the waves.

Curve DCBA, Fig.46. shows that the intensity of the primary positive wave does not decrease in proportion to the area of the outlet. In Fig.47. the reductions of area and the corresponding reductions of intensity are expressed as percentages. Except at the limiting conditions the reduction of intensity is much less than that of area, e.g. when the area is reduced by 50%, the noise is only reduced by 29%, and for a noise reduction of 50% the area must be reduced by 70%. It might be possible by using a series of small contractions in a pipe to stop the "impact" sound wave without appreciably hindering the gas flow, but there is no evidence as to whether/

whether the 'hiss' would be stopped at the same time. The nozzles used in this experiment appeared to have more effect on the 'bang' than on the 'hiss'.

A somewhat similar experiment was also made on the car engine, page , but without the noise recording device. In this case the nozzle was a hole in a flange screwed on the pipe near the exhaust valve, and there was no reduction of noise.

When a single baffle without reduction of area was fitted at the end of a long pipe, page 7, there was a distinct reduction of noise.

It would therefore appear necessary to have any reflecting device situated at some distance along the pipe, otherwise the reflected wave has no liberty to move back. Some types of sound reflectors will now be considered.

FISH-TAIL AND CONICAL PASSAGES.

The practice of flattening and expanding the outlet of an exhaust pipe into the shape of a fish-tail has become general, but no theory has been advanced to account for the silencing properties of such a shape. In order to investigate the action of sound waves in a fish-tail the writer made the following experiments:-

A fish-tail of rectangular section was made of sheet steel and attached to a pipe from the exhaust of the cylinder used to produce the 'hiss', Drawing No.12. The exhaust valve was set to give a continuous hiss with the air supply at medium pressure. The intensity of the noise inside the fish-tail was examined by inserting a small metal tube attached to a rubber pipe the end of which terminated in a pad that could be held to the ear.

The results were not very conclusive, and a large fish-tail was made of wood in such a way that it could be/

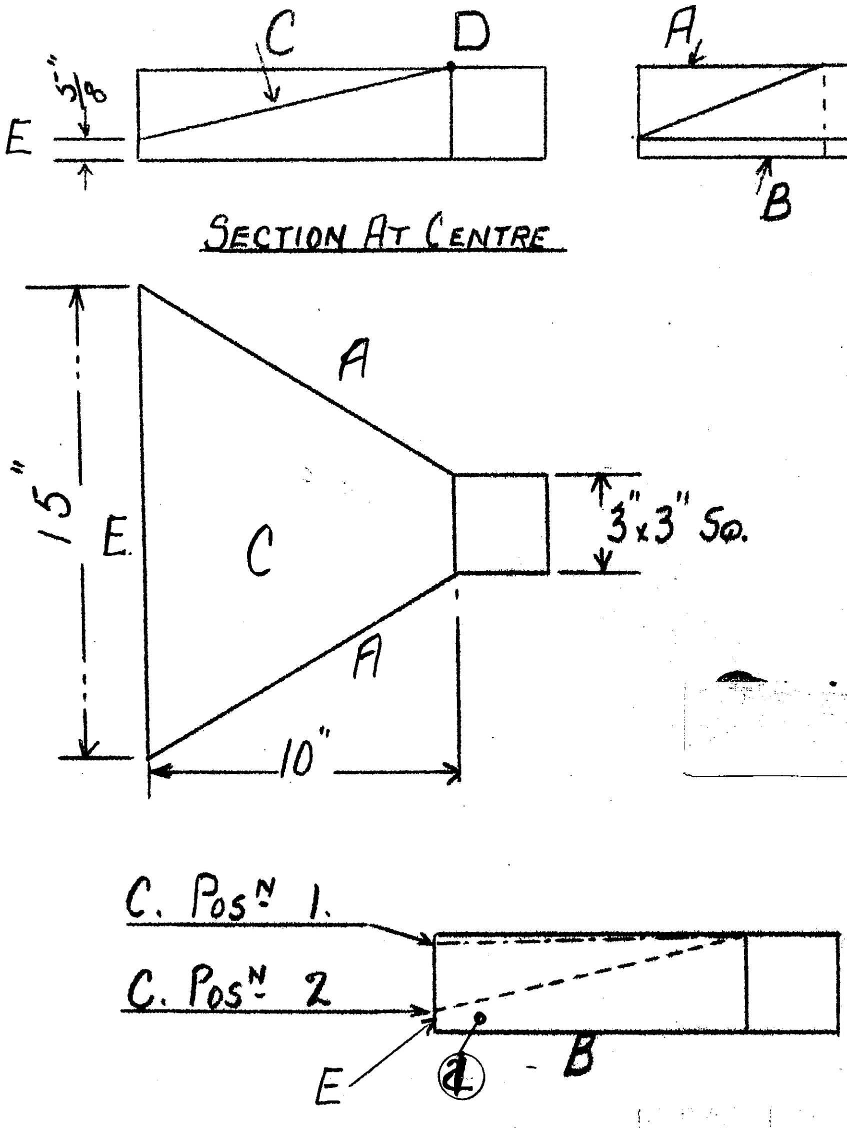

C
D
E
5/8"
A
A
B
SECTION AT CENTRE
A
15"
E.
C
3"x3" SQ.
A
10"
C. POSN 1.
C. POSN 2
E
B

FIG. 48

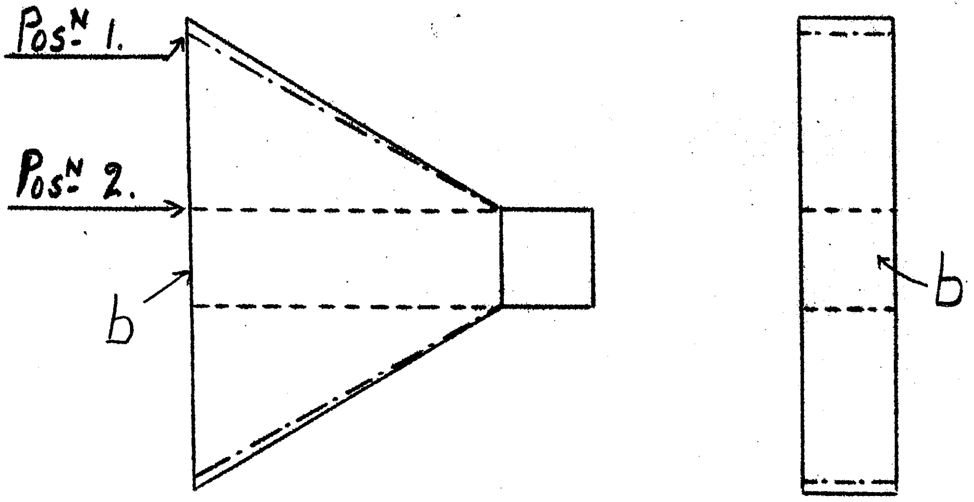

POSN 1.
POSN 2.
b
b

FIG 49.

be varied in form, Fig.48. In the first test the narrow sides A,A, and the side B, were fixed in position while the side C was hinged at D. With the side C parallel to B the intensities of the noise at (a) and of the noise that got outside were observed. The side C was dropped till the opening at E was about 5/8" giving the same cross-sectional area as the inlet at D. The noise at (a) increased as C descended, but the noise outside was much less. The inference is that the sound waves were being condensed and partly reflected back into the pipe.

When the sides B and C were kept parallel and the sides A A moved from position (1) to position (2), Fig.49, there was an increase in intensity at (b), but the noise outside was not changed. This was to be expected as there were no converging surfaces and no reflection.

Another example of the converging passage is the ear trumpet which collects and condenses the sound waves. If, as is generally supposed, the condensation of sound in passing from the large end of a cone to the small end is in the ratio of the areas of the ends, it is to be expected that the total sound energy entering at the large end would emerge at the small end.

The following experiment showed this belief to be incorrect:

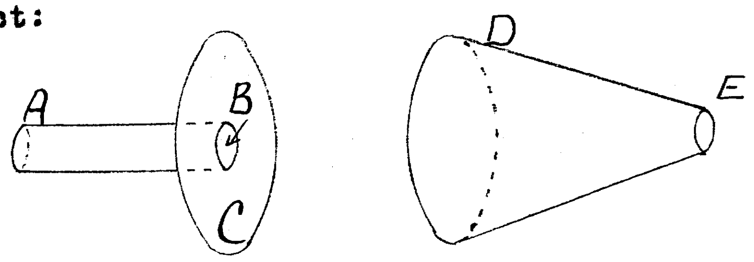


FIG. 50.

AB is the pipe from the end, B, of which the sound issues. C is a flange at right angles to the pipe. DE is/

is a cone which can be pressed on the flange C. In the experiment the cone was alternately pressed on the flange and removed, and it was very evident that there was less noise when the cone was on the flange. This proved that all the energy entering the cone did not emerge at the small end. The intensity of the sound inside the cone was observed by inserting a search tube as in the case of the fish-tail. The intensity was much greater at the small end E than at D, but it was impossible even to guess at the ratio of the intensities. The conclusion, therefore, is that a conical horn condenses the sound waves to some extent, and retains, presumably by reflection, a proportion of the sound energy.

The motion of sound waves in a conical horn has been studied theoretically and experimentally by several investigators. Their work as reported in the Physical Review comprises:-

(1) a Photographic Study of Sound Pulses between curved walls and Sound Amplification by Horns; by A.L. Foley. Vol.XX, 1922.

(2) The Performance of Conical Horns; by G.W.Stewart, Vol.XVI, 1920.

(3) Theory of the Optimum Angle in a Receiving Conical Horn; by V.A.Hoersch, Vol.XXV, 1925.

(4) Notes on Hoersch's Theory of the Optimum Angle of a Receiving Conical Horn; by G.W.Stewart, Vol.XXV, 1925.

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(1) The particular manner in which photographs of sound waves were obtained need not be detailed, but it should be understood that passages of rectangular and not circular form were used, e.g. for a horn, Fig. 51

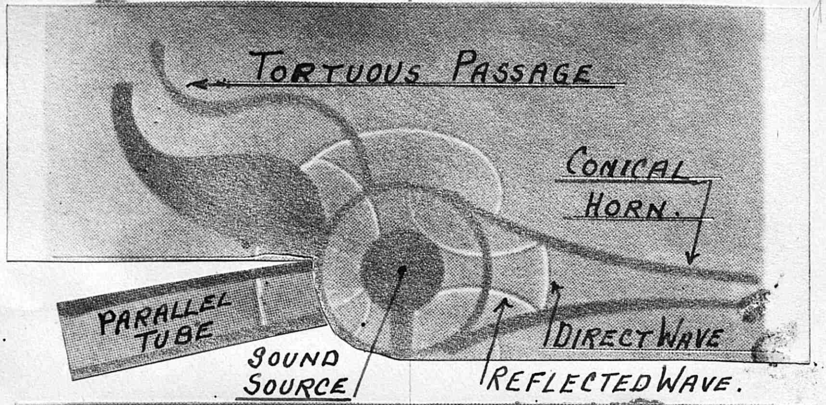


FIG. 1.

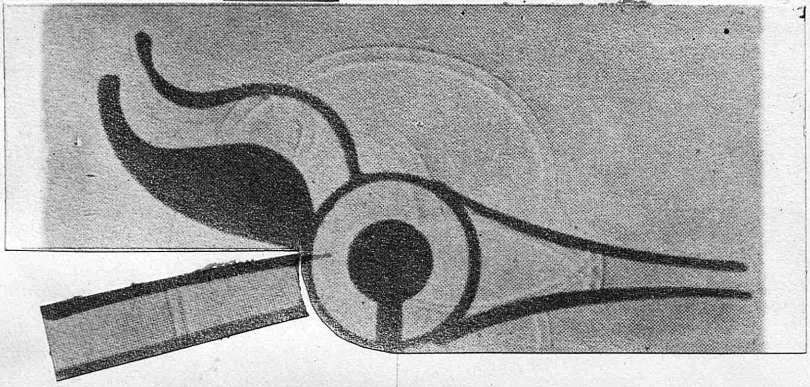


FIG. 2.

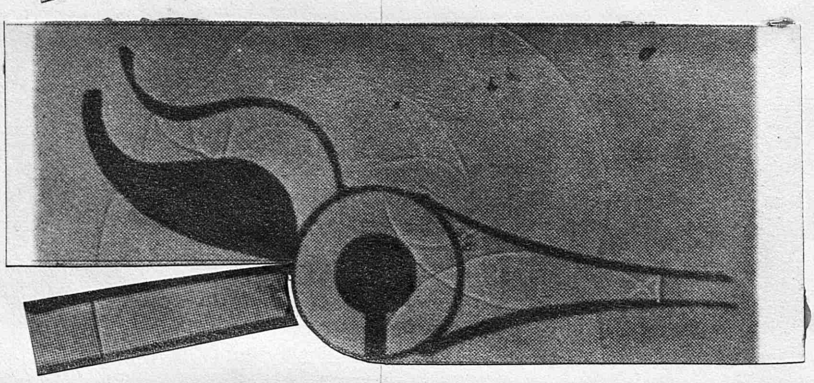


FIG. 3.

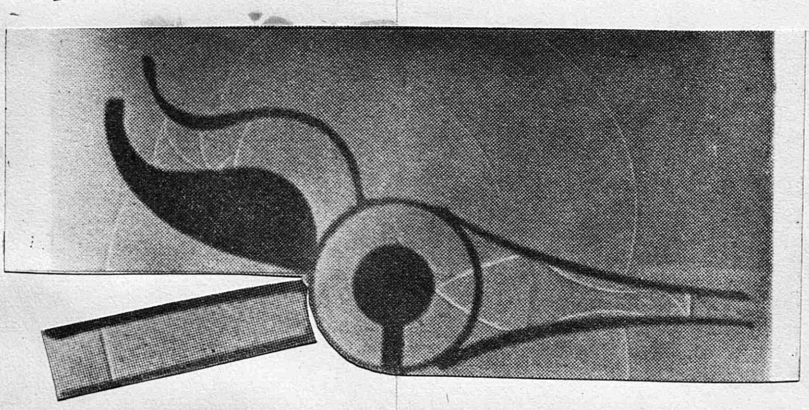


FIG. 4.

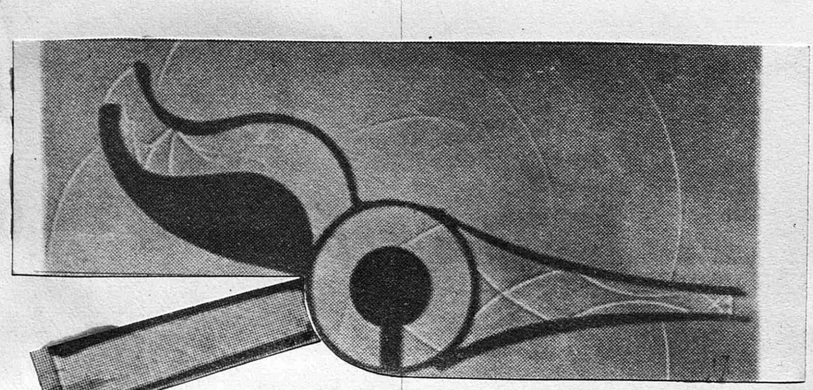


FIG. 5.

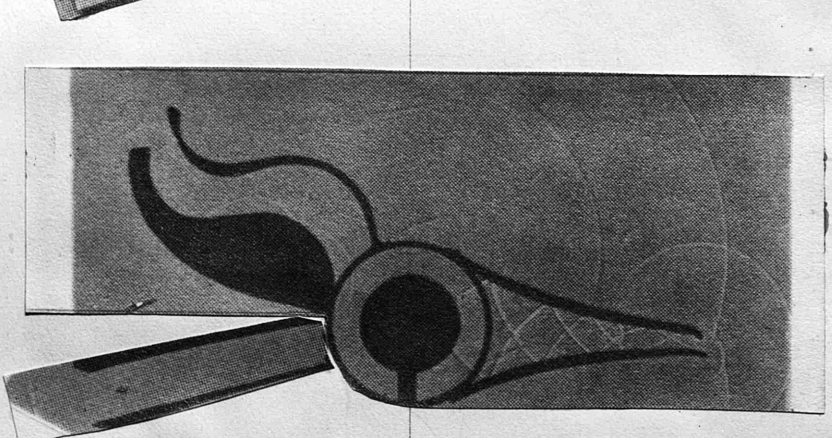


FIG. 6.

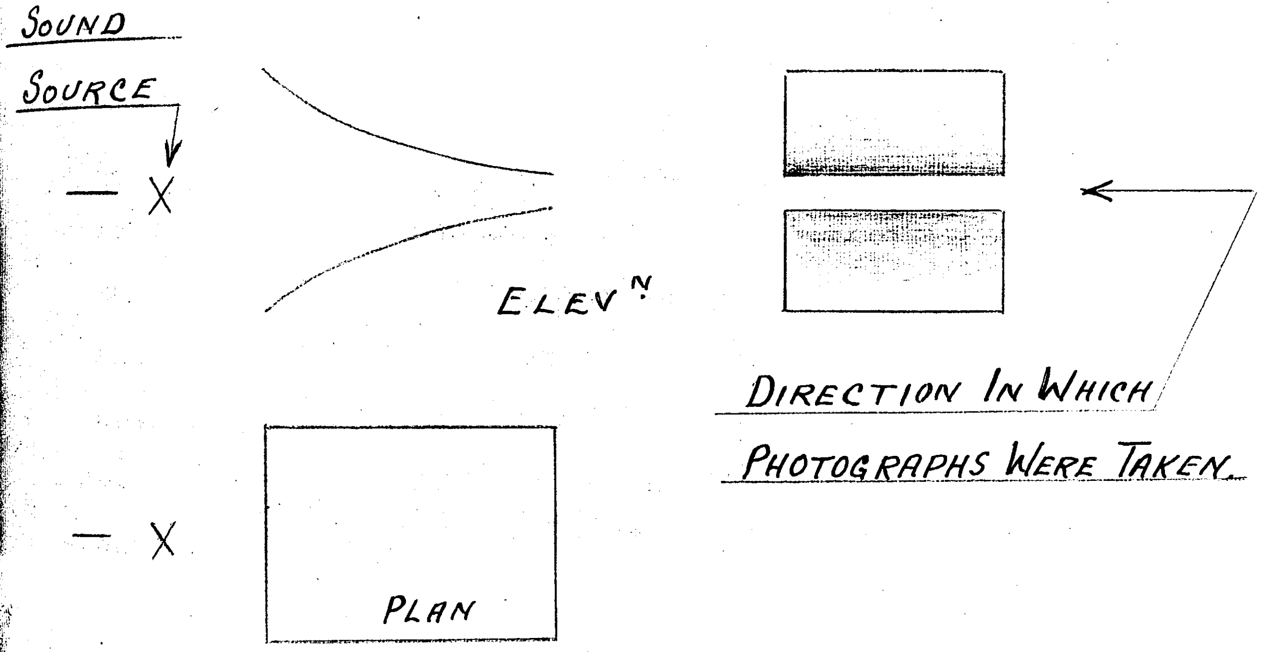


FIG. 51.

The sound wave was produced by an electric spark.

The progress of a wave along a parallel, a tortuous, and a converging passage, corresponding to a straight tube, a crooked tube, and a horn, respectively, is shown in the figures I to VI. The time interval between each figure is 0·00003 sec. and the double line in Fig. represents the wave positions at an interval of 0·000006sec.

The following is a quotation from the report. "All the pictures show that there was energy reflection in every case except when the wave front was at right angles to the surface and the motion of the air parallel to the surface of the tube. In the case of the horn there was continuous reflection from one end to the other even at the small end where the angle of the cone is very small. In the case of the crooked tube there were successive reflections. For this tube figures V and VI show respectively an emerging and emerged wave much more attenuated than in the case of the straight tube of the same size. A considerable portion of the wave energy appears to be trapped inside the tube. However, it will be observed that the reflected waves in general were headed towards/

towards the outer end of the tube. This is not true, however, of the horn. Here the advancing wave shows unmistakable evidence of intensity increase, and that it emerged from the small end of the horn considerably amplified. But most of the energy was lost so far as the small end of the horn is concerned. The lost energy was contained in the reflected waves which, as the photographs show, headed the wrong way 'backing out' of the horn."

Some other experiments were made to get an idea of how the amplifying power of a horn varied in relation to the ratio of the end areas of the horn. The results which are stated to be only approximately correct, were as follows:-

Ratio of End Areas of Horn	Amplifying Factor.
7.8	3.1
33.	7.9
122.	11.9
256.	13.

This shows clearly that the amplifying power of a horn is very much less than that given by the ratio of the end areas.

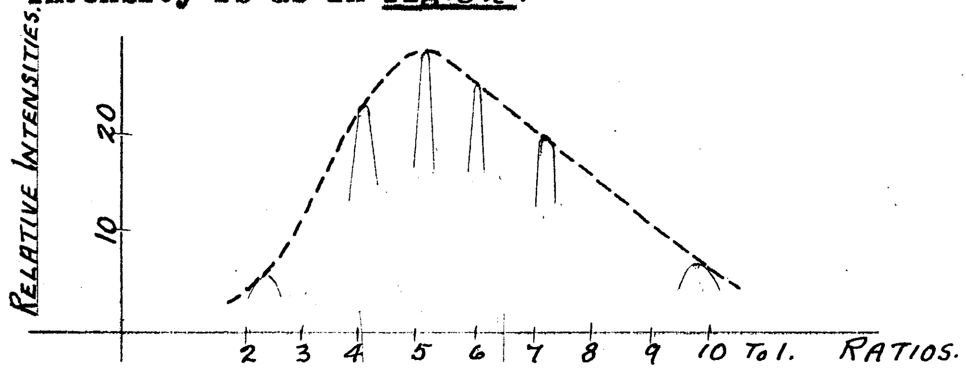
(2) In the Report by Mr. Stewart, experimental results are given of tests of amplifying power by horns, some of the conclusions being as follows:-

(a) When the angle of the horn is constant and the length of the horn is varied the maximum amplification is attained when the length is such that the horn forms a resonator for the particular note employed. The amplification at any other length is very much less than this maximum.

(b) When the angle of the horn is varied and that length is retained at which the resonance gives the maximum/

maximum amplitude, the intensity increases until the angle reaches a certain value and then decreases showing that there is an optimum angle for any note.

For a note having a frequency of 256 the curve of intensity is as in Fig. 52.



The optimum angle in this case is just under 12% ie a 5 To 1 ratio. For a frequency of 512 the ratio is 4 to 1 showing that the ratio decreases as the frequency increases.

(3) In this paper Mr. Hoersch gives a theoretical treatment showing that there is an optimum angle of horn for any note and its overtones.

(4) Mr. Stewart here comments briefly on the theory given in paper (3), and uses the expression there given for the optimum angle to check the values which he had determined experimentally. The theoretical and experimental values agree fairly well.

The horns used in all these cases were open at the large end so that the conclusions drawn are not strictly applicable to the conical end of a silencer where the large end of the cone is really the termination of a tube, e.g. Fig. 53

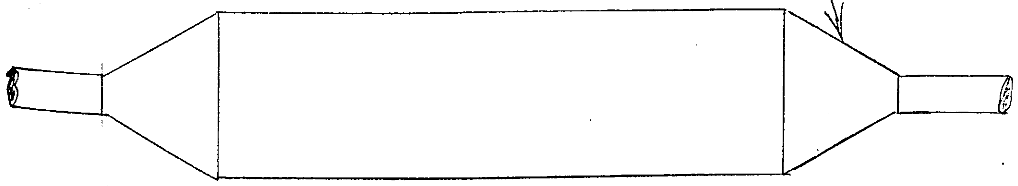


FIG. 53.

In/

In the following case the above considerations of optimum angle would apply: Fig. 54

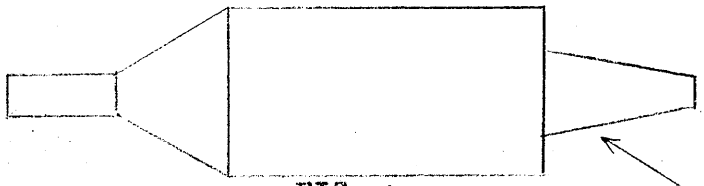


FIG. 54.

Here the conical outlet pipe and silencer end are similar to a horn with a flange, and it would be possible to get resonance of the high frequency waves in the conical pipe. Then if the angle of the cone were the optimum for resonant frequency the maximum amplification would occur. This would be very detrimental to the action of the silencer.

It is very unlikely that such a combination of circumstances would occur accidentally, but it is well to be aware of the possibility of such an occurrence in order that it may be avoided.

The most important result of these investigations is the evidence that the increase of intensity of a sound wave passing between two converging surfaces is not proportional to the decrease in area of the cross section of the wave front but is very much less than this decrease especially in the case of a wide angle cone.

This explains why a fish-tail has some effect in reducing exhaust noise. Although the fish-tail spreads out in one direction to give a cross section of constant area, it tapers in the other direction; and it is presumably this taper which reflects some of the energy and lessens the amount of sound that gets out. It was clear from the writer's experiments that a large fish-tail was more effective than a small one, even when the large one had a greater area at exit than the smaller one.

These/

These conclusions are drawn with reference to high frequency waves only. The low frequency waves of high pressure were not greatly suppressed by the small Brooklands fish-tail, see page 88. It would seem that there must be some relation between the size of a fish-tail and the wave length of the sound which it can reflect.

It is improbable that the design of the fish-tail can be improved to the extent of making a single fish-tail give sufficient silence, but several in series might have the desired effect. The attractive feature about the fish-tail is that it does not absorb much of the kinetic energy of the high speed exhaust gases.

The capacity of a cone to reflect sound explains the action of a silencer fitted to a motor cycle possessed by the writer in 1922. The shape was simply a cone without any baffles, Fig.55

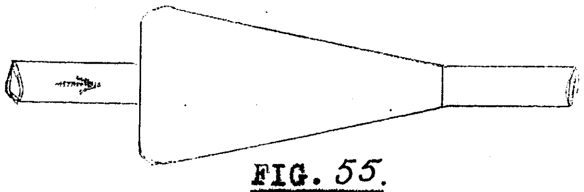


FIG. 55.

The alleged reason for the design was that the gases were allowed to expand at the inlet and cool down, becoming decreased in volume as they passed along the cone and reaching the outlet at reduced pressure.

The writer did not believe this theory but was unable at the time to offer any reason for the appreciable degree of silencing that took place.

THE EFFECT OF TORTUOUS PASSAGES.

The photographs of a sound wave moving in a curved path, page 95, show that the wave front is reflected by the/

the sides of the passage a number of times and does not emerge in the same form as it entered.

The wave does not return towards the source as it does in a horn and the only destruction of energy of the wave must be due to successive reflections from the sides of the passage. The fact that the wave does not all emerge at once would reduce the intensity as apparent to the ear.

REFLECTION OF SOUND BY BAFFLES.

The commonest form of baffle is a perforated plate across the axis of cylindrical silencers:-

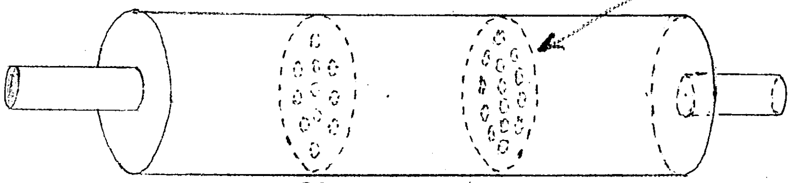


FIG. 56.

Previous experiments made by the writer showed that if the total area of the holes in the baffle was equal to the cross sectional area of the inlet pipe there was very little reflection of sound by the baffle especially when the diameter of the holes was greater than 1/8". If the total area of the holes was made considerably less than that of the inlet pipe a reduction in sound became evident. The reflection was then due more to the reduction of area, see page 91, than to pure reflection.

The question of the propagation of sound through an orifice in a flat plate is discussed in Lamb's Dynamical Theory of Sound, page 246 et seq.

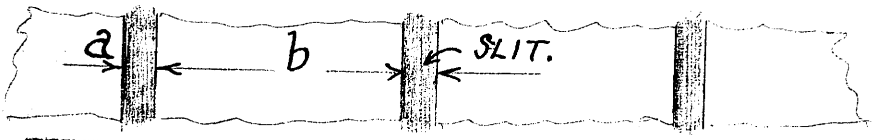
He concludes that for a circular orifice the proportion of sound that gets through is 0·816 of the amount that would have passed an area equal to that of the orifice if the plate had not been there. Now the coefficient of/

of discharge of a gas through a sharp-edged orifice is estimated to be about 0.55 when the difference in pressure between the two sides of the orifice is low, consequently such an orifice offers more resistance to the flow of gas than to the passage of sound.

The case of slits instead of holes is given on page 247 of Lamb's book:-

"For a long narrow slit the energy transmitted is comparable with and may even exceed that corresponding to an equal area of wave front in the primary wave. In the case of a grating composed of equal, parallel, and equidistant slits in a thin screen, the fraction of the total incident energy which is transmitted is found to be $\frac{1}{1 + k^2 l^2}$, where $k = 2\pi \div$ wave length, and $l = \frac{a + b}{\pi} \log . \sec . \frac{b}{2(a + b)}$, where 'a' denotes the breadth of the opening and 'b' that of each intervening portion of the screen."

Using the above expression and taking values of the wave length from $\frac{1}{2}$ ft. to 14 ft. the percentage of the incident energy which is reflected was obtained for the following values of 'a' and 'b'.



Case	1	2	3	4	5
a ft.	0.02	0.01	0.005	0.02	0.01
b ft.	0.2	0.2	0.2	0.4	1.0

The calculated values of the % of incident energy reflected by the gratings are given in the following table.

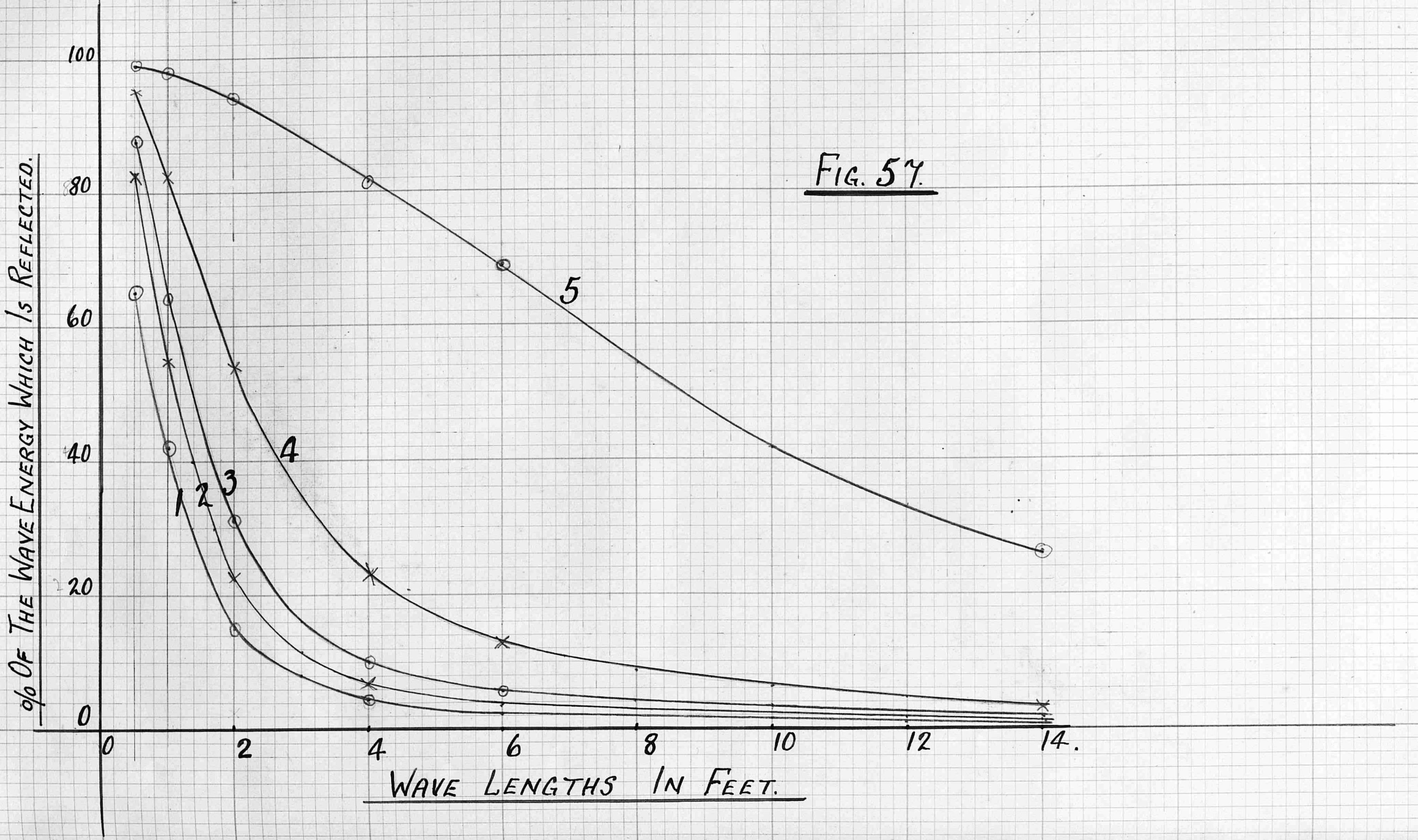

% Of The Wave Energy Which Is Reflected.
100
80
60
40
20
0
1
2
3
4
5
0
2
4
6
8
10
12
14.
Wave Lengths In Feet.

Fig. 57.

Case		1	2	3	4	5
Wave length ft.	0.5	65.7%	82.7%	87.8%	95. %	99. %
	1.	42.0	54.5	64.0	82.5	98.
	2.	15.4	23.0	31.0	54.3	94.
	4.	4.3	7.0	9.8	22.7	81.7
	6.	2.0	3.5	5.4	12.7	68.6
	14.	0.2	0.5	0.6	2.4	26.2

These values are plotted on the graph, Fig.57. It is evident that for all values of 'a' and 'b' the % of energy that is reflected is much larger for short than for long waves. The values for long waves are suspiciously low, and it is doubtful if the expression applies when the wave length is very great compared to the dimensions of 'a' and 'b'. When 'a' is zero the expression becomes $\frac{1}{1+\infty} = 0$, which is true because when 'a' is zero no energy can pass the screen. But when 'a' is very small the expression still gives a large value, which shows that there must be a limiting value for the relative sizes of 'a', 'b' and the wave length. No such limits are mentioned in the treatise.

Although the above expression refers only to parallel slits the results indicate that for any kind of aperture short waves are easier to reflect than long ones. This may explain the unusual degree of silence obtained in the case of a twin two-stroke motor cycle engine belonging to the writer, which has a pipe 6 ins. long leading to an open box silencer from which a tail pipe leads to another silencer. The short inlet pipe produces waves of a comparatively high frequency and these are very much softened by the second silencer.

Referring to the actual sizes of 'a' and 'b' it will be seen that in case 2, these are 0.01 and 0.2 ft. (or 0.12 and 2.4 ins) respectively. 'b' cannot be increased without making the whole plate inconveniently large nor can 'a' be made smaller without danger of sooting up. A number of baffles in series is a possible solution but this means the loss of kinetic energy of the gases since the velocity head is almost all lost after the passage through each baffle.

REFLECTION BY AN EMPTY BOX SILENCER.

There is a general belief that silencing depends on volume, e.g. in a review* of the improvements in motor cycles at the 1929 Motor Show it was stated that "Though there have been improvements of a minor nature in the silencers themselves, there appears to be no golden rule for silencer construction other than to employ the greatest possible volume for expansion purposes". This is just a repetition of the belief mentioned already on pages 84-5, that the exhaust gases are at high pressure and must be allowed to expand before reaching the atmosphere. Despite the fact that the belief about the pressure is incorrect it is nevertheless quite true that volume alone does, within limits, reduce the noise. The test of the Brooklands silencer without the fish-tail, curve No.95 page 88, shows that the intensity of the primary positive wave was reduced by the silencer to 70% of its value from an open pipe. This cannot be accounted for by reflection from the silencer end which is small and not shaped to give much reflection. The reason will be evident from a consideration of the ordinary theory of spherical or box resonators.

*See The Motor Cycle, Dec. 5th, 1929.

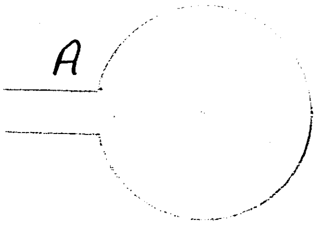


FIG. 58

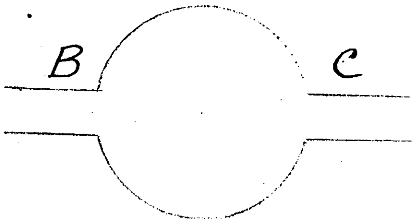


FIG. 59

The assumption is that the air in the neck of the aperture 'A' Fig. 58 acts as a piston compressing the air in the vessel and then being pushed out by it. Taking the case of a resonator with two necks 'B' and 'C', Fig. 59., suppose that the air piston in B receives an impulse and compresses the air in the vessel. This static pressure now pushes out the air pistons in B and C, each absorbing half the energy. Applying this to the engine conditions B is attached to the exhaust pipe, the vessel or resonator becomes the silencer, and C is the tail pipe. The initial wave of pressure enters at B and raises the pressure in the silencer. This pressure sends one wave out at the tail pipe and another back into the exhaust pipe. In the ideal case the reduction in intensity of the issuing wave could be 50%, but not more. Also the ordinary resonator theory is not strictly applicable to exhaust wave conditions, and the ideal case is not likely to be reached. Silencing by volume alone could therefore be achieved only by using a series of large chambers, which is not convenient.

DISSIPATION OF SOUND ENERGY.

One of the features of many silencers is curiously shaped baffles or passages which are stated to be for the purpose of "breaking up" the exhaust gases or explosions or sound waves. The scheme is probably intended to be similar to that of breaking up the force of the impact of water by causing it to flow over rough and irregular channels. A sound wave in a gas is, however, a rather different phenomenon and cannot be treated in quite the same way.

In Lamb's Dynamical Theory of Sound, p.196, he states, "It is to be observed that it is only through the action of dissipative forces, such as viscosity and thermal conduction, that sound can die out in an enclosed space, no mere modification of the waves by irregularities being of any avail". Therefore if the sound energy of the gases passing along the exhaust pipe is to be absorbed instead of reflected (as considered in the previous section) this can be done only by viscosity and thermal conduction.

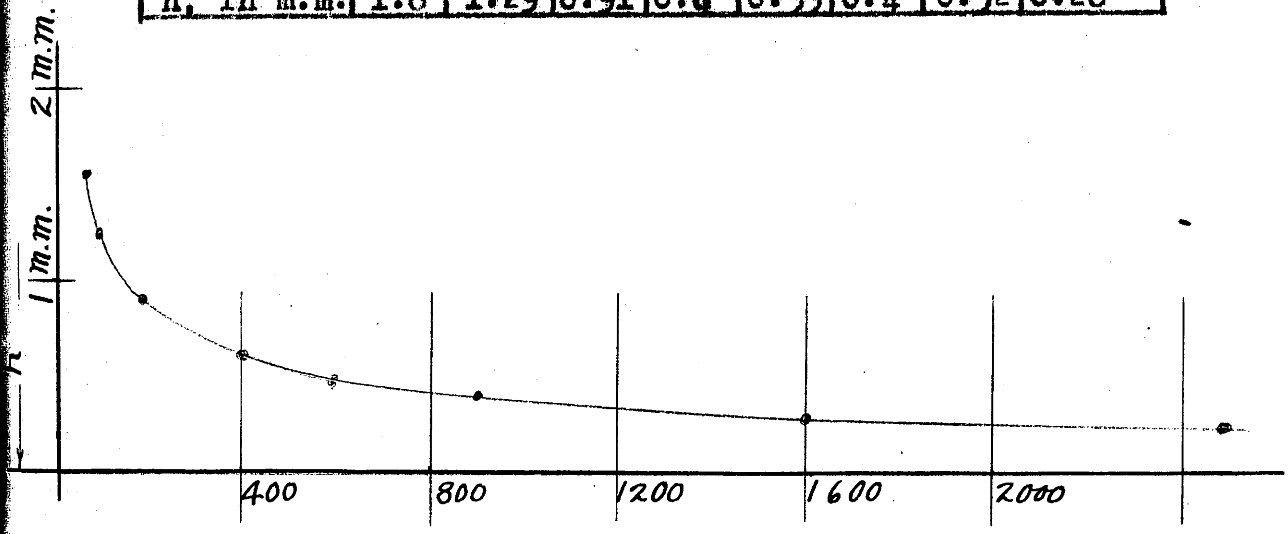
DISSIPATION BY VISCOSITY.

When a sound wave moves along a surface such as the inside of a tube, the layer of air which is in contact with the surface remains at rest. Due to the viscosity of the air there is a velocity gradient from zero at the surface to the point at which the surface drag ceases to have an effect. The depth to which the drag penetrates is stated by Lamb (p.192) to be given by 'h' $= 1.29 \div N^{\frac{1}{2}}$, where 'h' is in cms. and N is the frequency of the wave. He also states (p.194) "When 'h' is large compared with the width of the tube the mere inertia of the fluid ceases to have any appreciable influence, the mean velocity over a cross section being determined by an approximate statical equilibrium between the pressure gradient (in the direction of the walls) and the friction of the walls".

Consequently if a sound wave enters a tube of very small diameter its energy is soon lost in friction.

It is of interest to note the values of h for certain frequencies.

N	64	100	200	400	600	900	1600	2500
h, in m.m.	1.6	1.29	0.91	0.6	0.53	0.4	0.32	0.26



FREQUENCIES.

A tube or orifice having a radius equal to the value of h will have a considerable damping effect on a wave of the corresponding frequency, but for radii greater than h the effect will rapidly diminish. The limiting effective diameters will therefore be 1/8" for N = 64; 1/16" for N = 240; and 1/32" for N = 900.

From previous experiments with perforated baffle plates and tubes the writer found that holes less than 1/8" diameter gave better silencing than an equivalent area of holes of ¼" diameter. This is common knowledge and the makers of many silencers use baffles and pipes with holes of small diameter.

To obtain complete silencing by this method alone it would be necessary to use a number of small tubes of sufficient length to absorb the whole of the sound energy. A much larger cross sectional area than that of the exhaust pipe would be required, otherwise the tubes would cause back pressure.

The scheme is not really worth going into because small tubes would rapidly soot up. All silencers using small holes are objectionable because cleaning is difficult and requires to be done frequently.

DISSIPATION BY THERMAL CONDUCTION.

The designs of many silencers include schemes for cooling the hot gases on the supposition that reducing the volume (and therefore the pressure) lessens the noise.

The condition for cool exhaust obtains for a few seconds after the engine is started, and the noise is not then appreciably less than when the pipe and silencer have become heated. As the writer has shown, the noise is produced at the exhaust valve, and cooling the gases after that does not remove much of the sound energy.

The compressing of the air due to the passing of a sound wave raises the temperature of the air but the rise and fall of temperature is so rapid that in general there is no time for loss of heat. The propagation of sound in free air is thus adiabatic. If instead of moving in free air the wave were to pass between plates of material having a high thermal conductivity it is conceivable that there might be an exchange of heat between the air and the plates resulting in a net loss of energy of the wave. Successive repetitions would ultimately absorb all the energy of the wave, but as a practical scheme it is obviously impossible.

WAVE DAMPING SHOWN BY THE PRESSURE CURVES.

The curve of pressure variation in an open pipe No.20, sheet 7, shows a gradual diminution in the amplitude of the waves until the energy almost disappears by the end of the cycle. This diminution is due to two causes - viscosity and discharge of energy at the open end of the pipe. There is no restriction at the opening and a pressure wave emerges freely, but the whole of the energy discharged into the atmosphere is not recovered and this accounts for most of the damping, viscosity not having much effect in a large smooth pipe. Rayleigh gives the expression for the time taken for the energy to die out in a resonator by discharge from the open end into the atmosphere as $\frac{4\pi a^3}{n^4 S}$ where 'a' is the speed of sound, 'n' is 2π x frequency, S is the volume of the resonator in ft.^{3}. For curve No.20 the expression gives a time of .082 sec. the pipe length of 5' 2" being taken as the resonator. Considering the curve after the exhaust valve is shut it is evident that the vibration would not die out completely in one and a half revolutions, the time of which is .09 sec. Taking the actual damping from the curves it is found that the amplitude of each wave is about 0.7 of that of the preceeding one. The number of repetitions of the wave required to reduce the amplitude to 1/50th of its initial value is given by $(.7)^{n'} = .02$, from which $n' = 11$, and the corresponding time is 0.18 sec. The theoretical time is therefore an underestimate. This is to be expected as the theory is based on the assumption of a wave of small intensity, whereas in this case the wave initially contains a relatively large store of energy which would take longer to dissipate.

EFFECT OF RESTRICTED OUTLET ON THE DISCHARGE.

In the case of the parallel pipe open at the end the whole of the energy in the pressure wave discharges into the atmosphere leaving a negative wave in the pipe. The pressure wave then returns into the pipe but some of the energy has escaped into the atmosphere and there is a net loss of energy in the pipe.

If the pipe is reduced in diameter at the outlet less energy will escape and still less will be returned. The curve for the fish-tail No.38 Sheet 10 shows that only a small negative wave was produced and that the return of energy into the pipe was negligible so that the net loss was very rapid.

When the resistance of a silencer to the passage of the wave is great and when the silencer is of small volume a large proportion of the wave is reflected as in curves Nos.33-4-5, and there is no return from the atmosphere into the silencer. The energy is entirely lost in a few reflections.

When the silencer has a rather larger volume and also a big resistance part of the wave is reflected and part escapes, and the net discharge is not quite so rapid, see Curve No.36, Sheet 10.

Curve No.37 shows even slower damping than in the case of the open pipe. The Brooklands silencer has a very large volume the gas in which absorbs most of the energy of the wave coming out of the pipe, and returns it to the pipe, a small proportion - about 20% - being discharged from each wave at the fish-tail.

The study of the rate of discharge of energy from, or dissipation in, any exhaust system is important in the case of high speed engines where the time of a cycle is very small. The energy contained in the system at the instant when the exhaust valve is opened has a considerable influence on the ensuing wave motion.

INTERFERENCE.

The principle of Interference is well known:-

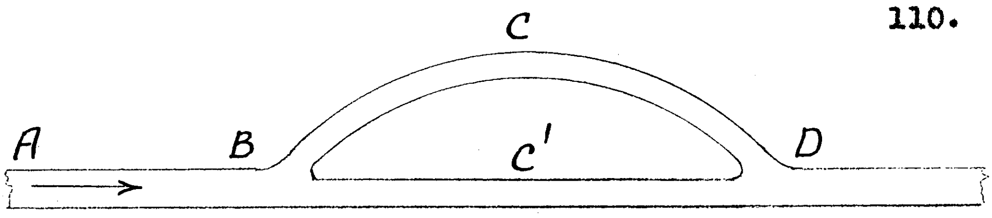


FIG. 60

A pipe AB along which a note is passing is branched at B and reunited at D. If the relative lengths of the branches BCD and BC'D and the wave length of the note are such that a positive node arrives by one branch at D at the same time as a negative node arrives by the other branch the two nodes cancel each other and the sound ceases. This also happens for notes having frequencies which are multiples of this particular frequency. In the case of a noise in which there are many notes having widely differing frequencies interference applied in the above manner can have only a very small effect. The writer tried another scheme which could not completely silence any one note but which would have a partial effect on a number of notes:-

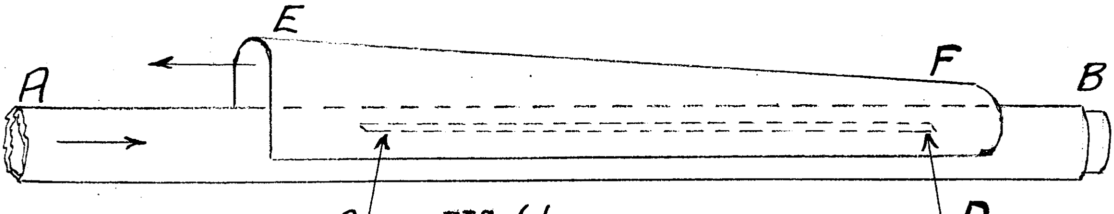


FIG. 61.

The pipe AB was plugged at B and a long slot CD was cut parallel to the centre line. A cover EF was fastened to the pipe the end E being open and F closed. The idea was that a wave passing from C to D as it issued from the slot would interfere with part which had already issued and was returning from D to C inside the cover. When tested on the "hissing" device the cover was a loose fit so that it could be put on and taken off quickly. There was a distinct decrease of sound when the cover was on in the above position, but when it was reversed - with the outlet at D - there was no decrease. This appeared to indicate the action of Interference as expected. The device was tried on the engine and did reduce the noise considerably.

The possibilities of the use of Interference would be worth while exploring as it is a scheme which does not involve reflecting the waves or restricting the outlet.

The writer is not aware of this principle having been intentionally made use of in the design of any standard silencer.

EXAMINATION OF THE SILENCERS WHICH WERE TESTED.

The various ways in which sound waves can be destroyed or prevented from getting out of the exhaust pipe have been discussed and the reasons why silencers which are designed on a relatively inaccurate conception of the conditions to be met, are partly successful, will now be evident.

Fish-tails, baffles, tortuous passages, large volume etc. etc. are meant to give gradual expansion of the gases and to prevent the formation of sound by so doing. Actually these devices do not prevent the formation of sound waves but they do prevent some of the existing waves issuing from the pipe. The order of merit deduced from the apparent sound suppressing qualities of the silencers when tested on the engine was as follows.

1. Howarth, and Vortex Standard.
2. Carbjector.
3. Vortex Tube.
4. Brooklands with fish-tail.
5. " without fish-tail.

When tested on the "hiss" and "bang" producing devices the order of merit was much the same.

The engine test for "bang", see pages 88 gave a different order:-

1. Carbjector.
2. Vortex Tube and Brooklands.
3. Howarth.

The Vortex Standard could not be given this test as it had been returned to the makers.

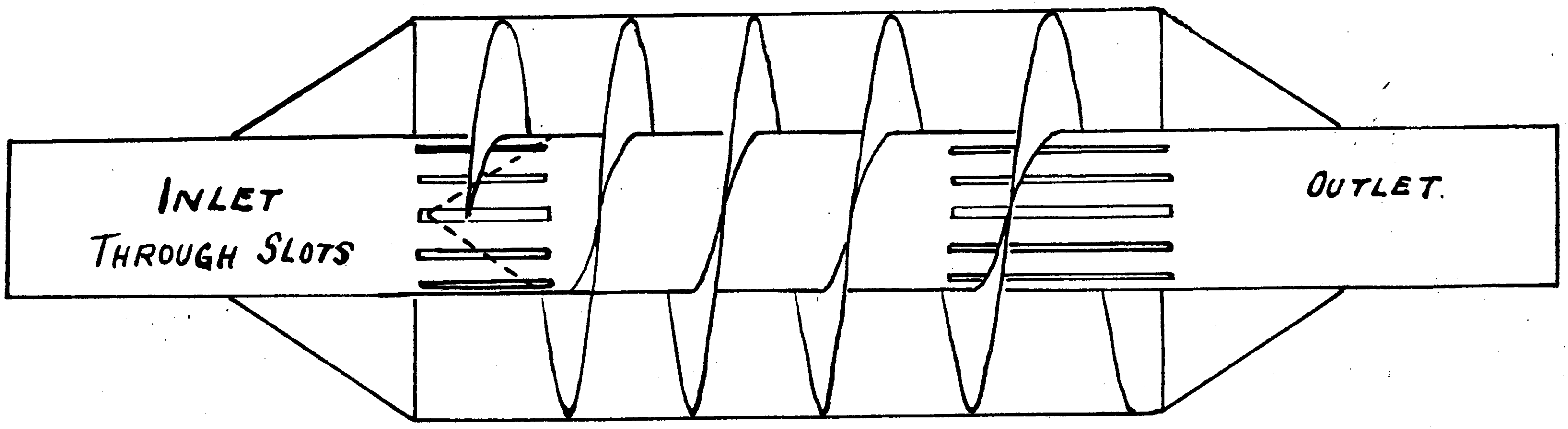
The difference between the apparent noise of the first list and the "bang" noise of the second list is accounted for by the differences in the capacities of the silencers to suppress "hiss", see pages 90 .

THE HOWARTH SILENCER. DRAWING NO.10.

The gases enter the first slightly conical chamber through long slots which do not restrict the flow, but owing to the flow being reversed there may be some reflection and interference. The small holes in the central baffle and the ¼" hole at the "ejector" end have a cross sectional area of 0.602 in^2. From Fig.46 , page 91 this restriction would account for the reduction of "bang" shown in curve No.91, before page 88 . The conical shape of the three chambers may be the factor which gives this silencer a better capacity to suppress "hiss" than the others. Also, all the chambers are "dead", i.e. they could not form resonators for notes of any pitch.

THE CARBJECTOR SILENCER. DRAWING NO.10.

The cross sectional area of the spiral passage is about 1 in^2 which would give little more than half the reduction of "bang", Curve No.82. The resistance of the long spiral passage may account for the rest of the reduction. The effect of such a passage on a wave front has been stated on page 99, and this action reduces the hiss, but not to a sufficient degree. A spiral passage alone does not give much sound reduction. This was shown



THE CONNOR SILENCER

FIG 62

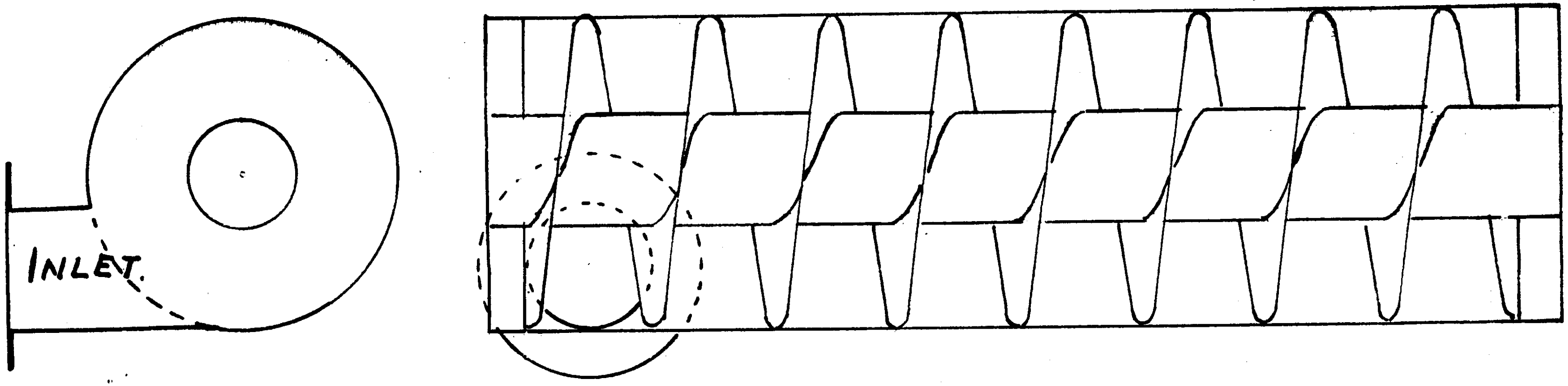


FIG 63

SPIRAL SILENCER WITH INLET & OUTLET TANGENTIAL.

in the R.A.F. Report (page 67). The "Connor" Silencer which was similar to the Carbjector, Fig.62 was tested against another Fig.63 in which the inlet was tangential to the curve of the spiral so that there was no abrupt change of direction of flow and no reflection at the inlet. The spiral passage was longer than in the "Connor" Silencer but the silencing was not so good showing that a spiral passage is not of itself a good sound reflector or destroyer.

THE VORTEX TUBE SILENCER. Drawing No.11.

The cross sectional area at the throat of each "cup" is 1.4 in^2 which would give half the reduction of "bang" shown by Curve No.93. There are 8 cups in series which might be expected to account for more than the remainder. The stream-lining of the throat of the constriction may lessen the reflection due to the reduction of area, and it is very evident from Curve No.94 that reflection is increased by reversing the silencer so that the flow is against a sharp edge.

These cups are not a good shape for reflecting high frequency waves and do not suppress the hiss very much.

THE VORTEX STANDARD. Drawing No.11.

The outlet area at the fish-tail is 0.9 in^2, but the internal areas could not be measured. The silencer had to be returned to the makers after a few days and was not tested so thoroughly as the others. Its silencing properties were due partly to restricted outlet, to reflection by other restrictions, and to volume.

THE BROOKLANDS STANDARD. Drawing No.11.

The reflection in this case is due partly to the large volume and to the fish-tail.

The silencing is not good, but this silencer is intended mainly for use on the race track where a considerable amount of noise is unavoidable.

BACK PRESSURE AND SILENCING.

In all the above cases silencing is due to successive reflections of the sound waves back into the exhaust pipe thus allowing the energy to emerge in several stages.

The degree of silencing is fixed by the proportion of the total wave energy ~~wave pressure~~ which is reflected. It is the reflected wave of high pressure which influences engine performance: the pressure of the high frequency waves has not yet been recorded and is unlikely to have any appreciable effect.

There are two factors which decide the effect of any silencer on power output - pressure reflected to the engine and the time of this reflection relative to piston position and exhaust valve closing. This can be seen by comparing the Howarth and the Brooklands, Curves Nos.33 and 37, Sheet 10. With the Howarth there are two reflections and the press is below atmospheric at T.D.C., whereas with the Brooklands the reflections take longer and the second one occurs at T.D.C.

Most investigators are agreed that silencing is in proportion to decrease of power output, and it is easy to see that this could obtain in many cases, e.g. suppose silencer "A" gives a curve such as shown in Fig.64.,

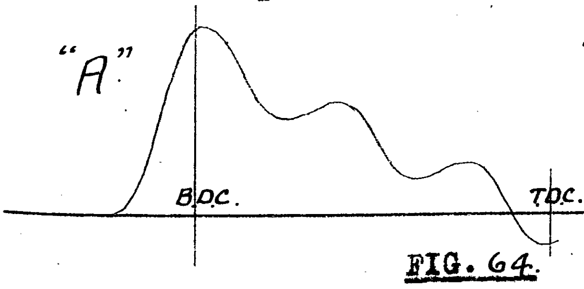


FIG. 64.

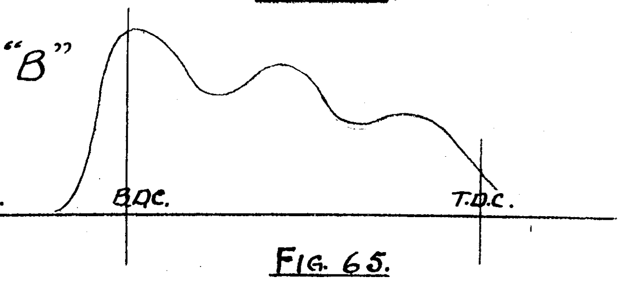


FIG. 65.

and that another "B" gives the curve Fig.65. "B" reflects more of the pressure which is higher at TDC.

But it would be quite possible to have two other silencers "C" and "D" having larger volumes giving quite different curves, Figs 66 and 67.

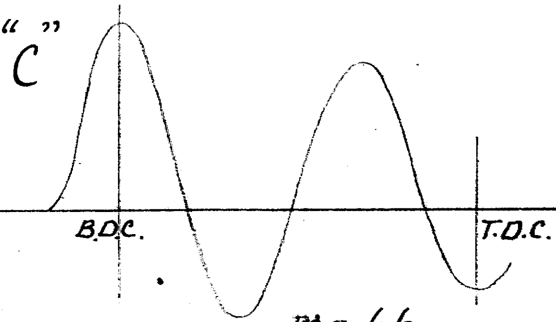


Fig. 66

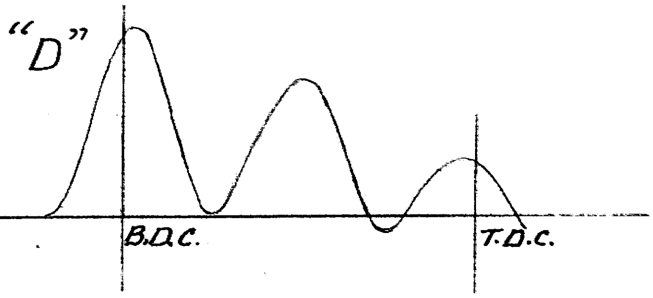


Fig. 67.

These might both have the same reflecting capacity as either "A" or "B", but "C" is better than A from the power point of view while D is worse.

It is therefore impossible to establish any general relation between reduction of sound and reduction of power.

C O N C L U S I O N S.

The ultimate object of this research was to arrive at a means of obtaining a completely silent exhaust without detriment to engine performance.

Although this goal has not yet been reached the primary object of a good working knowledge of almost all the conditions of pressure variation and sound production and suppression has been gained.

The following is a brief summary of the general conclusions arrived at by the writer:-

1. The conclusions drawn from the preliminary work on the car engine have already been given ...... pp.14-16.

2. THE MULTI-POINT INDICATOR:-
An indicator such as that devised by the writer is suitable for speeds and pressures such as obtain in the exhaust systems of high speed petrol engines ..... p.22.

3. CURVES OF PRESSURE IN THE EXHAUST PIPE:-
The gas pressures in any standard exhaust system are of sufficient magnitude to exert an appreciable influence on engine performance, and this influence is greatly affected by the periodic nature of the pressure variation. Many examples of pressure variation for different exhaust systems at various engine speeds are given ..... pp.43-61.

4. VELOCITY OF SOUND:-
The velocity of high pressure sound waves in gas contained in pipes appears to be less than that of normal/

normal sound waves ....pp. 68-74.

5. ANALYSIS OF EXHAUST NOISE:-

The noise is composed of a high pressure, low frequency, wave and an assortment of very high frequency waves the pressures of which have not been measured. ....pp. 76-85.

(See also Appendix).

6. NOISE INTENSITY:-

The intensities of the high pressure waves as measured by the indicator do not correspond exactly with the intensities as estimated by the ear showing that the high frequency waves account for a considerable proportion of the total noise.... pp. 85-90.

7. METHODS OF SUPPRESSION OF SOUND:-

In general the noise is suppressed by being reflected back into the exhaust pipe.

The efficiency of most of the devices used in silencer construction have been examined, e.g.

(a)	Restricted Outlet	pp. 90-92
(b)	Fish-tail and Conical Passages.	pp. 92-99
(c)	Tortuous passages	pp. 99-100
(d)	Baffles	pp.100-103
(e)	Empty box	pp.103-104

No single device is sufficient to reflect the required amount of sound without unduly restricting the fl w of gas, and a series of reflecting elements must be used.

8. DISSIPATION OF SOUND ENERGY:-

The dissipation of sound energy by viscosity or thermal conduction is impracticable in the case/

case of engine exhaust systems. .... pp. 104-107

9. WAVE DAMPING SHOWN BY THE PRESSURE CURVES.

The discharge of sound energy from an exhaust system may take an appreciable time and at high engine speeds this is an important consideration...... pp. 107-109.

10. INTERFERENCE:-

The complete elimination of exhaust noise by Interference is impossible owing to the noise being composed of many waves having different frequencies......pp.109-111

11. THE SILENCERS WHICH WERE TESTED:-

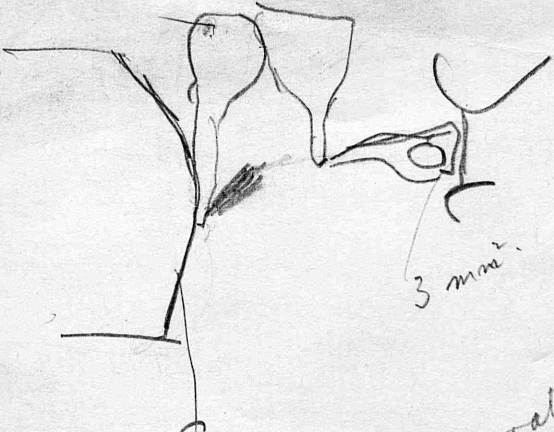
These illustrated to some extent the relative values of some of the devices which reflect sound and show that silencers which reflect high pressure, low frequency waves, do not necessarily reflect much of the high frequency waves. .... pp. 111-114.

12. BACK PRESSURE AND SILENCING:-

The connection between silencing and engine performance is very complicated and no general statement can be made about it. ......pp.114-115.

In conclusion the writer wishes to express his indebtedness to the University of Glasgow for providing the apparatus and the facilities for carrying out this research, and to Professor W.J. Goudie for his advice and encouragement.

———oOo———



Harvey Fletcher
Speech + Hearing
1929

The Ear

APPENDIX.

The Analysis of Noise.

In order to appreciate fully the difficulties attendant on a proper analysis of the sounds which are being dealt with it is advantageous to consider the physiological aspect of the question.

The following comments on noise as distinct from musical sounds are taken from a paper on "A Modification of the Helenholtz Theory of Hearing"* by Dr. A.A.Gray.

The ear is a complicated structure and only that part which estimates pitch and intensity of tone need be referred to. This part is shown diagrammatically in Fig.68. AB, CD, etc. are fibres embedded in a long, thin flat membrane MN, which is fixed to rigid supports and is in tension in the transverse direction only. Above the membrane is a pad of unyielding material. The membrane is tapered so that the fibres vary in length and each fibre responds to a sound vibration of one particular frequency. The membrane is covered with minute hairs which touch the pad, and when a fibre is bent upwards by a sound pulse the hairs are pressed on the pad with a greater or less degree of force. This pressure affects the auditory nerves located at the roots of the hair cells, thus conveying to the brain the presence of vibration in this particular fibre. The sense of hearing is therefore found to be a

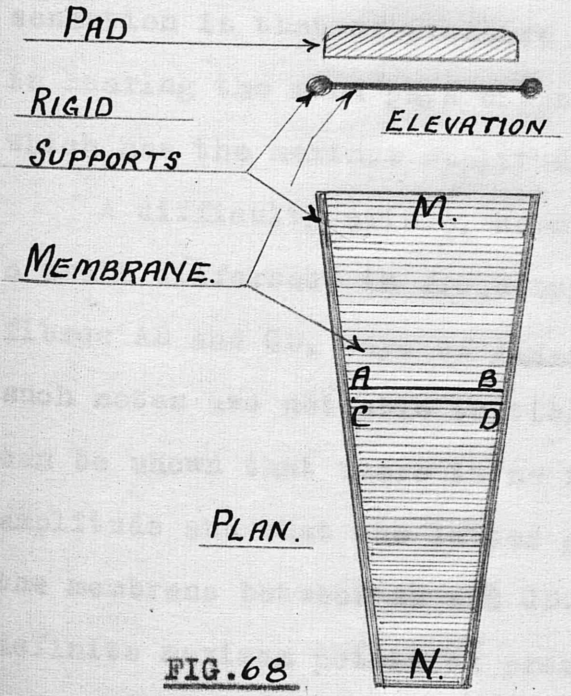


FIG.68

*British Association Meeting, Dover 1899.

highly specialised sense of touch.

Now the taper of the membrane gives a very gradual change in the length of the fibres so that when a pure tone sets a particular fibre in vibration the adjacent fibres are also set in sympathetic vibration, though to a less extent, and this is enhanced by the fact that the fibres are bound together by the membrane. It would seem that the brain should be conscious of more than one note or should at least have difficulty in estimating the pitch accurately. The case is analogus to that of a sensation of touch when a sharp point is pressed into the skin without penetrating it. The nerves all round the point of maximum pressure are excited and yet the sensation is that of pressure at a single point. Similarly in hearing the mind pays attention only to that fibre which has the maximum amplitude.

A difficulty arises, however, when two notes having a small difference in frequency are sounded together. The fibres AB and CD, say, of Fig.68 corresponding to two such notes are not able to vibrate independently, and it can be shown that there is no fixed point of maximum amplitude and that the latter point moves to and fro along the membrane between AB and CD. It is this lack of definite maximum points of pressure which causes the sensation of noise and prevents the noise from being analysed into its component parts. Two pure notes sounded together can thus produce a noise, but most noises heard in ordinary experience are composed of many notes of different frequencies either sounding together or in rapid sequence. The approximate pitch of a noise composed of two or more notes not widely differing in frequency can be estimated, and if there are separate groups of such notes there being a considerable difference in average frequency between the groups, it is possible in a limited degree to distinguish the separate groups.

On the other hand single notes which are not in musical harmony but which are fairly far apart on the scale can be readily distinguished. These may, however, produce a discord which is not necessarily the same thing as a noise.

The writer has observed that the difficulty of analysing mixed sounds is greatly added to when the sound is extremely loud or when some components of the sound are much louder than others. It is evident that if the membrane Fig.68 is pressed very hard against the pad its sensitiveness to differences of pressure will be very much decreased. Also in the case of a long high pressure wave caused by an explosion it is probable that the whole membrane is pressed upwards and consequently any smaller vibrations superimposed on this motion would be difficult to detect.

The general opinion regarding the kind of noise produced by a high speed engine exhaust is that it is entirely explosive. An explosive noise may be produced by a single pressure pulse such as the crack of a rifle, but it may also be composed of a large number of high frequency notes sounded together for a very short interval of time. The essential features are shortness of time and magnitude of intensity. The ease with which the ear may be deceived is illustrated by the following experience of the writer's. The air compressor in the laboratory had been running for some time and was then stopped and immediately after there was a loud explosion as if a valve on the air bottle had been blown off. At least so it seemed to the writer, but an investigation showed that the apparent "explosion" had really been caused by a very large coil of hard copper wire falling on the floor! The "ring" of two pieces of hard metal striking each other is a high pitched note, and a large number of pieces striking together as in the above case made a noise exactly similar to that of a sudden blow-off of high pressure air.

The writer had studied exhaust noise for some years before discovering that it was partly composed of a violent hiss. The plunger arrangement for silencing the small engine, see page 12, was tried on a larger engine without the same success, and the reason appeared to be that the noise from the large engine had a certain continuing quality which was absent from the "pop" of the small one. On analysing the sensation produced by the "continuing" noise it was felt to have a rasping quality similar to that produced by sharpening a hand saw.

This pointed to the presence of high frequency waves which could only be caused by a "hiss". It was impossible to tell what proportion of the total noise was due to this cause, and the experiments described on pages 78-82 were intended to give some idea of this proportion.

In view of the fact that the ear is capable of such fine accuracy in the estimation of pitch it seems strange that it has no definite means of estimating loudness with anything approaching accuracy. The loudness of a sound is gauged in much the same way as the weight of a body - by the intensity of a muscular sensation which is a very variable factor.

The study of noise without the aid of mechanical devices is thus seen to be extremely difficult.

Drawings

Accompanying Thesis On

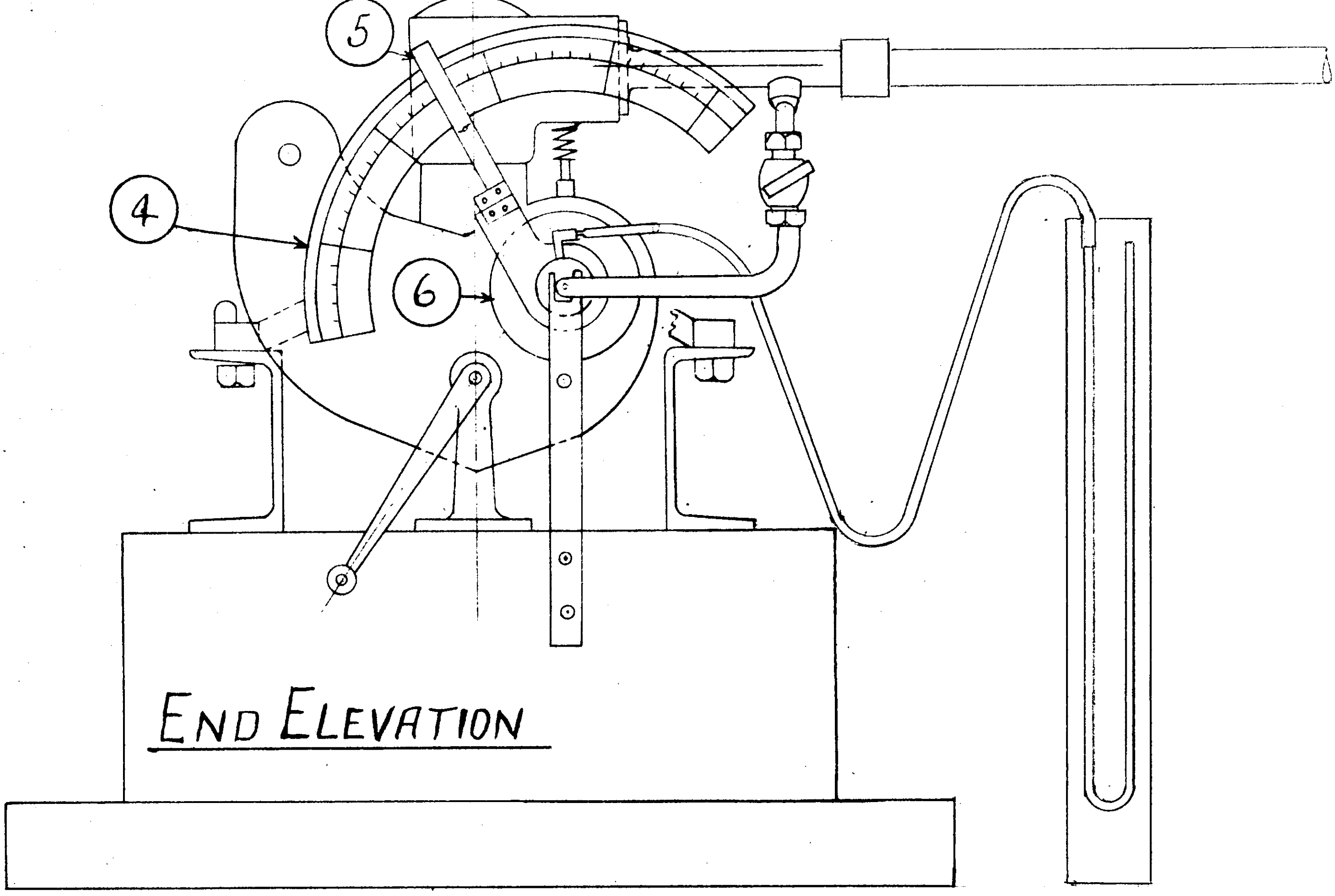
An Experimental Investigation

Of Exhaust Systems

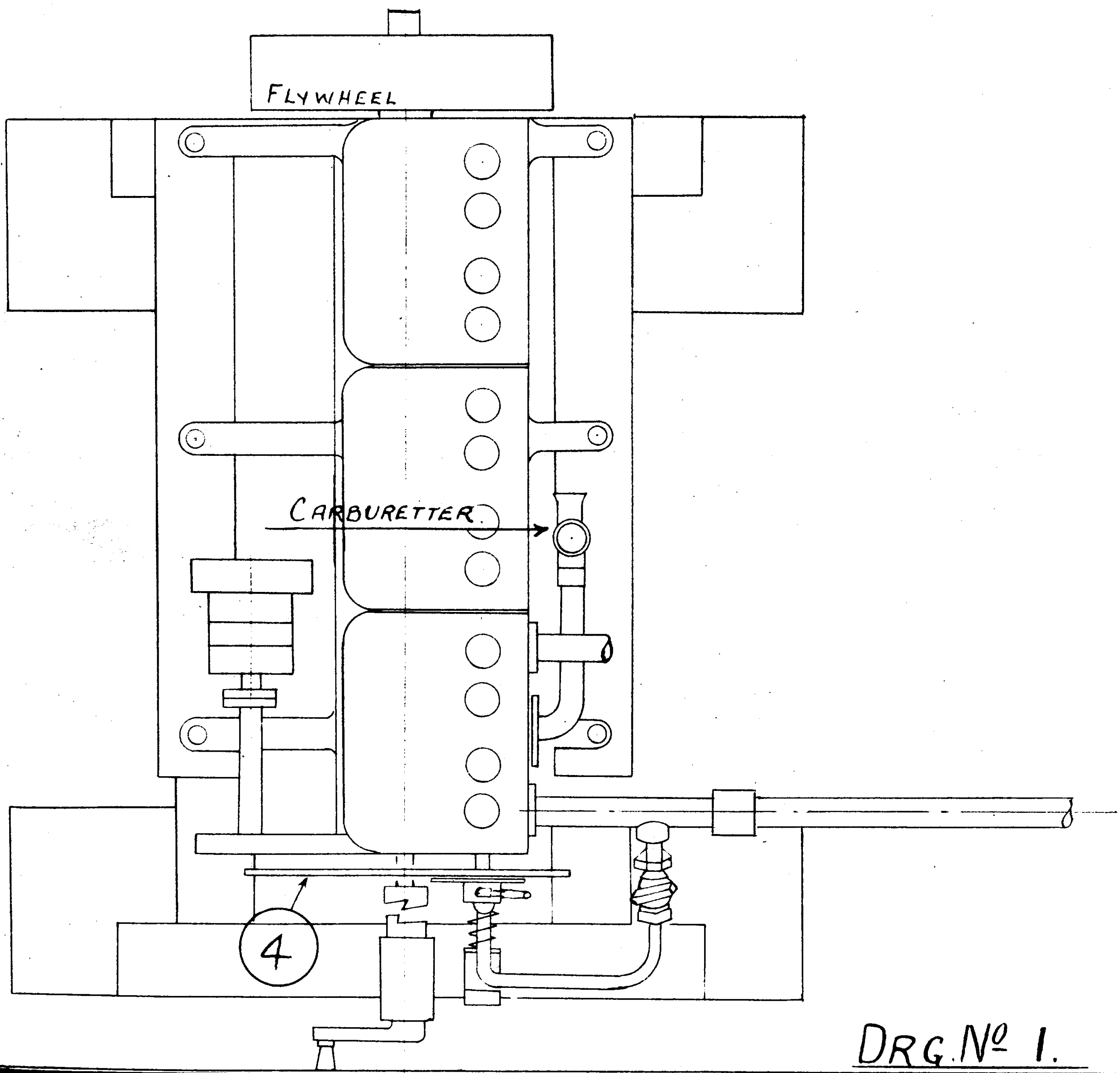
Of I. C. Engines.

By

John C. Morrison.

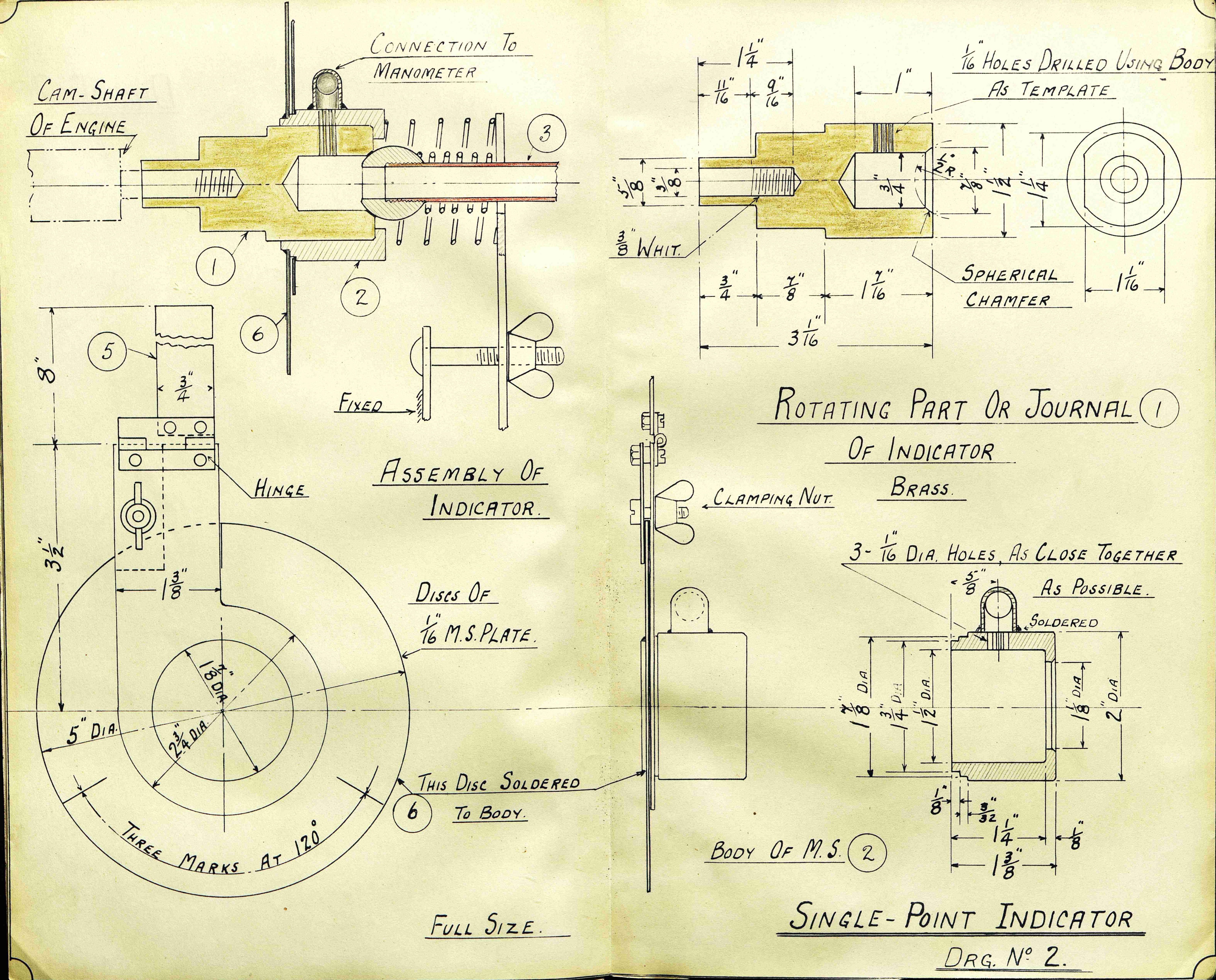


ARR^{GT} OF ENGINE AND INDICATOR.



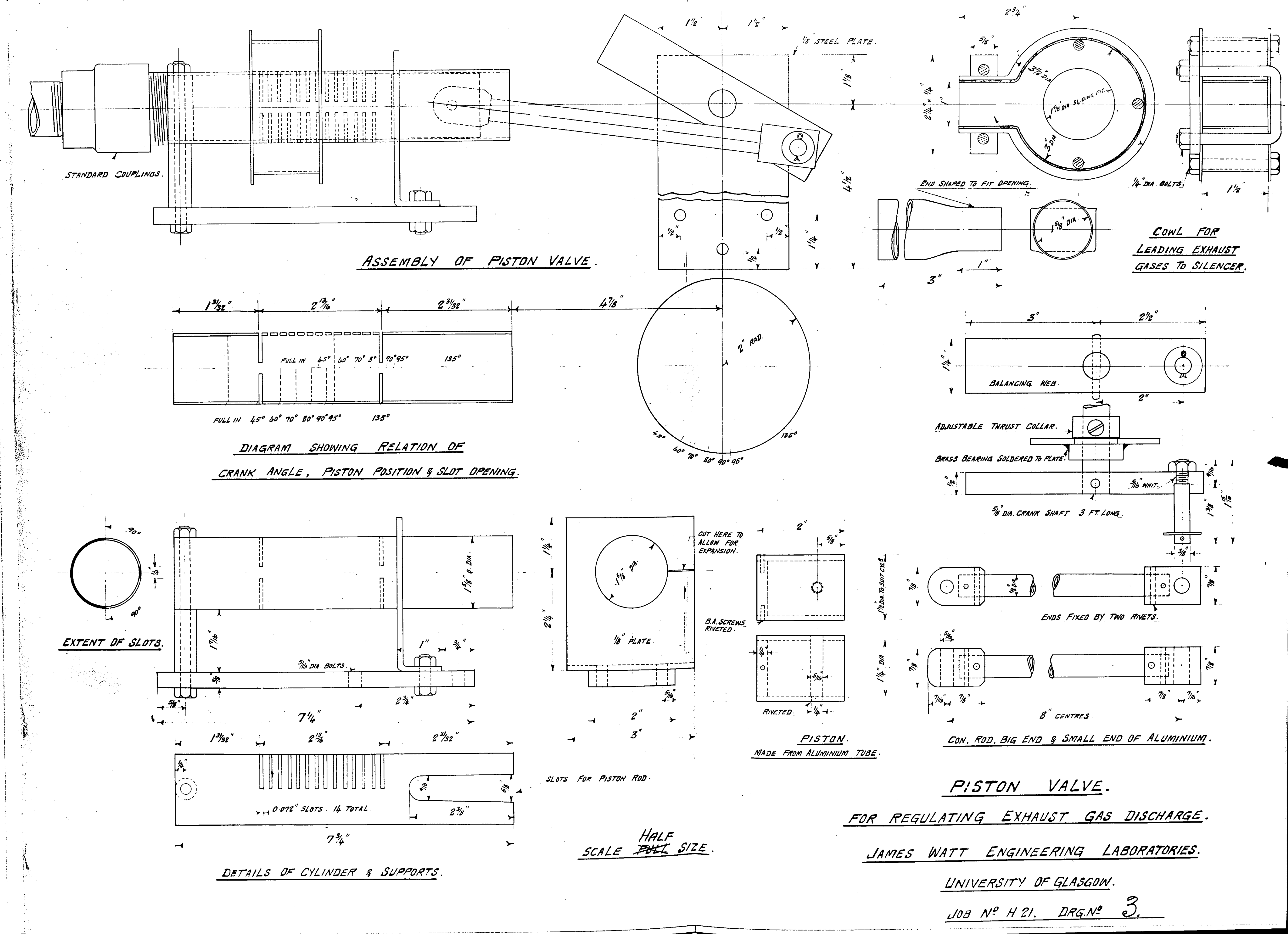
DRG. No 1.

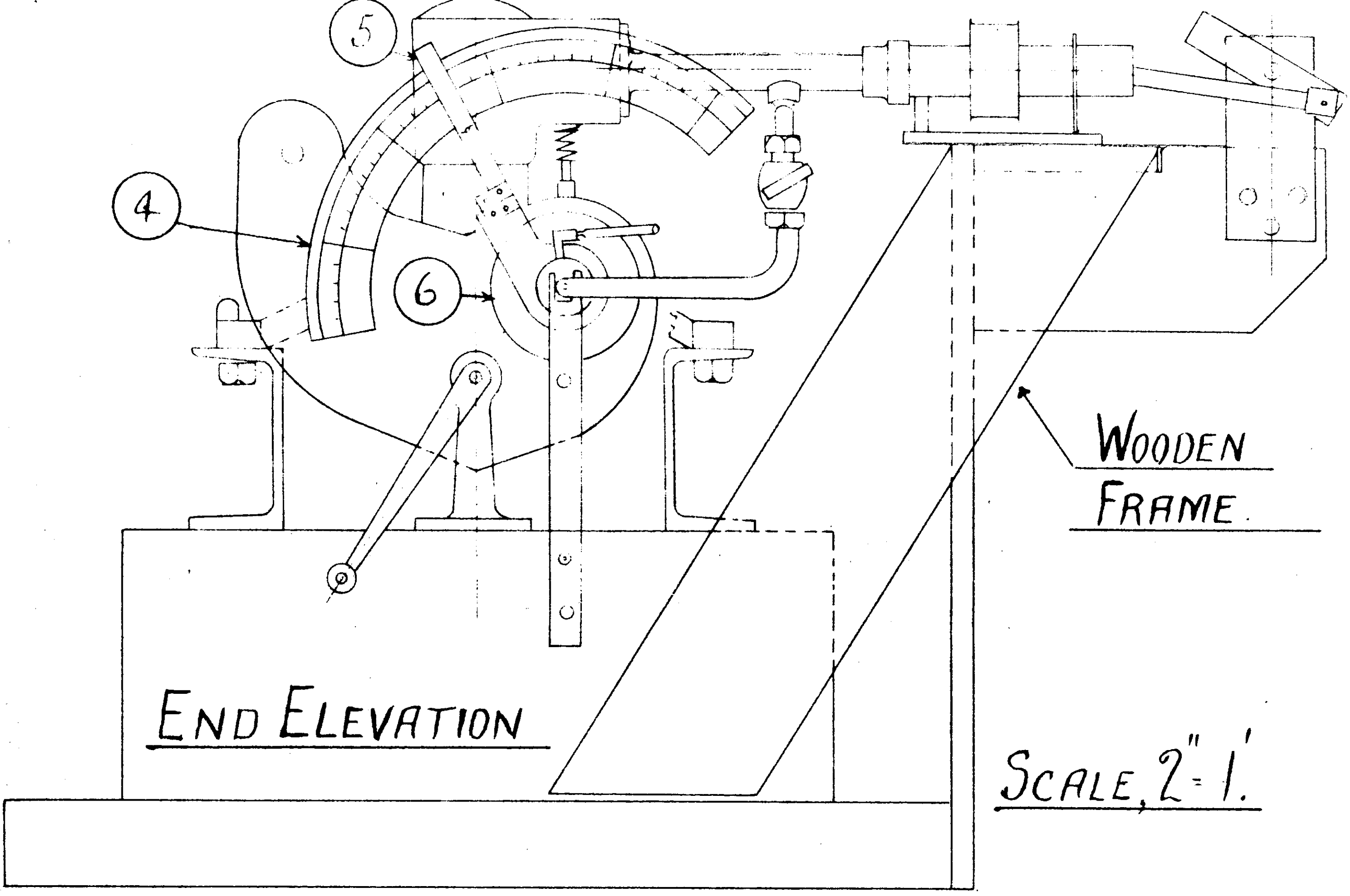
DRG. N° 2.



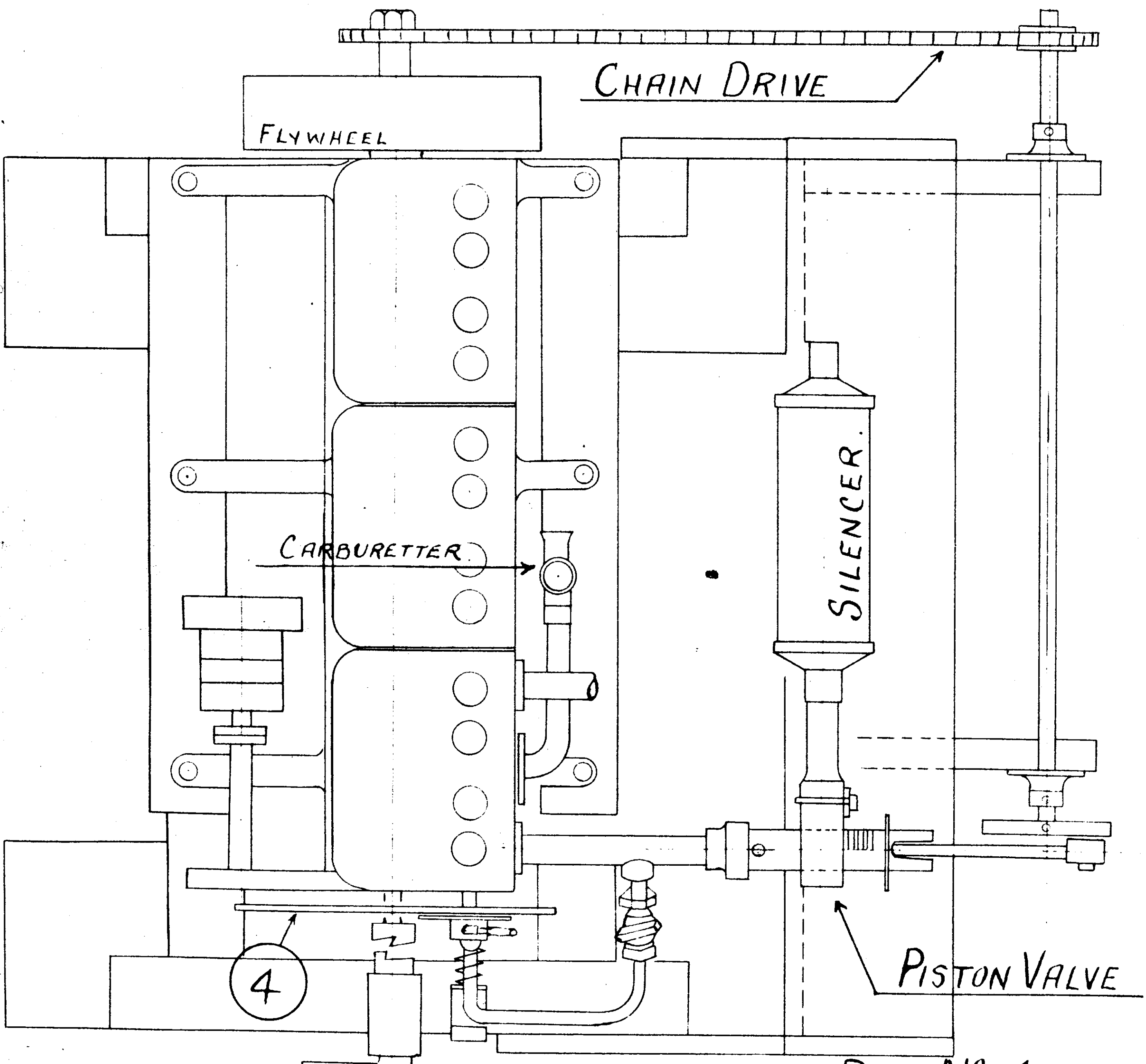
CAM-SHAFT OF ENGINE
CONNECTION TO MANOMETER
1
2
3
5
6
FIXED
ASSEMBLY OF INDICATOR.
HINGE
8"
3/4"
3 1/2"
1 3/8"
DISCS OF 1/16" M.S. PLATE.
1 7/8" DIA.
2 3/4" DIA.
5" DIA.
THREE MARKS AT 120°
THIS DISC SOLDERED TO BODY.
6
FULL SIZE.
1 1/4"
11/16"
9/16"
1"
1/16" HOLES DRILLED USING BODY AS TEMPLATE
5/8"
3/8"
3/4"
1/2"R.
7/8"
1 1/2"
1 1/4"
3/8" WHIT.
3/4"
7/8"
1 7/16"
3 1/16"
SPHERICAL CHAMFER
1 1/16"
ROTATING PART OR JOURNAL 1 OF INDICATOR
BRASS.
CLAMPING NUT.
3 - 1/16" DIA. HOLES, AS CLOSE TOGETHER AS POSSIBLE.
5/8"
SOLDERED
1 7/8" DIA.
1 3/4" DIA.
1 1/2" DIA.
1 1/8" DIA.
2" DIA.
1/8"
3/32"
1 1/4"
1/8"
1 3/8"
BODY OF M.S. 2
SINGLE-POINT INDICATOR
DRG. No 2.

Drg. Nº 3.

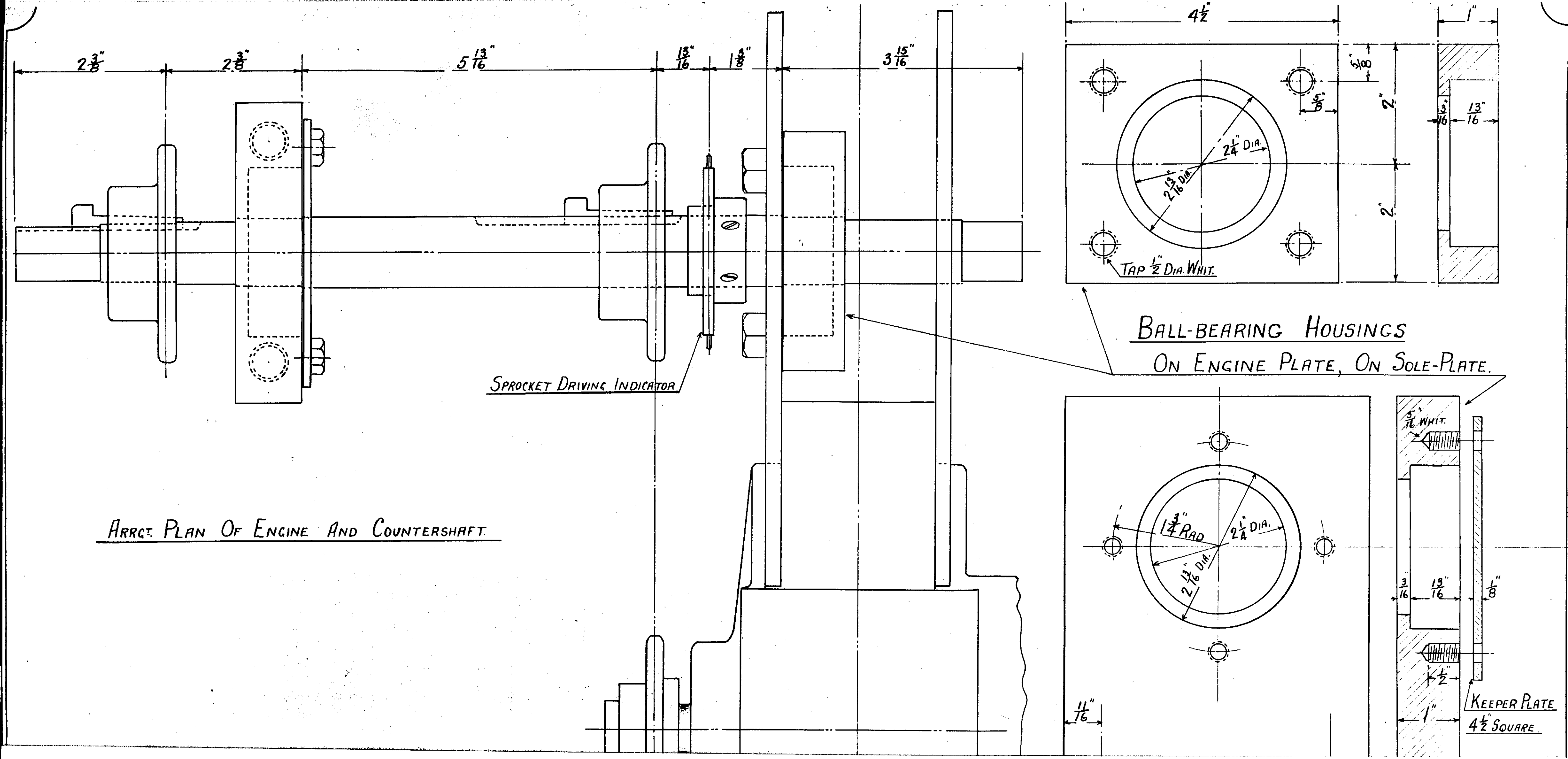

STANDARD COUPLINGS.
ASSEMBLY OF PISTON VALVE.
1/8" STEEL PLATE.
DIAGRAM SHOWING RELATION OF
CRANK ANGLE, PISTON POSITION & SLOT OPENING.
FULL IN 45° 60° 70° 80° 90° 95° 135°
2" RAD.
COWL FOR
LEADING EXHAUST
GASES TO SILENCER.
END SHAPED TO FIT OPENING.
1/4" DIA. BOLTS.
BALANCING WEB.
ADJUSTABLE THRUST COLLAR.
BRASS BEARING SOLDERED TO PLATE.
5/16" WHIT.
5/8" DIA. CRANK SHAFT 3 FT. LONG.
EXTENT OF SLOTS.
5/16" DIA. BOLTS.
1 5/8" O. DIA.
CUT HERE TO ALLOW FOR EXPANSION.
1/8" PLATE.
B.A. SCREWS RIVETED.
RIVETED.
PISTON.
MADE FROM ALUMINIUM TUBE.
ENDS FIXED BY TWO RIVETS.
8" CENTRES.
CON. ROD, BIG END & SMALL END OF ALUMINIUM.
0.072" SLOTS. 14 TOTAL.
SLOTS FOR PISTON ROD.
DETAILS OF CYLINDER & SUPPORTS.
SCALE HALF SIZE.
PISTON VALVE.
FOR REGULATING EXHAUST GAS DISCHARGE.
JAMES WATT ENGINEERING LABORATORIES.
UNIVERSITY OF GLASGOW.
JOB No H 21. DRG. No 3.



ARR^{GT} OF ENG., INDICATOR, & PISTON EXHST VALVE.

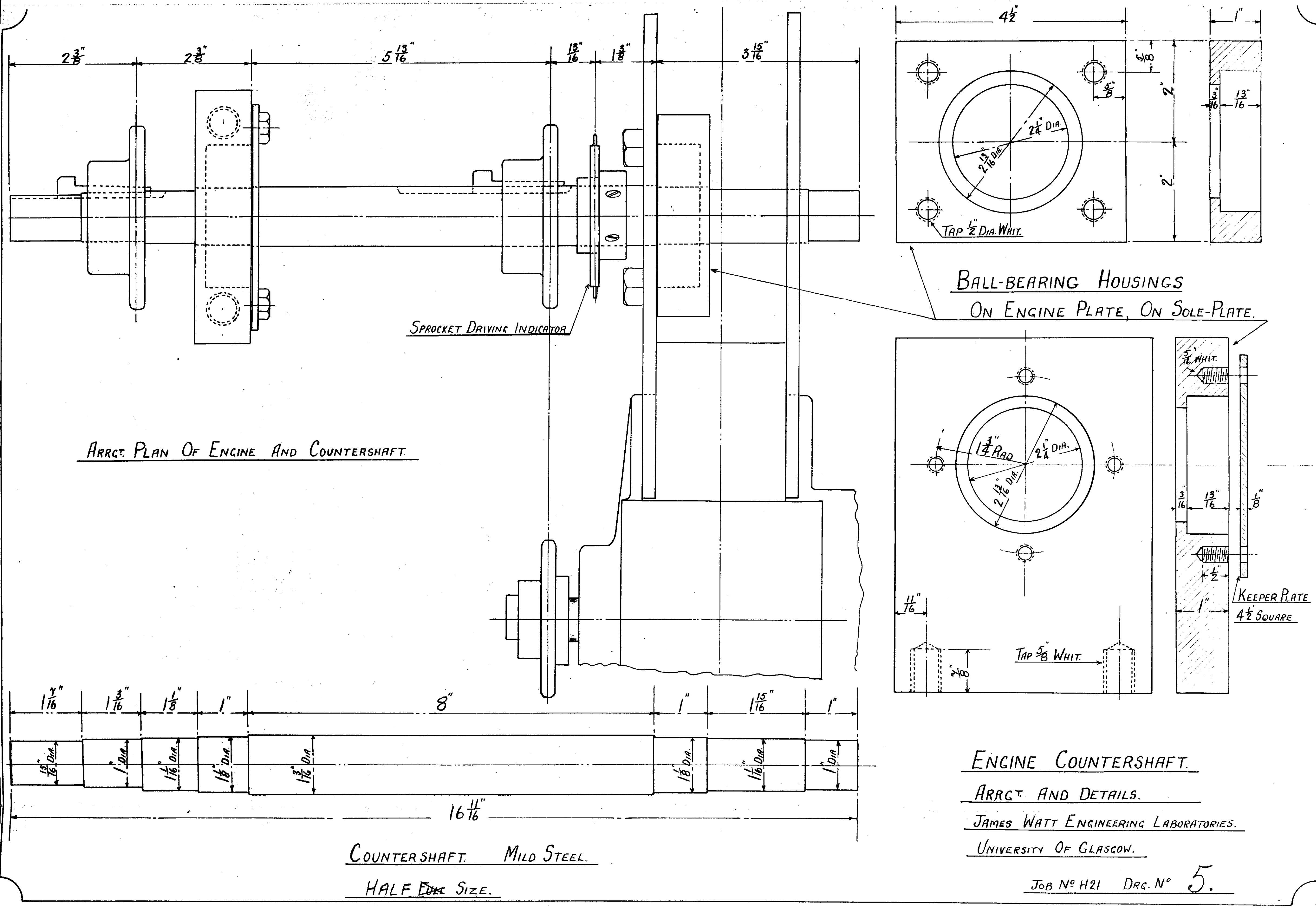


DRG. No 4.

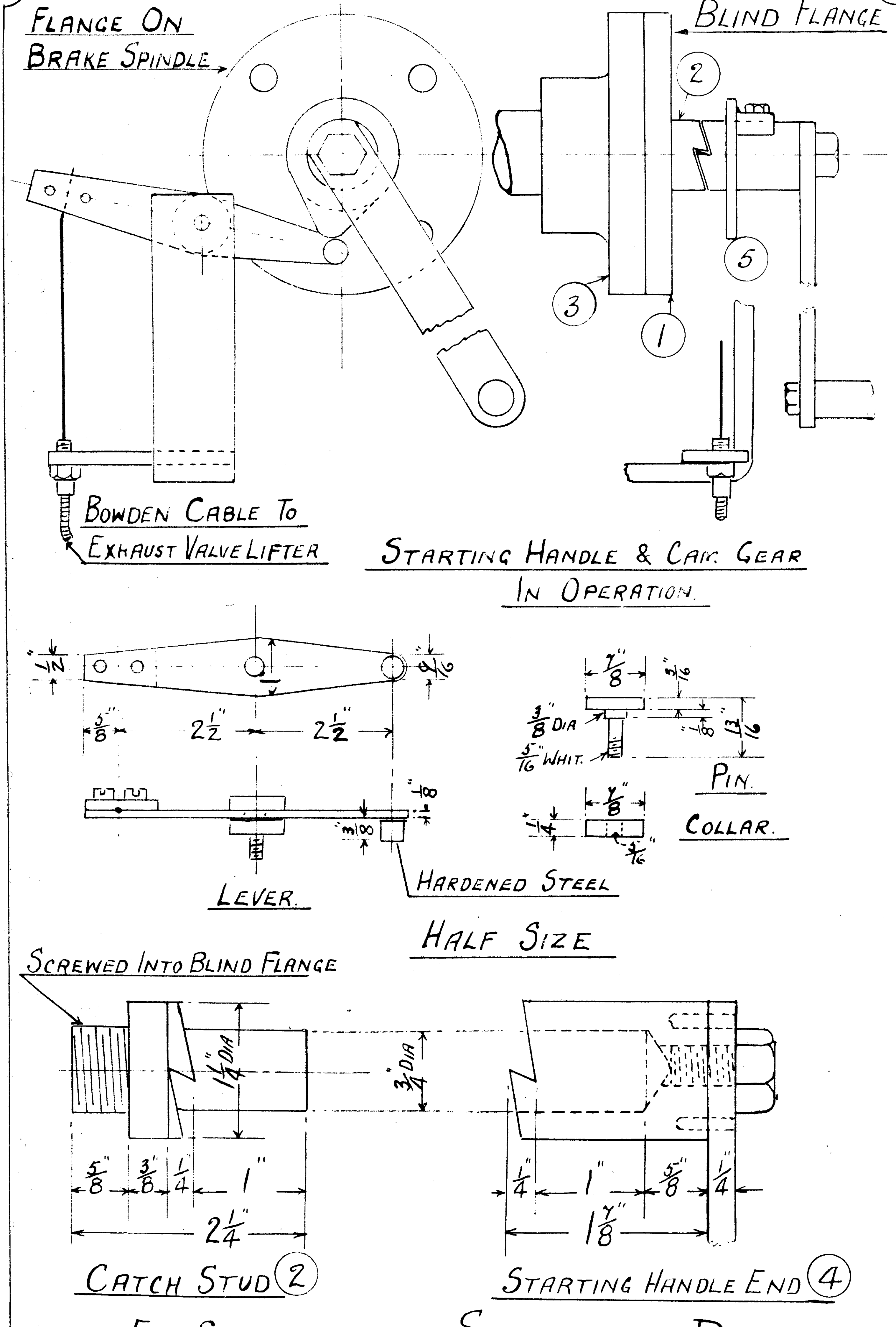


BALL-BEARING HOUSINGS
ON ENGINE PLATE, ON SOLE-PLATE.
TAP 1/2" DIA. WHIT.
5/16" WHIT.
KEEPER PLATE
4 1/2" SQUARE.
SPROCKET DRIVING INDICATOR
ARRGT. PLAN OF ENGINE AND COUNTERSHAFT.

DRG. 5.

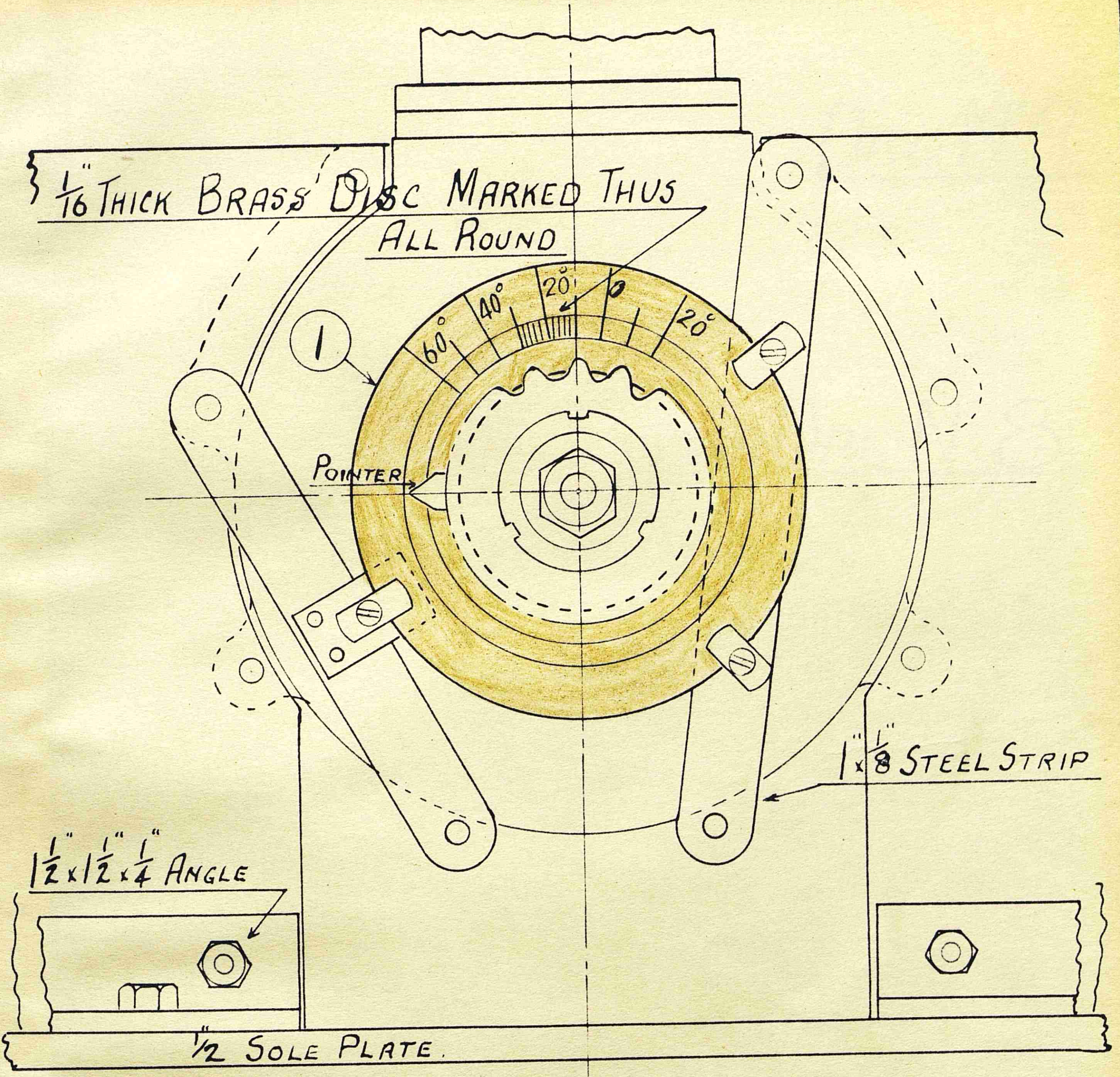


Ball-Bearing Housings
On Engine Plate, On Sole-Plate.
Tap 1/2" Dia. Whit.
Tap 5/8" Whit.
5/16" Whit.
Keeper Plate
4 1/2" Square
Sprocket Driving Indicator
Arrgt. Plan Of Engine And Countershaft.
Countershaft. Mild Steel.
Half ~~Full~~ Size.
Engine Countershaft.
Arrgt. And Details.
James Watt Engineering Laboratories.
University Of Glasgow.
Job No H21 Drg. No 5.

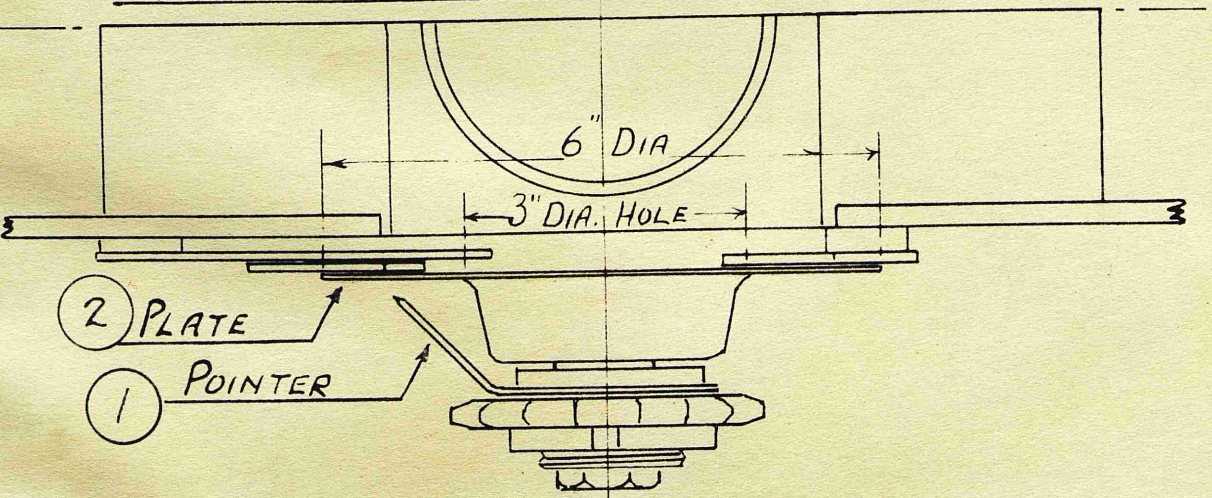


STARTING DEVICE.

JOB № H21. DRG № 6.



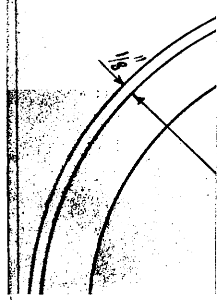
ELEVATION OF DRIVING SIDE OF ENGINE
SHOWING ENGINE PLATE FIXINGS AND
METHOD OF HOLDING INDICATOR PLATE.



CRANK ANGLE INDICATING DEVICE

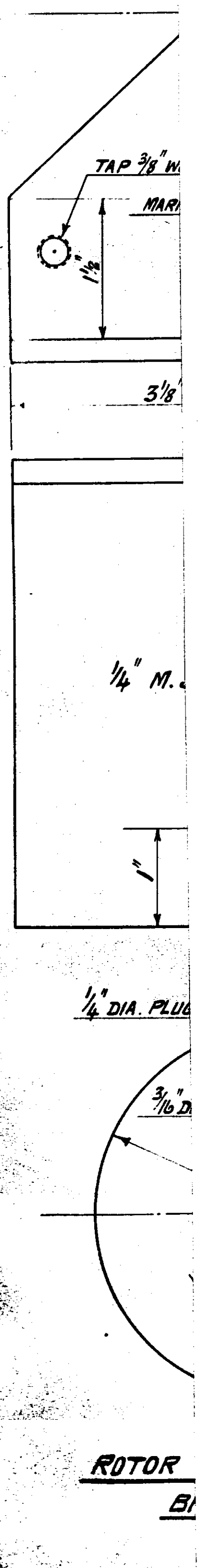
HALF SIZE

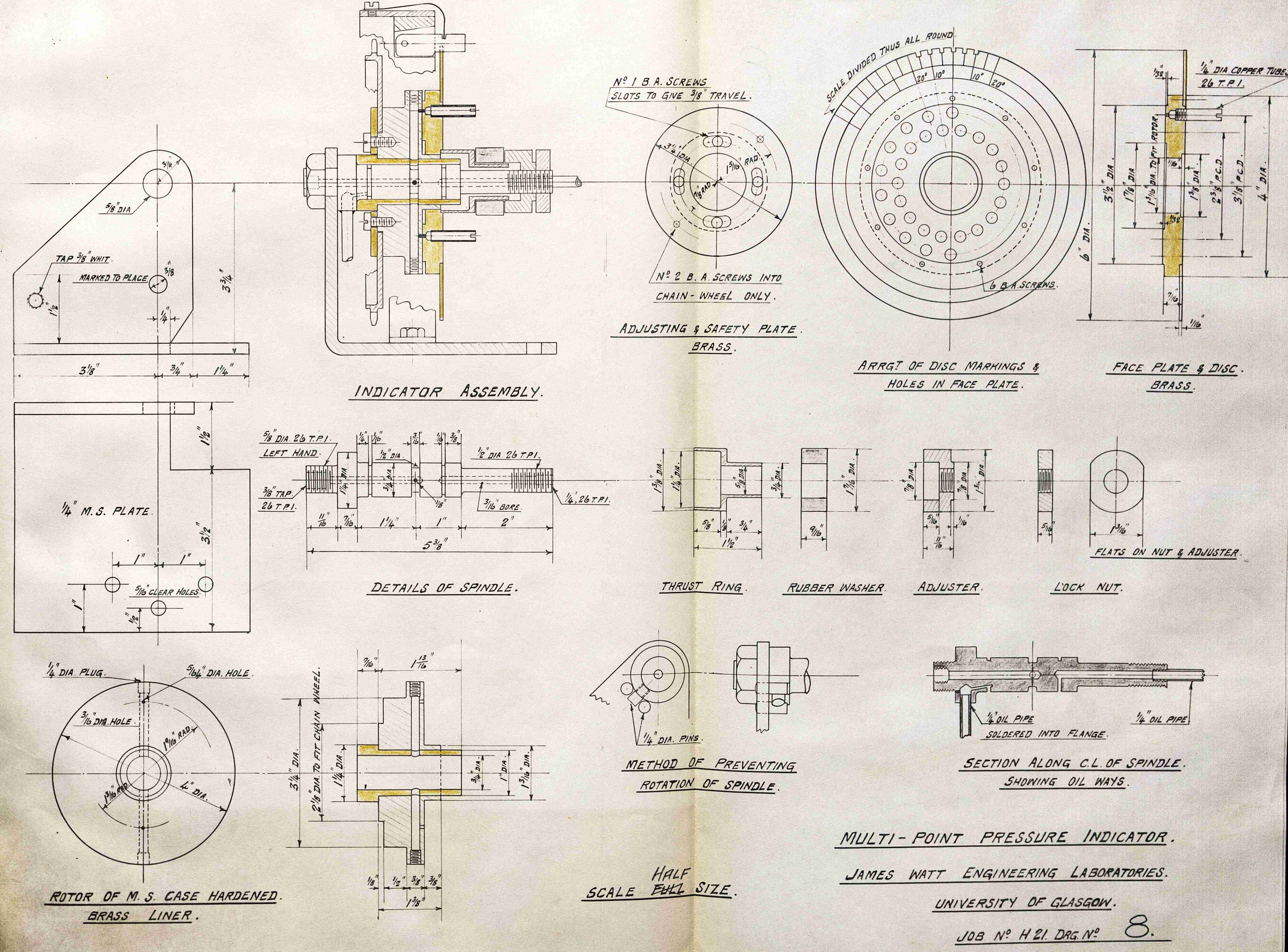
JOB No H21. DRG. No 7.



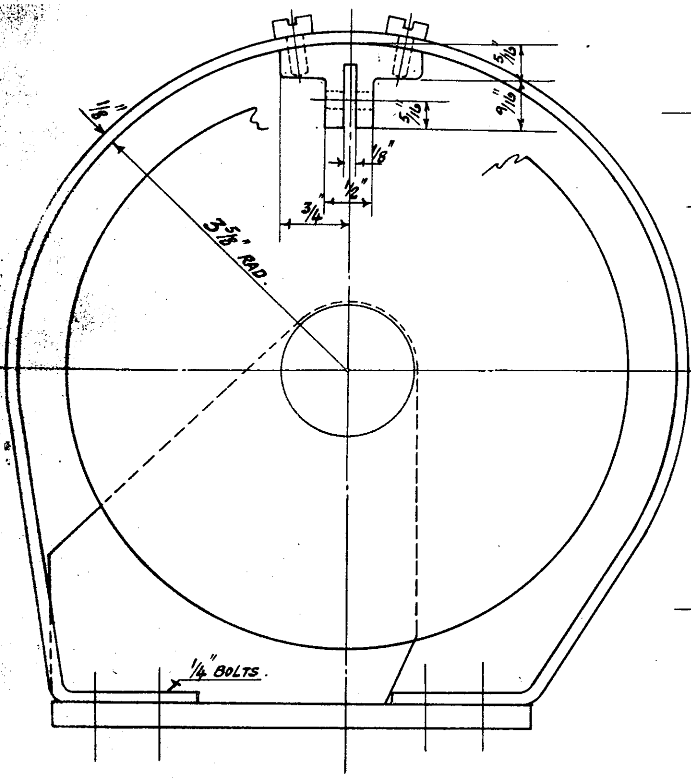
DRG. Nº 9.

DRG. Nº 8

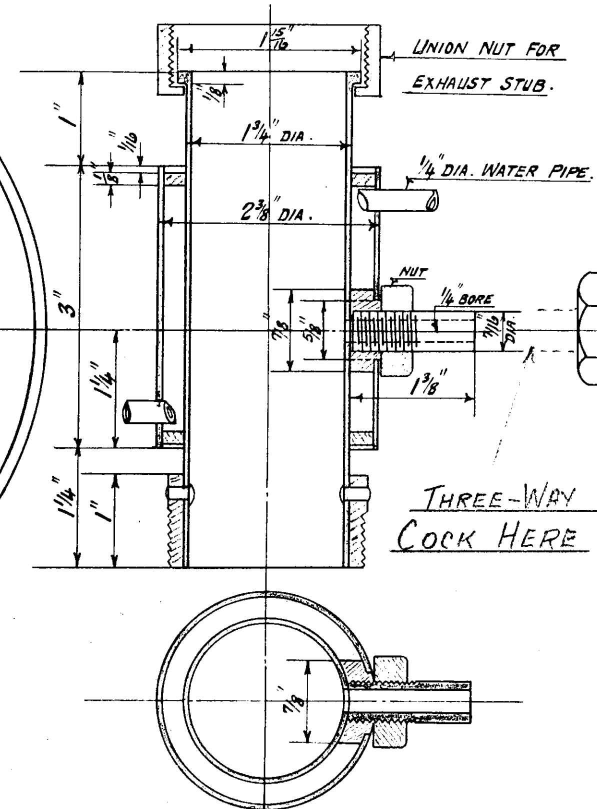




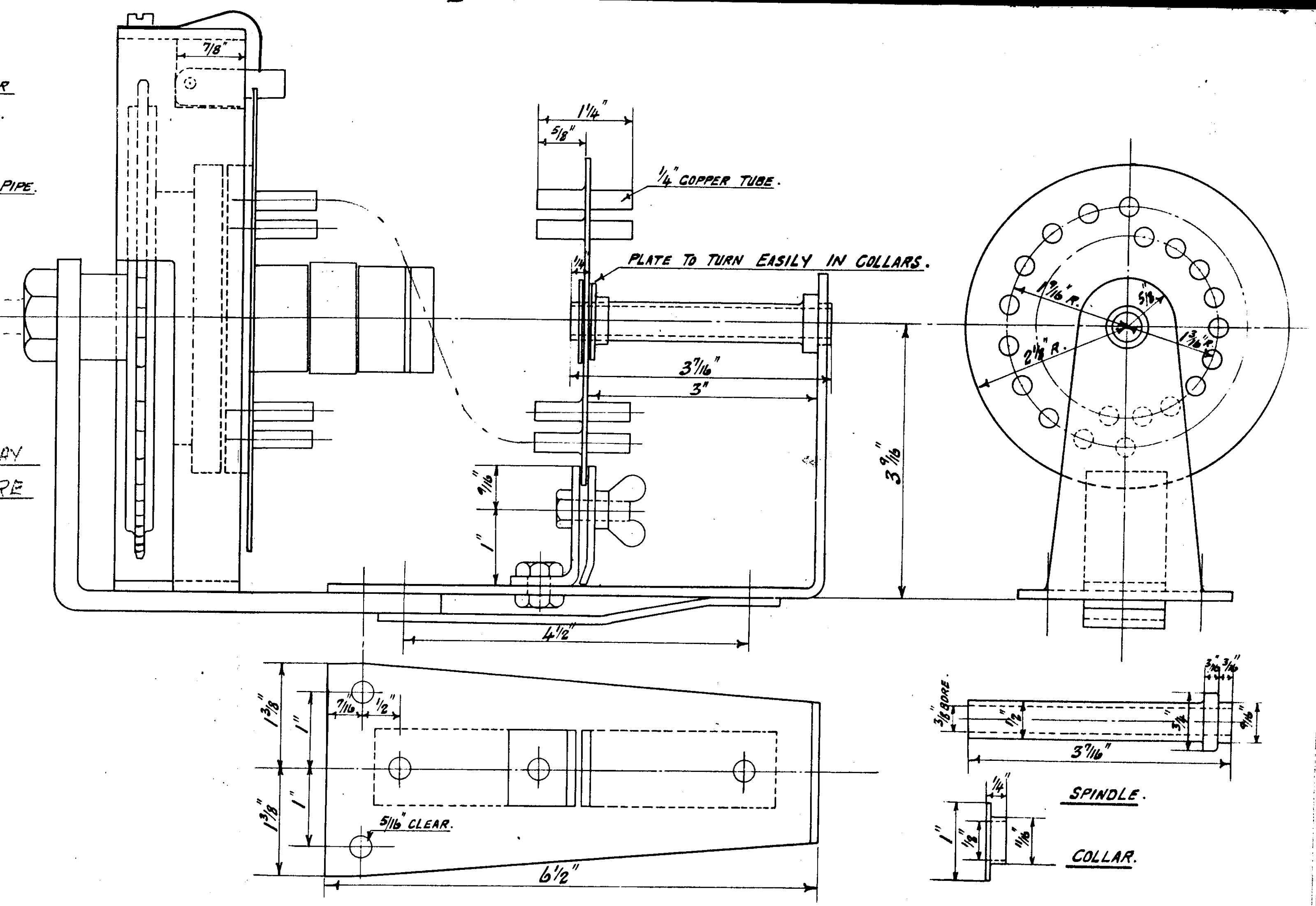
TAP 3/8" WHIT.
MARKED TO PLACE
5/8" DIA
1/4" M.S. PLATE.
5/16" CLEAR HOLES.
INDICATOR ASSEMBLY.
Nº 1 B.A. SCREWS
SLOTS TO GIVE 3/8" TRAVEL.
Nº 2 B.A. SCREWS INTO
CHAIN-WHEEL ONLY.
ADJUSTING & SAFETY PLATE.
BRASS.
SCALE DIVIDED THUS ALL ROUND.
6 B.A. SCREWS.
ARRGT. OF DISC MARKINGS &
HOLES IN FACE PLATE.
1/4" DIA COPPER TUBES
26 T.P.I.
FACE PLATE & DISC.
BRASS.
5/8" DIA. 26 T.P.I.
LEFT HAND.
3/8" TAP.
26 T.P.I.
1/2" DIA. 26 T.P.I.
3/16" BORE.
1/4", 26 T.P.I.
DETAILS OF SPINDLE.
THRUST RING.
RUBBER WASHER.
ADJUSTER.
LOCK NUT.
FLATS ON NUT & ADJUSTER.
1/4" DIA. PLUG.
5/64" DIA. HOLE.
3/16" DIA. HOLE.
ROTOR OF M.S. CASE HARDENED.
BRASS LINER.
2 1/8" DIA. TO FIT CHAIN WHEEL.
1/4" DIA. PINS.
METHOD OF PREVENTING
ROTATION OF SPINDLE.
1/4" OIL PIPE
SOLDERED INTO FLANGE.
1/4" OIL PIPE
SECTION ALONG C.L. OF SPINDLE.
SHOWING OIL WAYS.
SCALE ~~FULL~~ HALF SIZE.
MULTI-POINT PRESSURE INDICATOR.
JAMES WATT ENGINEERING LABORATORIES.
UNIVERSITY OF GLASGOW.
JOB Nº H 21. DRG. Nº 8.



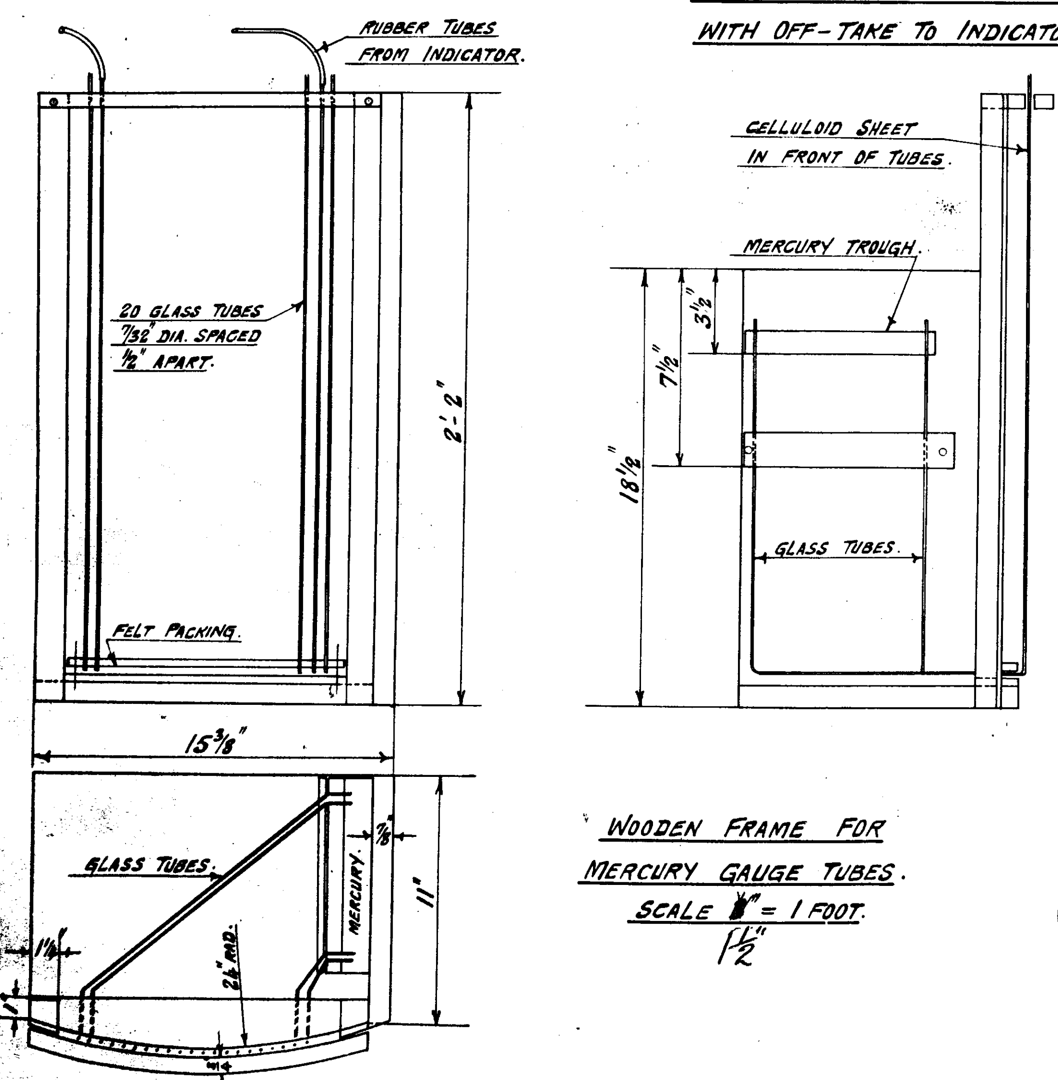
CHAIN GUARD & STOP FOR INDICATOR FACE PLATE.



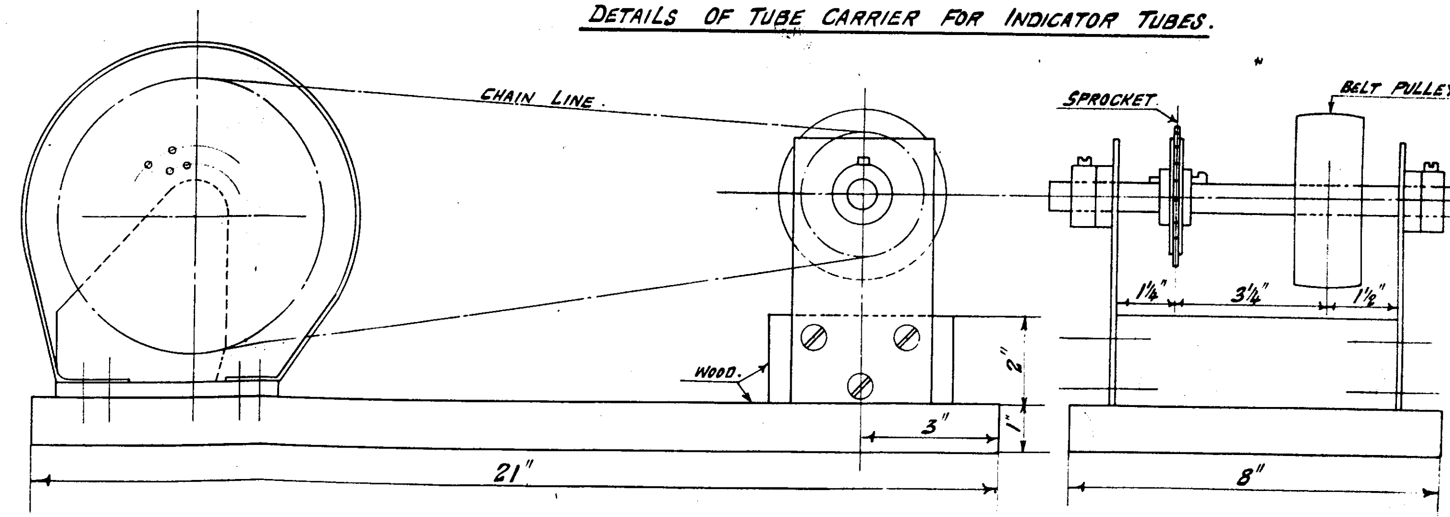
WATER COOLED EXHAUST PIPE WITH OFF-TAKE TO INDICATOR.



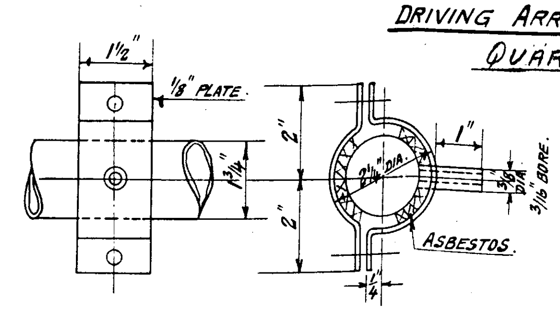
DETAILS OF TUBE CARRIER FOR INDICATOR TUBES.



WOODEN FRAME FOR MERCURY GAUGE TUBES.
SCALE 1 1/2" = 1 FOOT.



DRIVING ARRANGEMENT FOR INDICATOR TESTING.
QUARTER ~~HALF~~ SIZE.



OFF-TAKE FROM EXHAUST PIPE TO INDICATOR, FOR ATTACHMENT AT ANY POINT ON PIPE.
QUARTER ~~HALF~~ SIZE.

MULTI-POINT INDICATOR.

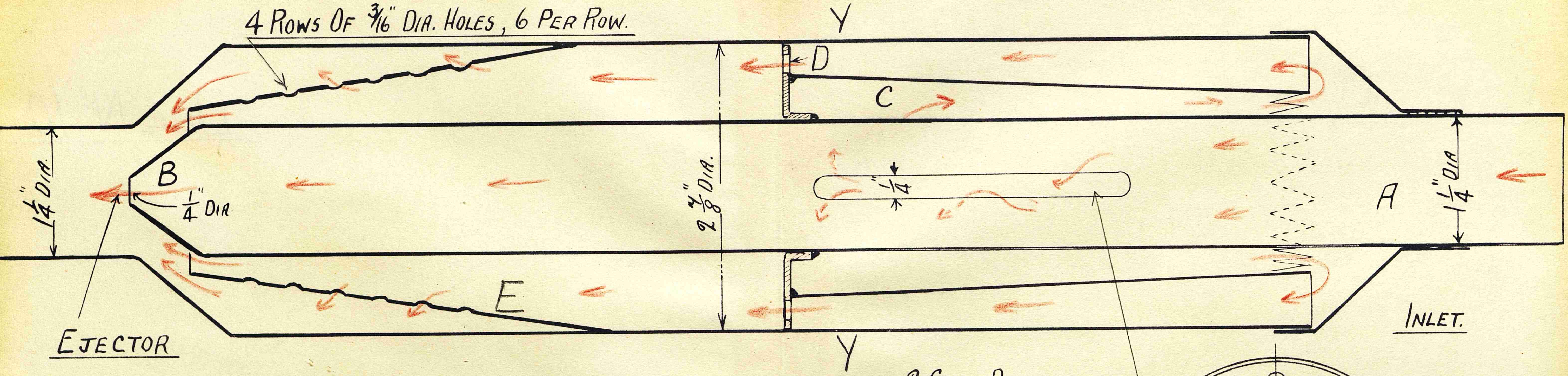
AUXILIARY FITTINGS & TESTING DEVICE.

JAMES WATT ENGINEERING LABORATORIES.

UNIVERSITY OF GLASGOW.

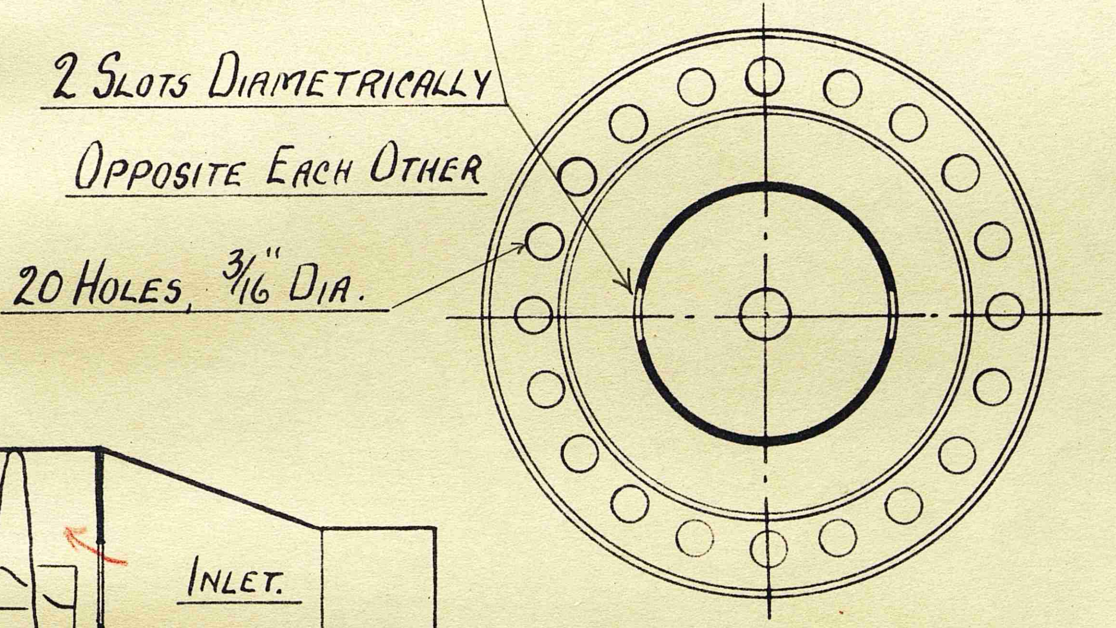
JOB Nº H21. DRG Nº 9.

DRG. Nº 10.



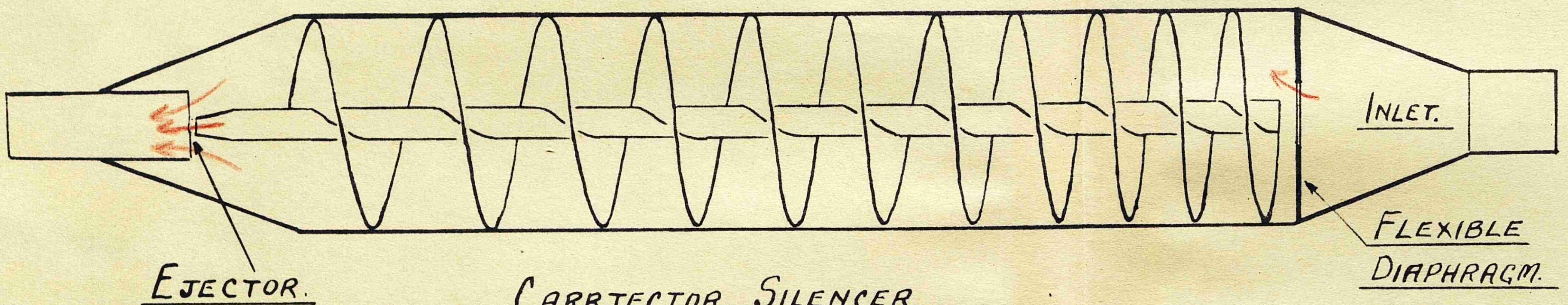
HOWARTH SILENCER.

FULL SIZE.



SECTION "YY"

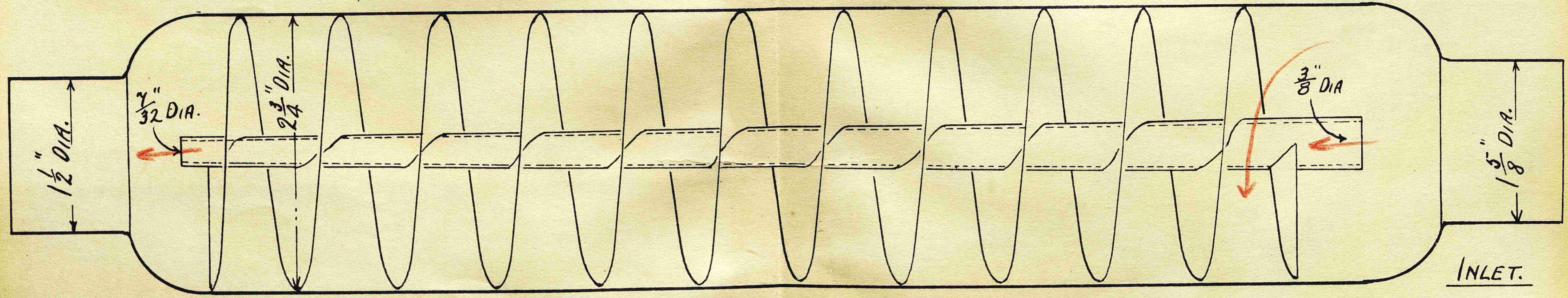
LOOKING TO LEFT.



CARBJECTOR SILENCER

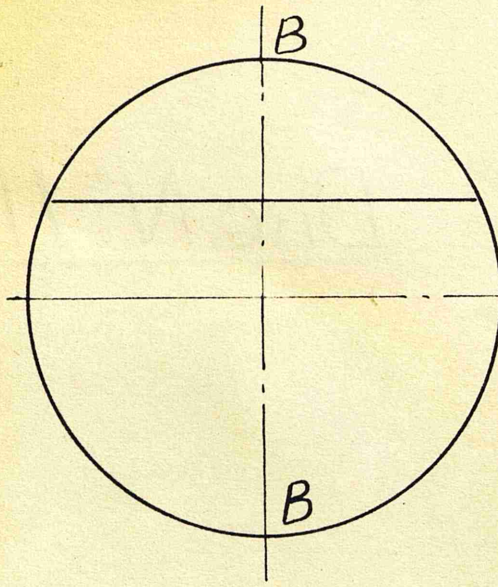
AS DETAILED IN THE PATENT SPECIFICATION.

NOT TO SCALE.



CARBJECTOR SILENCER AS USED IN TESTS. FULL SIZE. DRG. No 10.

DRG. Nº 11.



SECTION "AA"

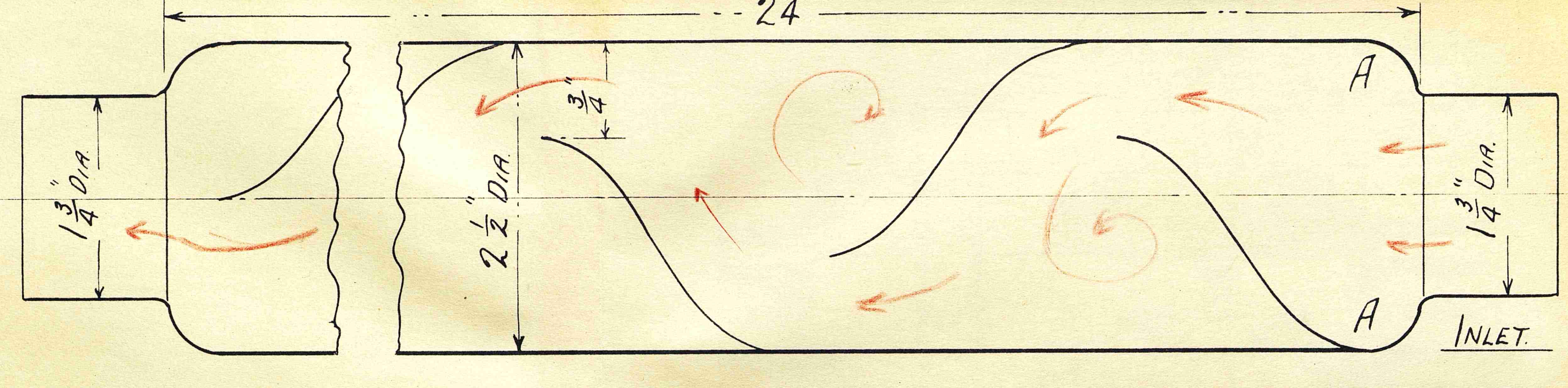
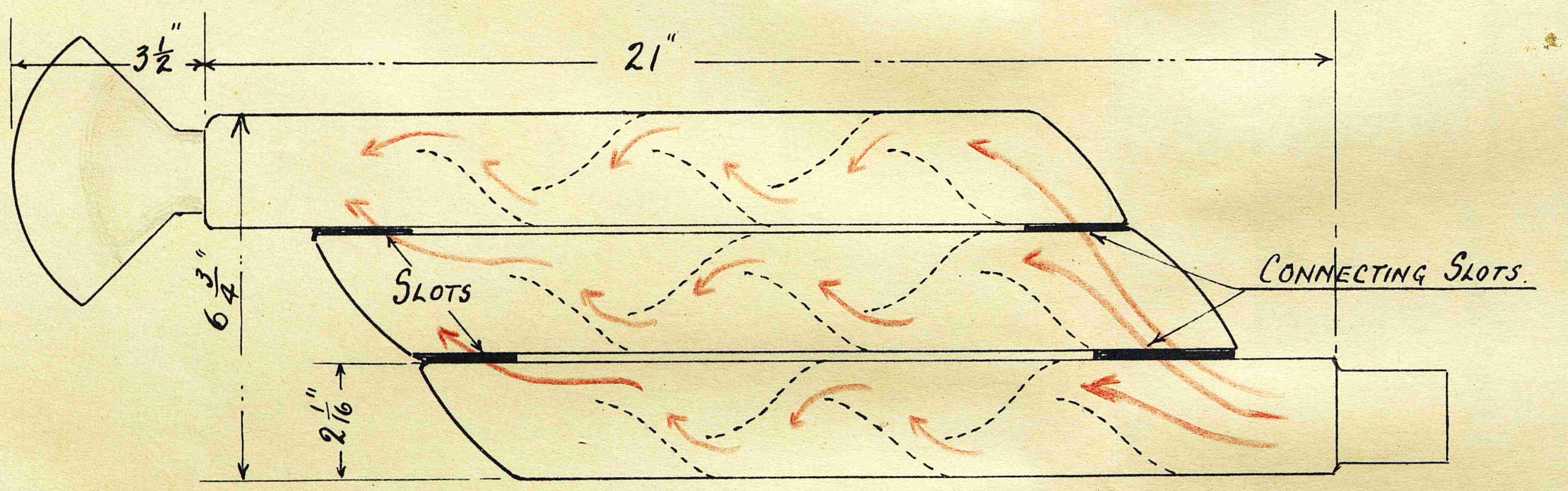
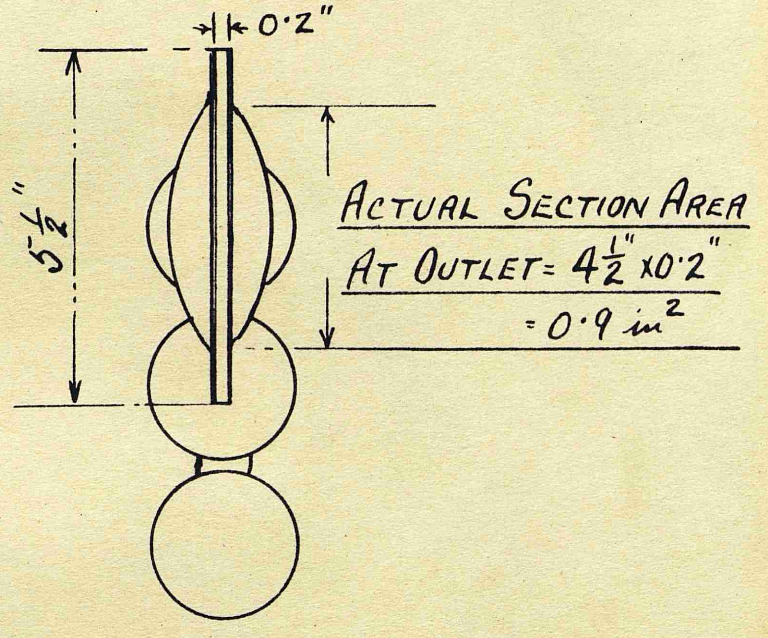


FIG. 1.

VORTEX TUBE. FULL SIZE. SECTION "BB"



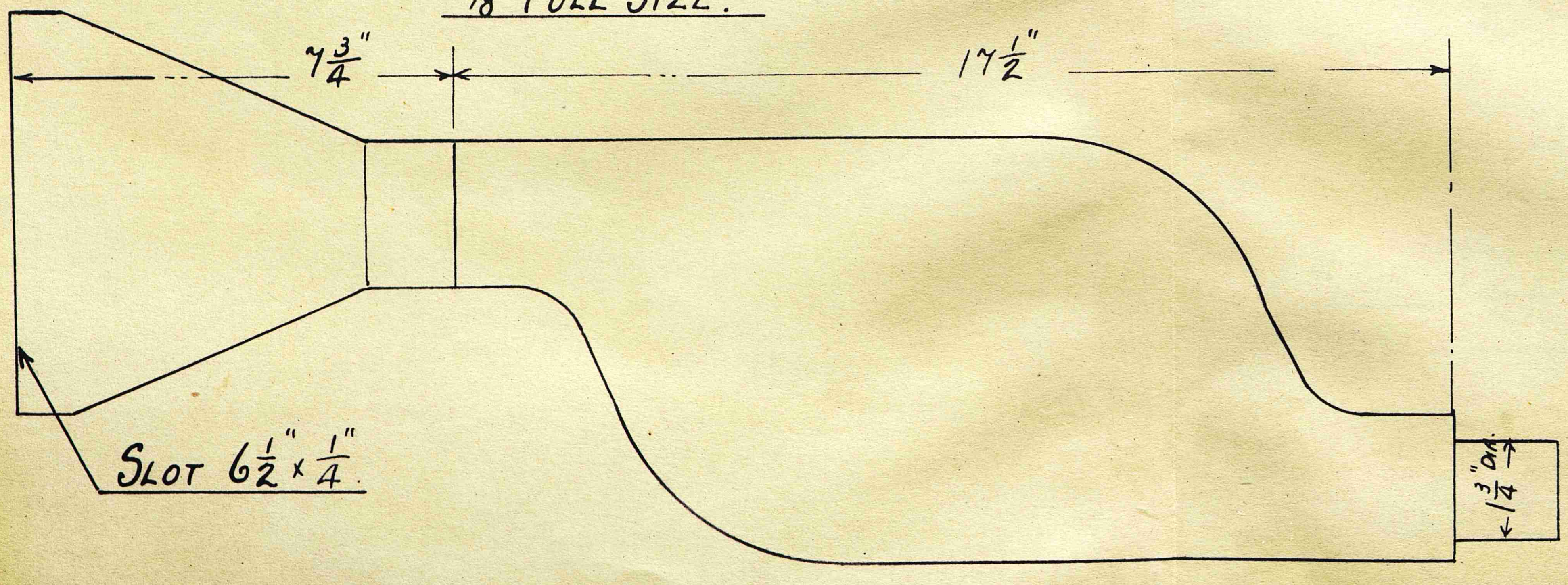


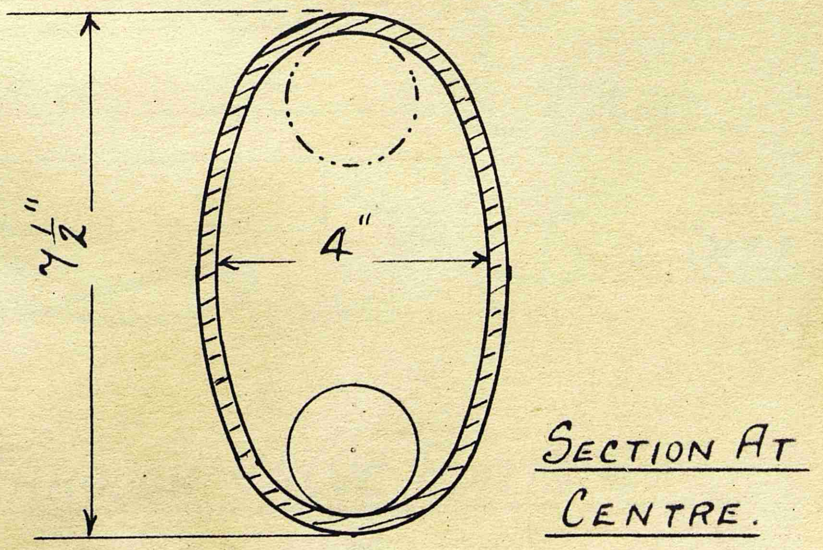
VORTEX STANDARD SILENCER. FIG. 2.

3/8 FULL SIZE.

MATERIAL:- LIGHT SHEET STEEL.

CAST ALUMINIUM, 1/4" THICK.

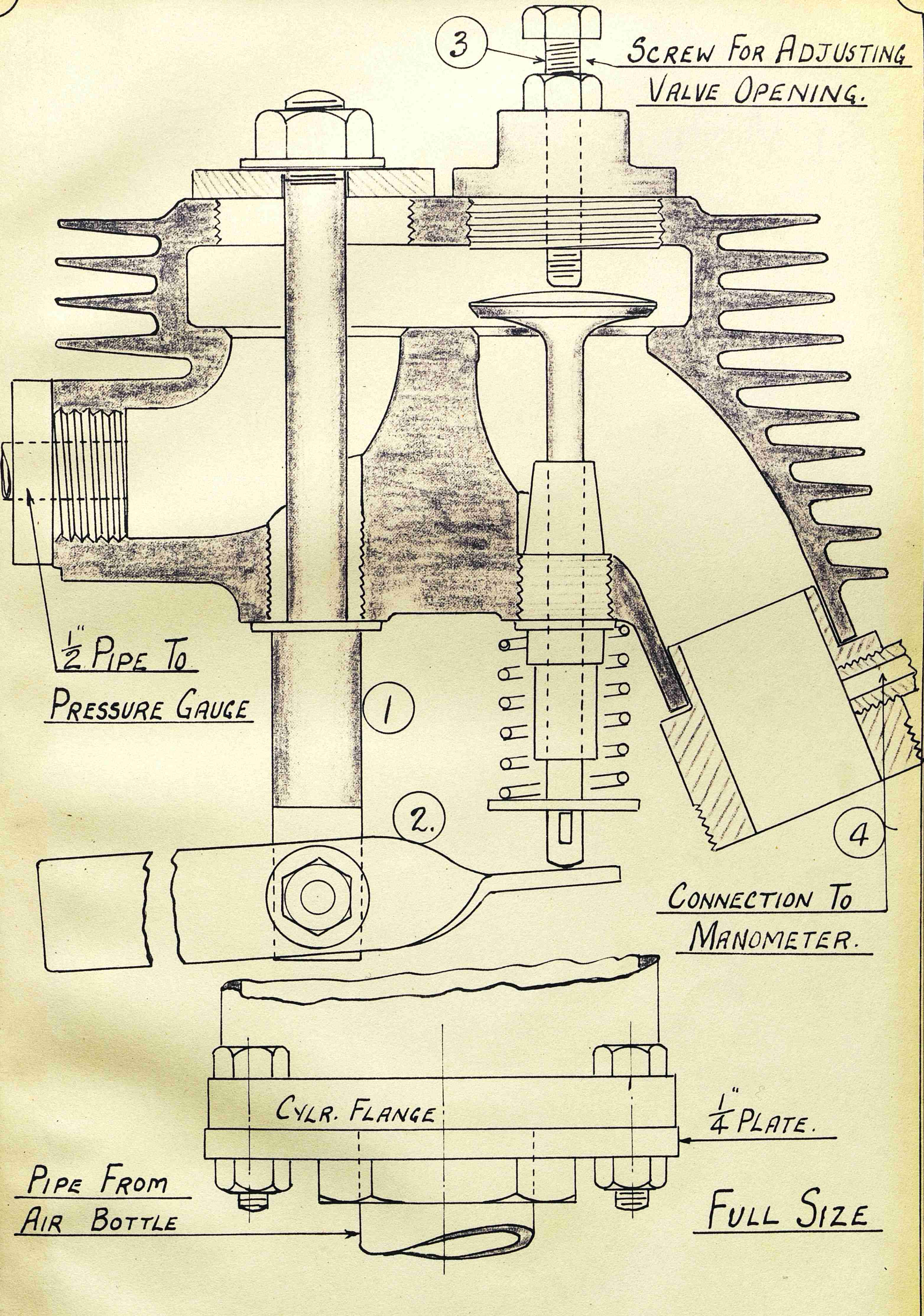




SECTION AT CENTRE.

BROOKLANDS STANDARD SILENCER. 3/8 FULL SIZE.

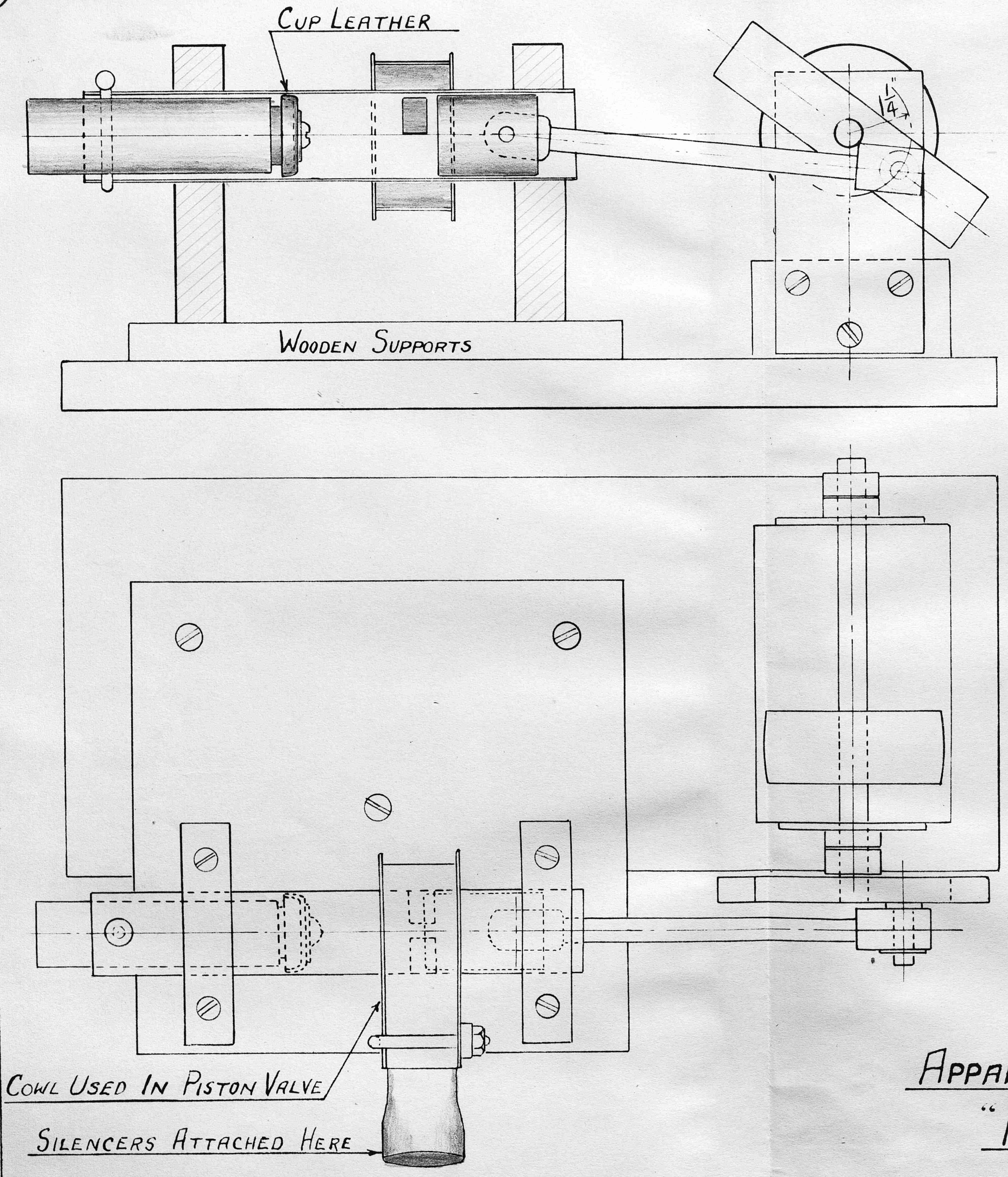
DRG. N° 11.



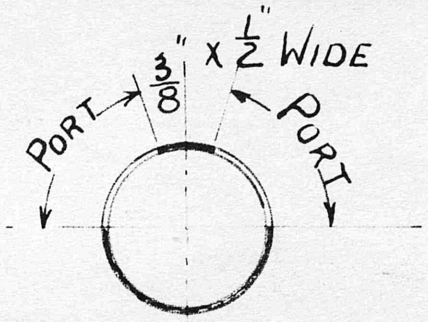
SECTION THROUGH VALVE PORTS OF SPECIAL CYLR. FOR NOISE TESTING.

JOB Nº H21. DRG. Nº 12.

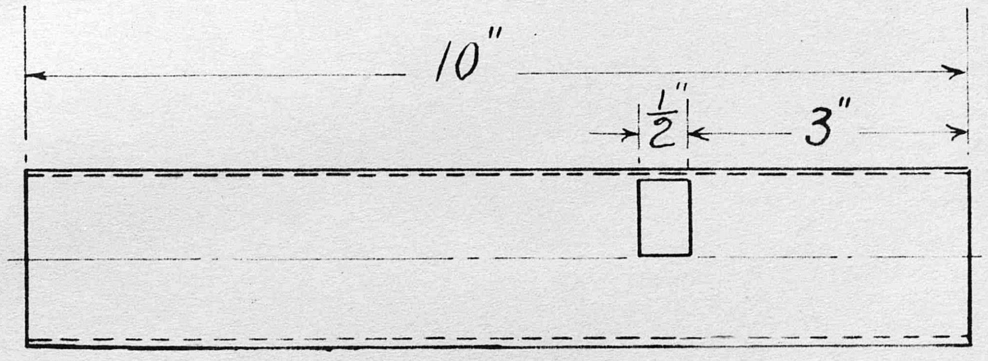
Drg. Nº 13.



HALF SIZE



SECTION THROUGH CYL^R^
SHOWING EXTENT OF PORTS



CYLR. MADE FROM $1\frac{5}{8}$" DIA.
S.D. STEEL TUBING.

SAME PISTON & CON. ROD AS USED
IN PISTON VALVE, SEE DRG. Nº 3.

CRANK SHAFT, ETC, AS USED IN
INDICATOR TESTING DEVICE, DRG. Nº 9.

APPARATUS FOR PRODUCING "IMPACT" NOISE

JOB Nº H21. DRG. Nº 13.

INDICATOR CURVES

ACCOMPANYING THESIS ON

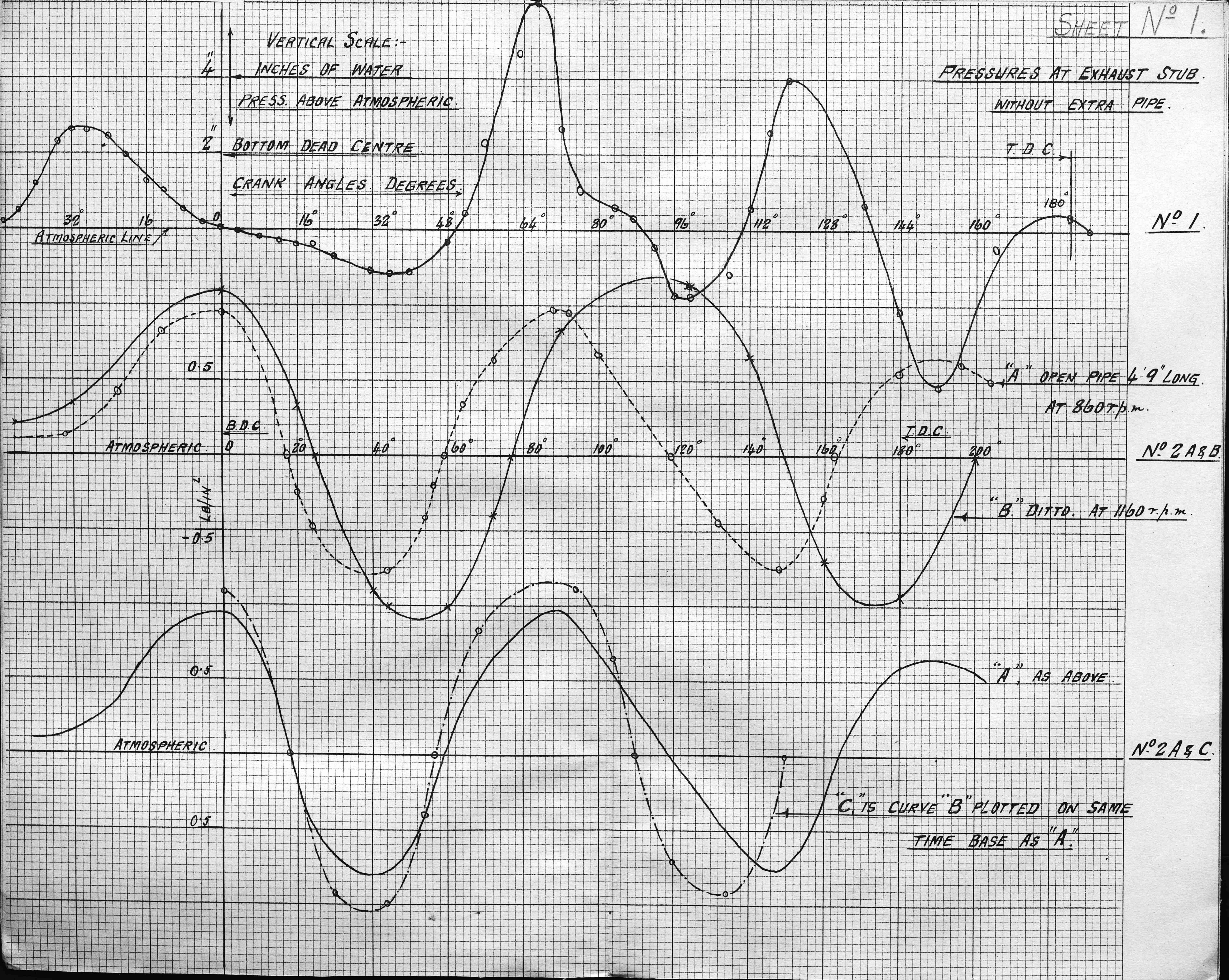
AN EXPERIMENTAL INVESTIGATION

OF EXHAUST SYSTEMS

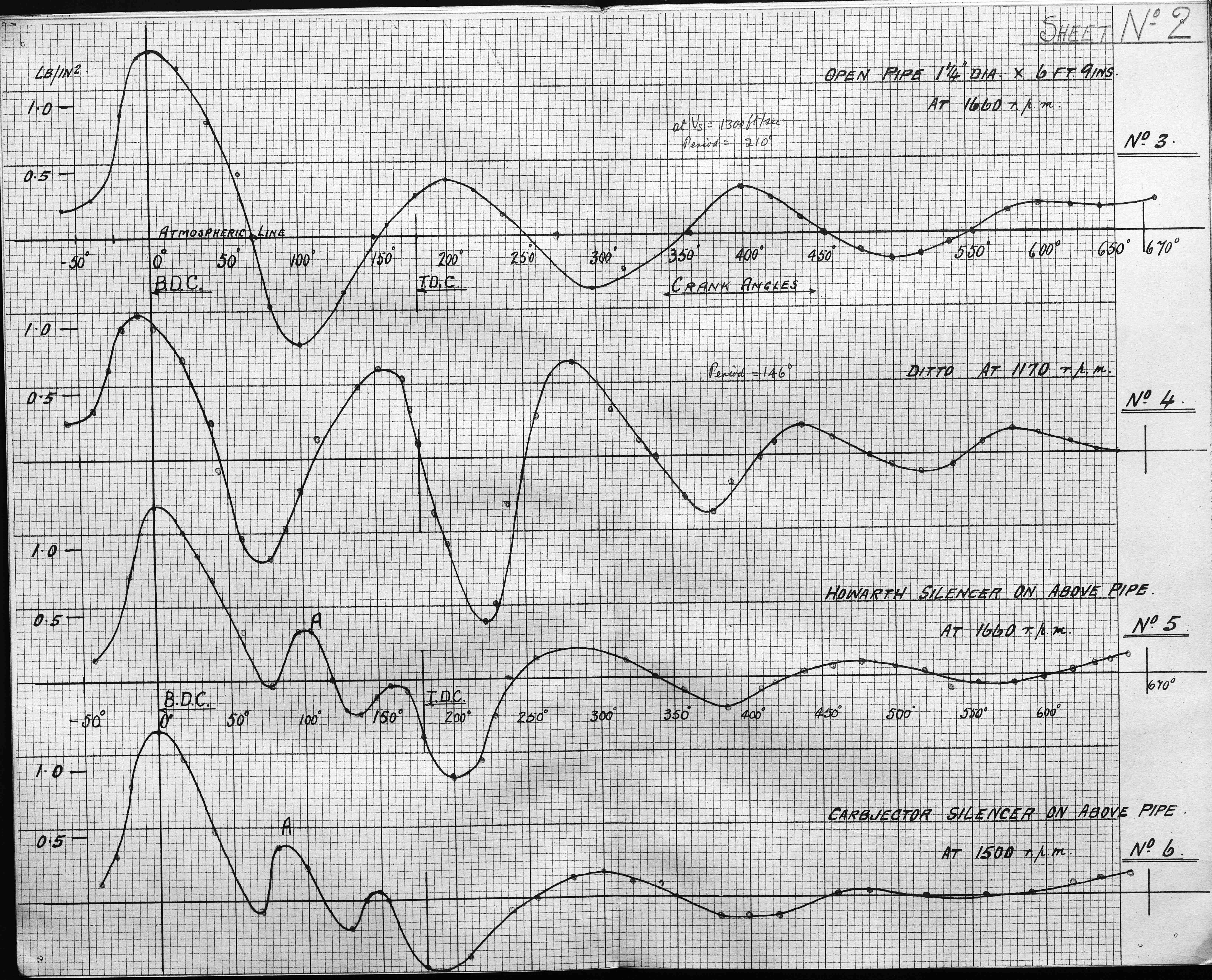
OF I. C. ENGINES.

BY

JOHN C. MORRISON.

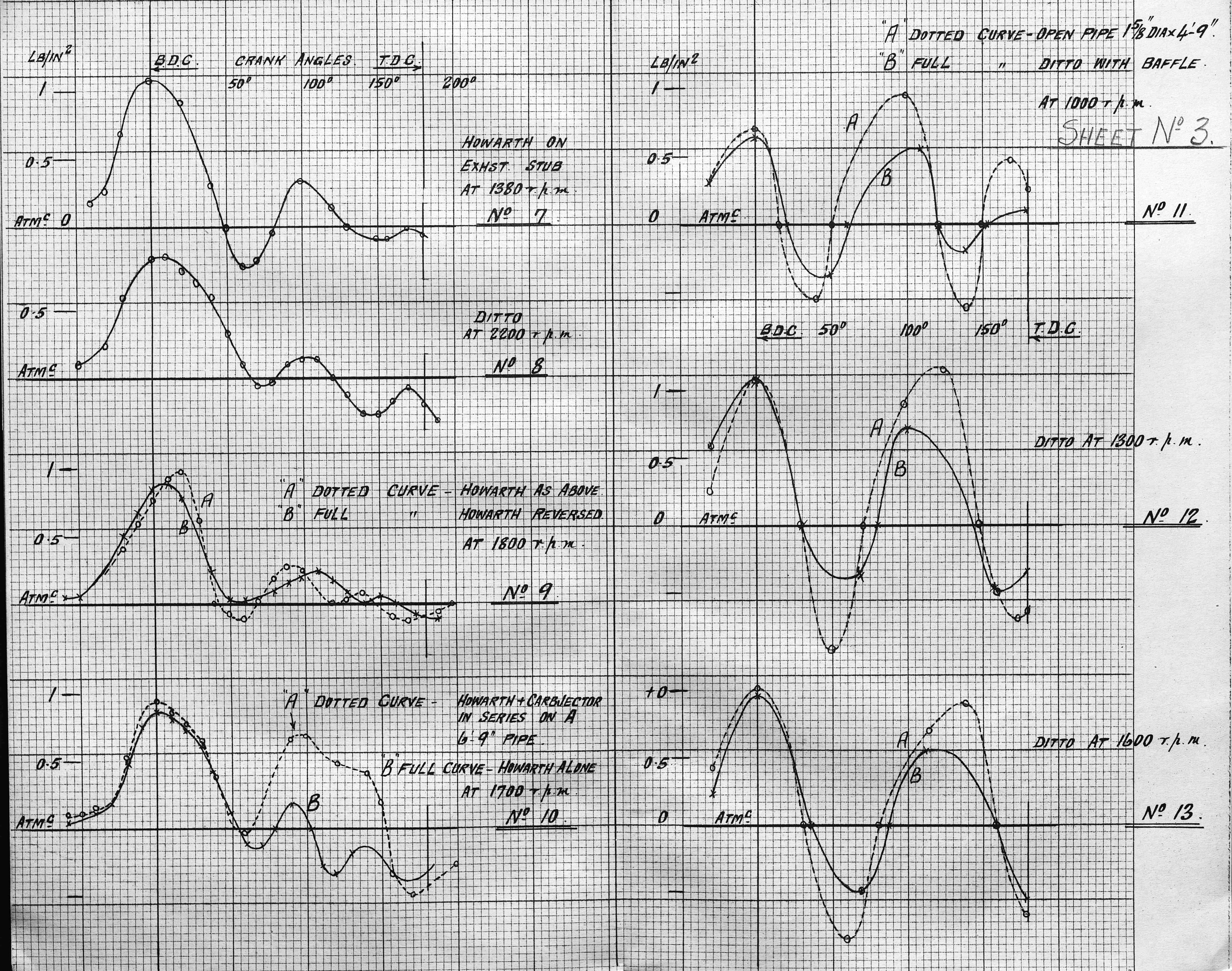

SHEET Nº 1.
VERTICAL SCALE:-
INCHES OF WATER
PRESS. ABOVE ATMOSPHERIC
BOTTOM DEAD CENTRE
CRANK ANGLES. DEGREES.
PRESSURES AT EXHAUST STUB.
WITHOUT EXTRA PIPE.
T.D.C.
ATMOSPHERIC LINE
Nº 1.
"A" OPEN PIPE 4'9" LONG.
AT 860 r.p.m.
B.D.C.
T.D.C.
ATMOSPHERIC
LB/IN.²
Nº 2 A & B
"B" DITTO. AT 1160 r.p.m.
"A", AS ABOVE
ATMOSPHERIC
Nº 2 A & C
"C" IS CURVE "B" PLOTTED ON SAME TIME BASE AS "A".

SHEET Nº 2.



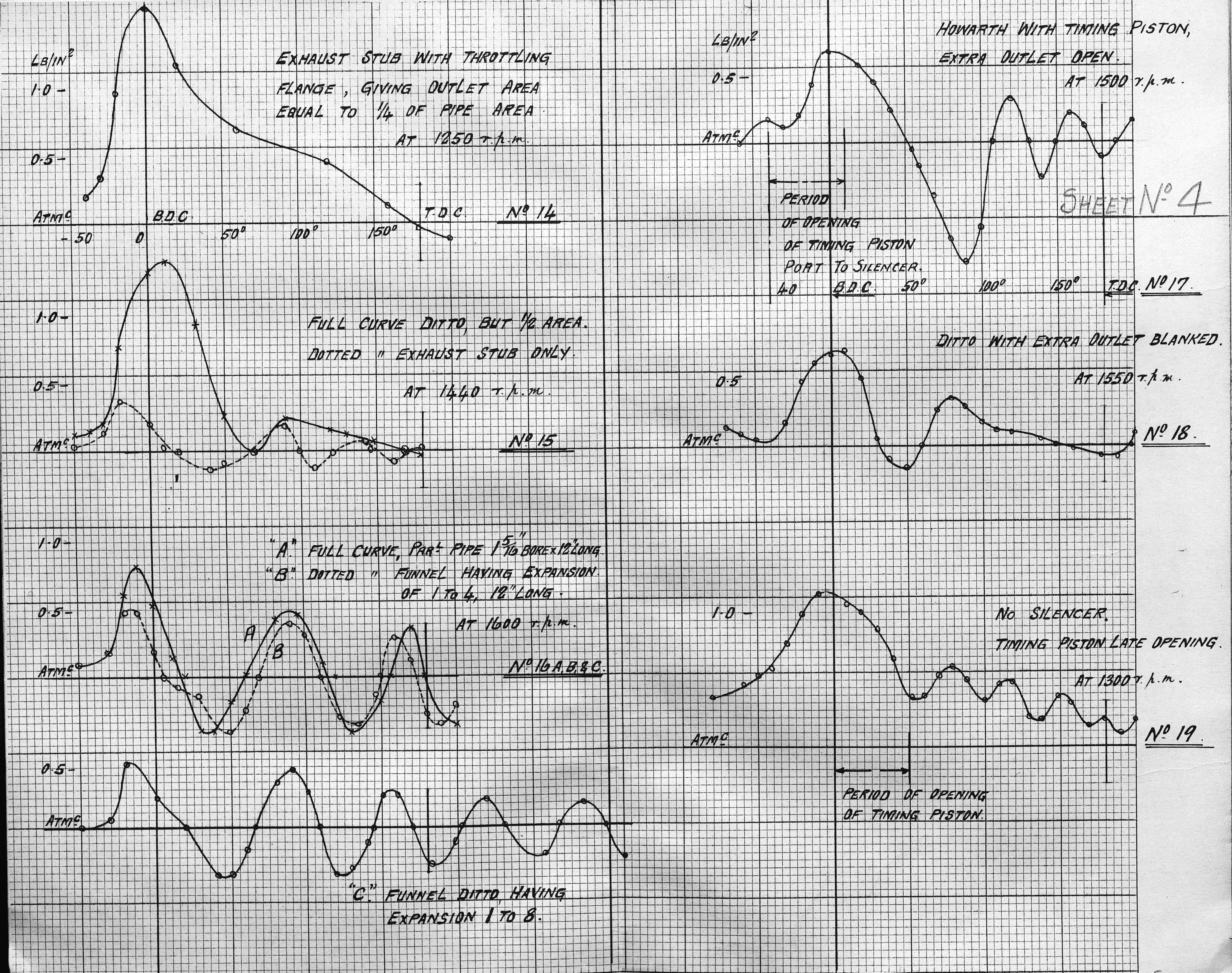
SHEET No 2
LB/IN²
1.0
0.5
OPEN PIPE 1¼" DIA. × 6 FT 9 INS.
AT 1660 r.p.m.
at Vs = 1300 ft/sec
Period = 210°
No 3.
ATMOSPHERIC LINE
-50° 0° 50° 100° 150° 200° 250° 300° 350° 400° 450° 550° 600° 630° 670°
B.D.C.
T.D.C.
CRANK ANGLES
Period = 146°
DITTO AT 1170 r.p.m.
No 4.
HOWARTH SILENCER ON ABOVE PIPE.
AT 1660 r.p.m.
No 5
A
-50° 0° 50° 100° 150° 200° 250° 300° 350° 400° 450° 500° 550° 600°
670°
CARBJECTOR SILENCER ON ABOVE PIPE.
AT 1500 r.p.m.
No 6.

SHEET Nº 3.



SHEET Nº 3.
LB/IN²
B.D.C.
CRANK ANGLES
T.D.C.
50°
100°
150°
200°
1
0·5
ATMᶜ 0
HOWARTH ON
EXHST STUB
AT 1380 r.p.m.
Nº 7
0·5
ATMᶜ
DITTO
AT 2200 r.p.m.
Nº 8
1
0·5
ATMᶜ
A
B
"A" DOTTED CURVE - HOWARTH AS ABOVE
"B" FULL " HOWARTH REVERSED
AT 1800 r.p.m.
Nº 9
1
0·5
ATMᶜ
"A" DOTTED CURVE - HOWARTH + CARBJECTOR IN SERIES ON A 6'-9" PIPE
"B" FULL CURVE - HOWARTH ALONE
AT 1700 r.p.m.
Nº 10
A
B
"A" DOTTED CURVE - OPEN PIPE 1⅝" DIA x 4'-9"
"B" FULL " DITTO WITH BAFFLE
AT 1000 r.p.m.
LB/IN²
1
0·5
0
ATMᶜ
A
B
Nº 11
B.D.C.
50°
100°
150°
T.D.C.
1
0·5
0
ATMᶜ
A
B
DITTO AT 1300 r.p.m.
Nº 12
+0
0·5
0
ATMᶜ
A
B
DITTO AT 1600 r.p.m.
Nº 13.

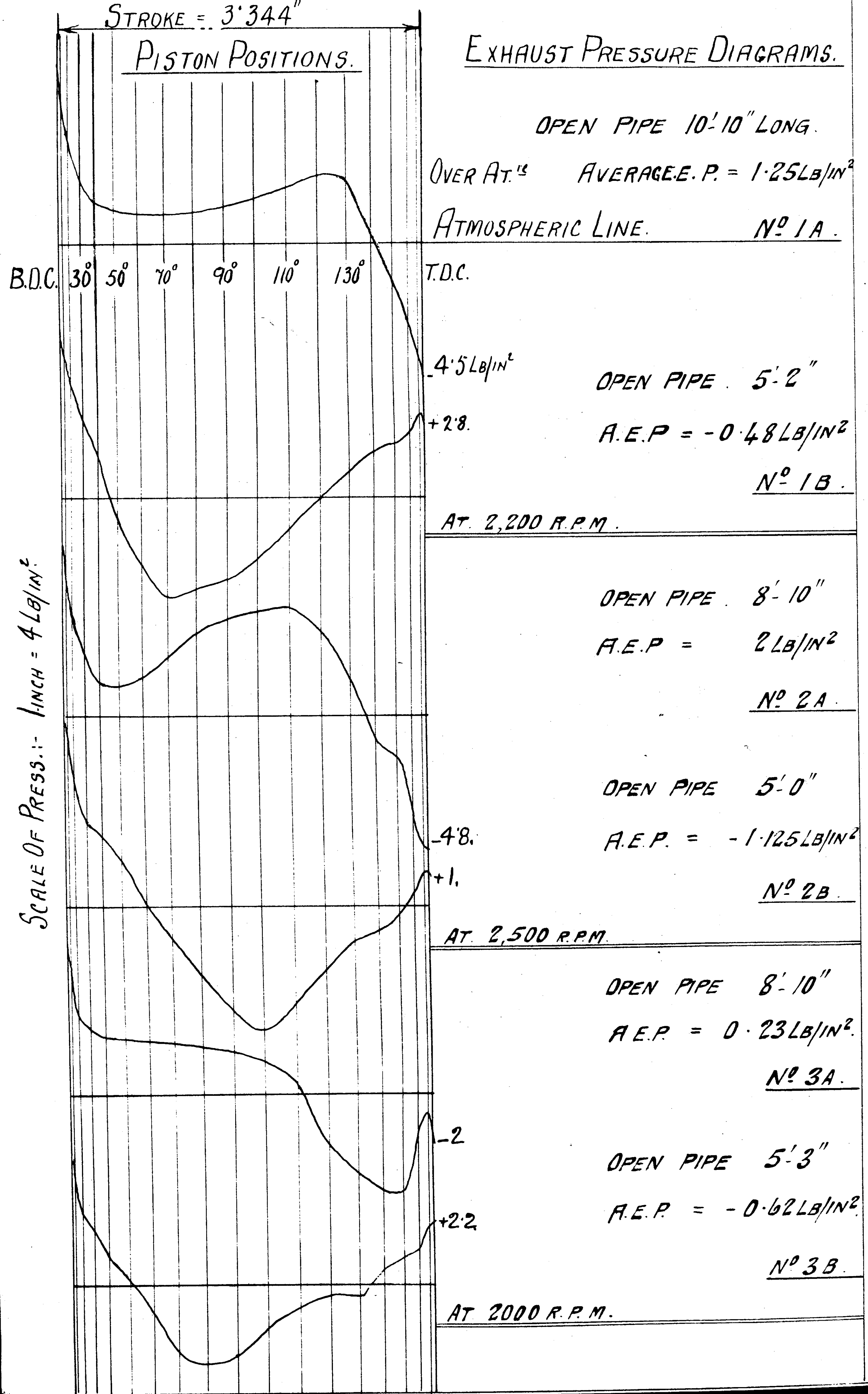
Sheet Nº 4.

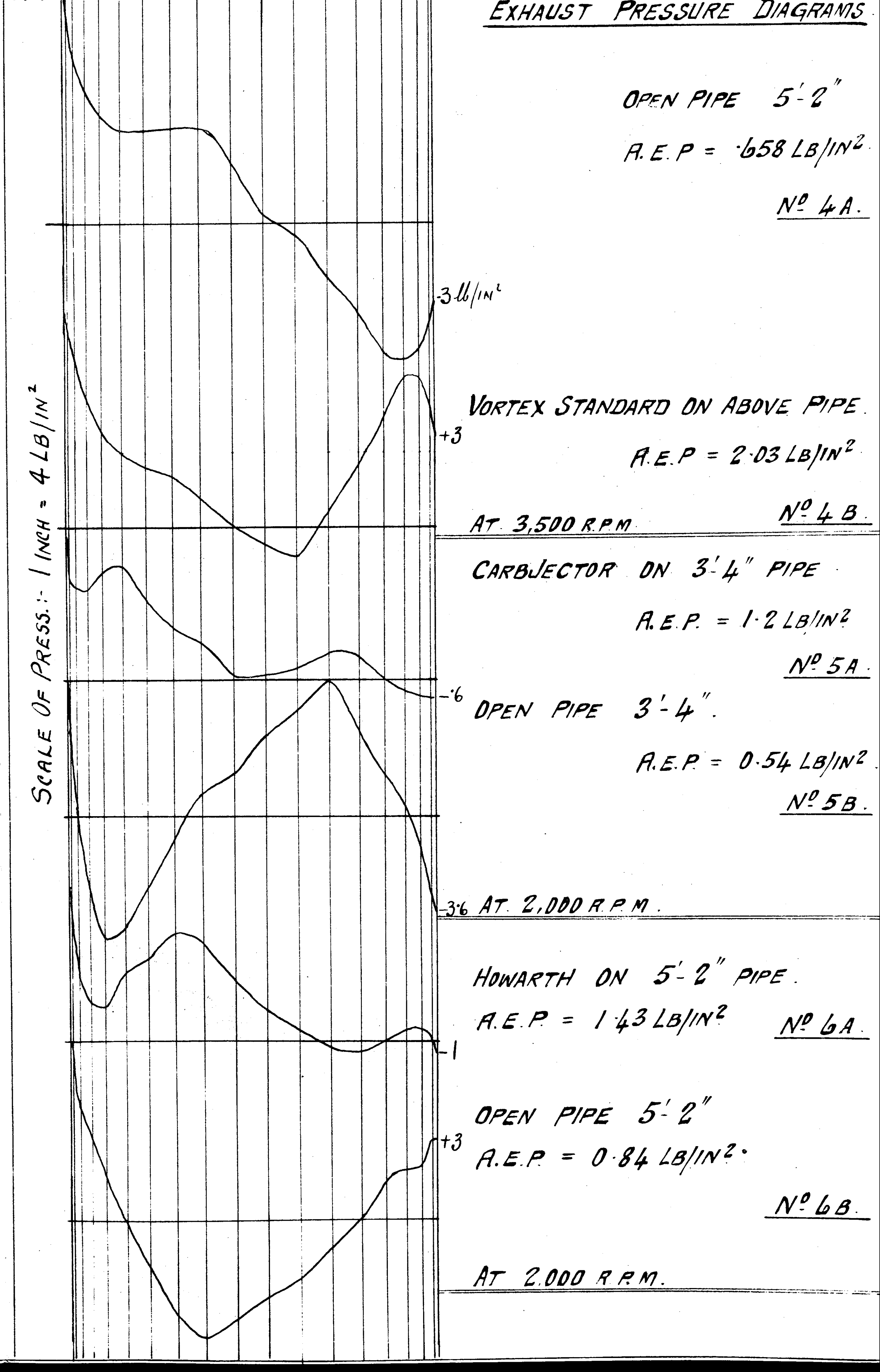


SHEET No 4
LB/IN²
1.0
0.5
ATMC
-50
0
B.D.C
50°
100°
150°
T.D.C.
EXHAUST STUB WITH THROTTLING FLANGE, GIVING OUTLET AREA EQUAL TO 1/4 OF PIPE AREA
AT 1250 r.p.m.
No 14
FULL CURVE DITTO, BUT 1/2 AREA.
DOTTED " EXHAUST STUB ONLY
AT 1440 r.p.m.
No 15
"A" FULL CURVE, PARL PIPE 1 5/16" BORE x 12" LONG.
"B" DOTTED " FUNNEL HAVING EXPANSION OF 1 TO 4, 12" LONG.
AT 1600 r.p.m.
A
B
No 16 A, B & C.
"C" FUNNEL DITTO, HAVING EXPANSION 1 TO 8.
HOWARTH WITH TIMING PISTON, EXTRA OUTLET OPEN.
AT 1500 r.p.m.
PERIOD OF OPENING OF TIMING PISTON PORT TO SILENCER.
40
No 17
DITTO WITH EXTRA OUTLET BLANKED.
AT 1550 r.p.m.
No 18
NO SILENCER.
TIMING PISTON LATE OPENING.
AT 1300 r.p.m.
PERIOD OF OPENING OF TIMING PISTON.
No 19

SHEET N: 5.

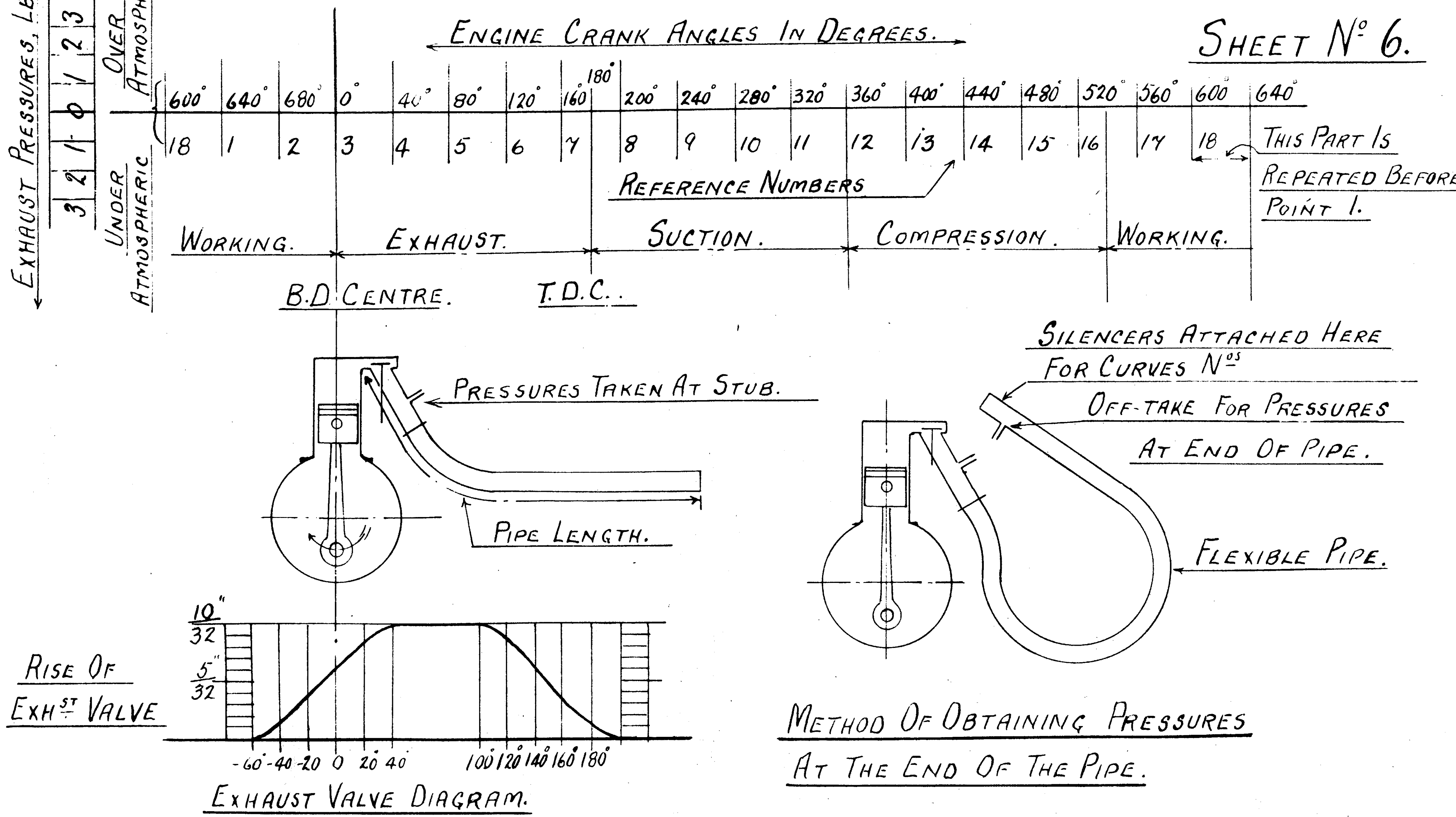
SHEET No. 5.


STROKE = 3·344"
PISTON POSITIONS.
EXHAUST PRESSURE DIAGRAMS.
OPEN PIPE 10'-10" LONG.
OVER ATᴸ AVERAGE E.P. = 1·25 LB/IN²
ATMOSPHERIC LINE. No. 1A.
B.D.C. 30° 50° 70° 90° 110° 130° T.D.C.
-4·5 LB/IN²
+2·8.
OPEN PIPE. 5'-2"
A.E.P = -0·48 LB/IN²
No. 1B.
AT 2,200 R.P.M.
OPEN PIPE. 8'-10"
A.E.P = 2 LB/IN²
No. 2A.
OPEN PIPE 5'-0"
-4·8.
+1.
A.E.P. = -1·125 LB/IN²
No. 2B.
AT 2,500 R.P.M.
OPEN PIPE 8'-10"
A.E.P = 0·23 LB/IN².
No. 3A.
-2
OPEN PIPE 5'-3"
+2·2.
A.E.P = -0·62 LB/IN².
No. 3B.
AT 2000 R.P.M.
SCALE OF PRESS.:- 1 INCH = 4 LB/IN².


B.D.C. 30° 50° 70° 90° 110° 130° 150° T.D.C.
EXHAUST PRESSURE DIAGRAMS.
OPEN PIPE 5'-2"
A.E.P = ·658 LB/IN²
No. 4A.
-3 LB/IN²
+3
VORTEX STANDARD ON ABOVE PIPE.
A.E.P = 2·03 LB/IN²
AT 3,500 R.P.M.
No. 4B.
CARBJECTOR ON 3'-4" PIPE
A.E.P. = 1·2 LB/IN²
No. 5A.
-·6
OPEN PIPE 3'-4".
A.E.P = 0·54 LB/IN²
No. 5B.
-3·6
AT 2,000 R.P.M.
HOWARTH ON 5'-2" PIPE.
A.E.P = 1·43 LB/IN²
No. 6A.
-1
OPEN PIPE 5'-2"
+3
A.E.P = 0·84 LB/IN².
No. 6B.
AT 2,000 R.P.M.
SCALE OF PRESS.:- 1 INCH = 4 LB/IN²

SHEET N^{o} 6.

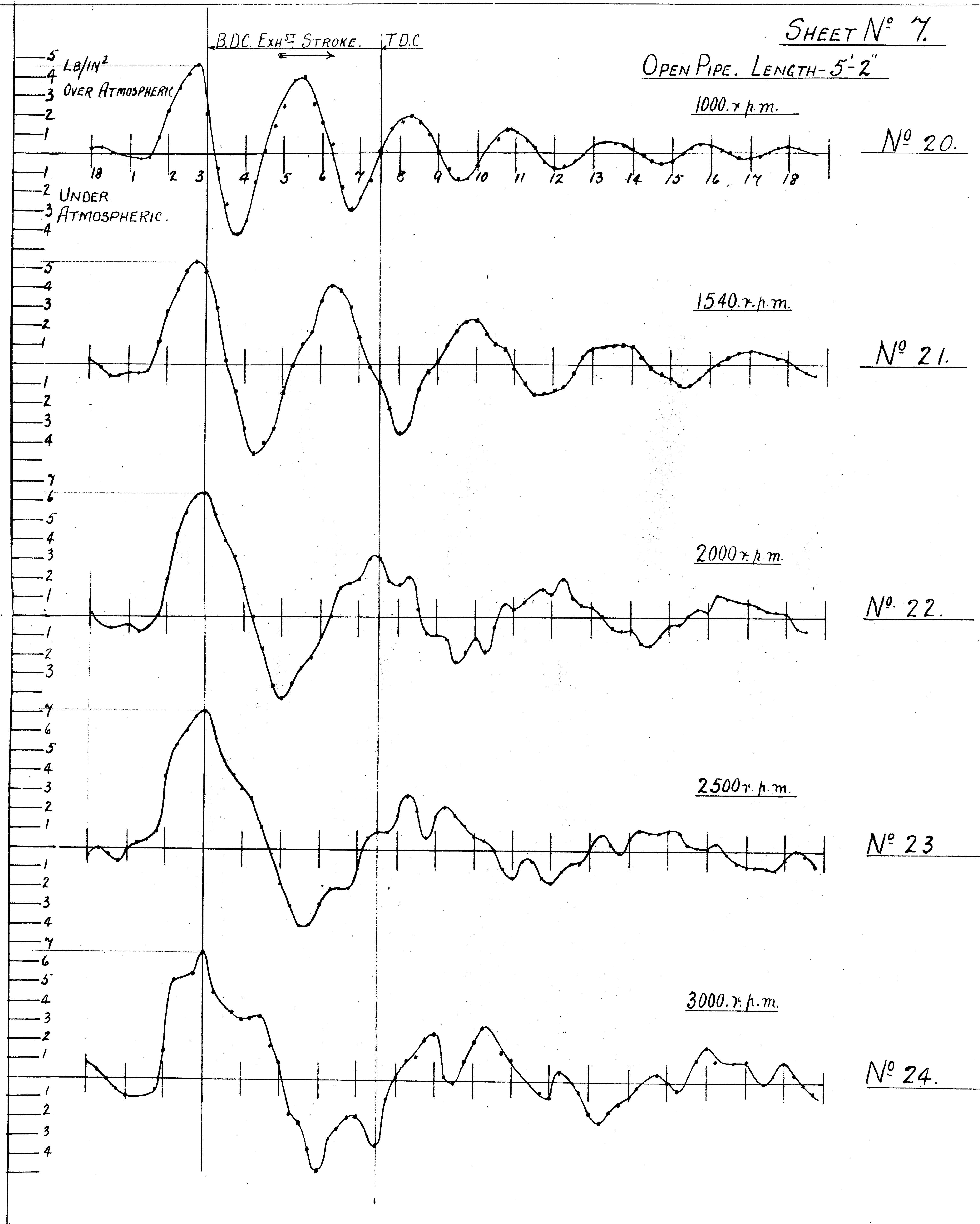
KEY TO SHEETS OF CURVES TAKEN WITH THE MULTI-POINT INDICATOR.



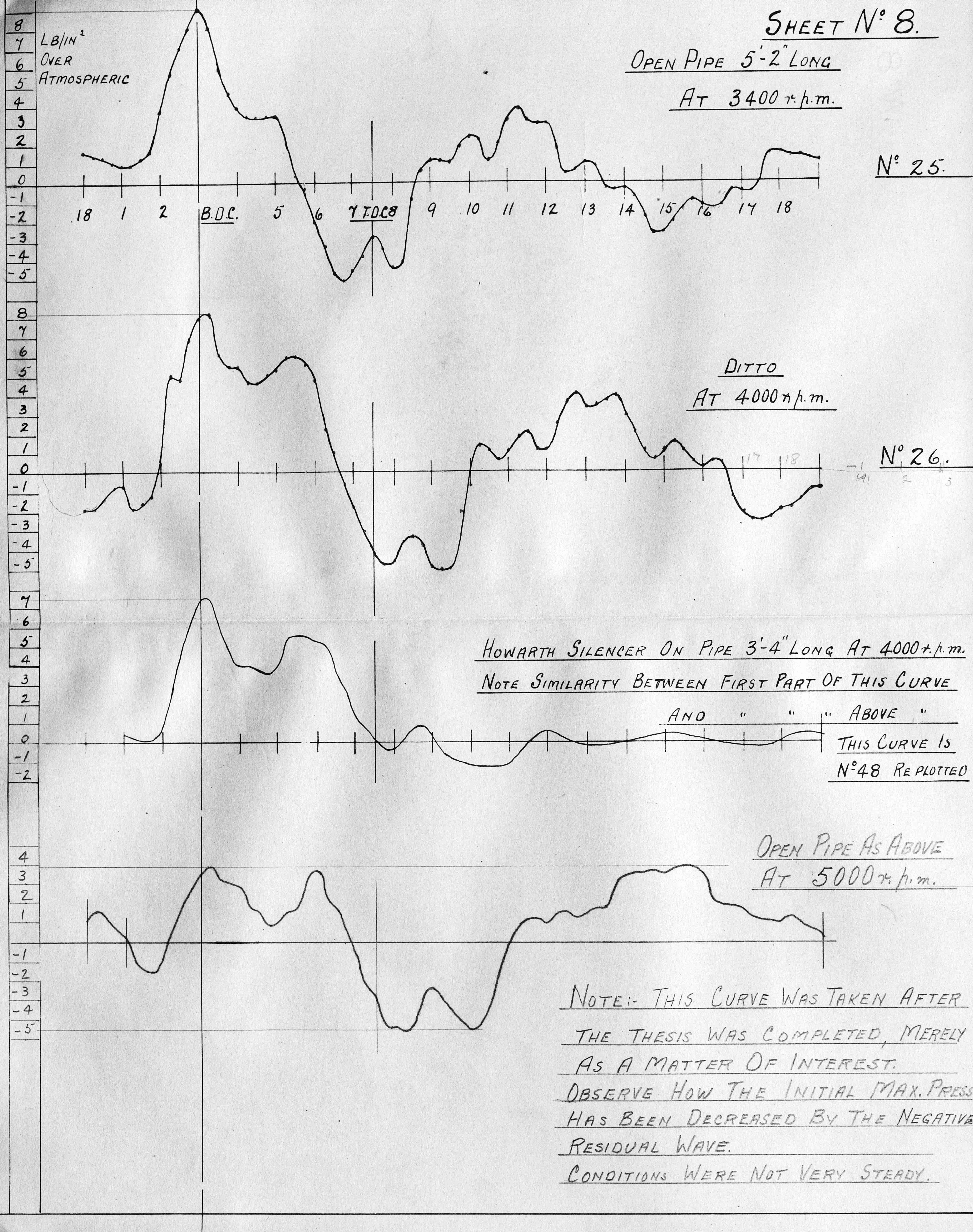
SHEET No. 7.

SHEET Nº 7.

OPEN PIPE. LENGTH-5'-2"

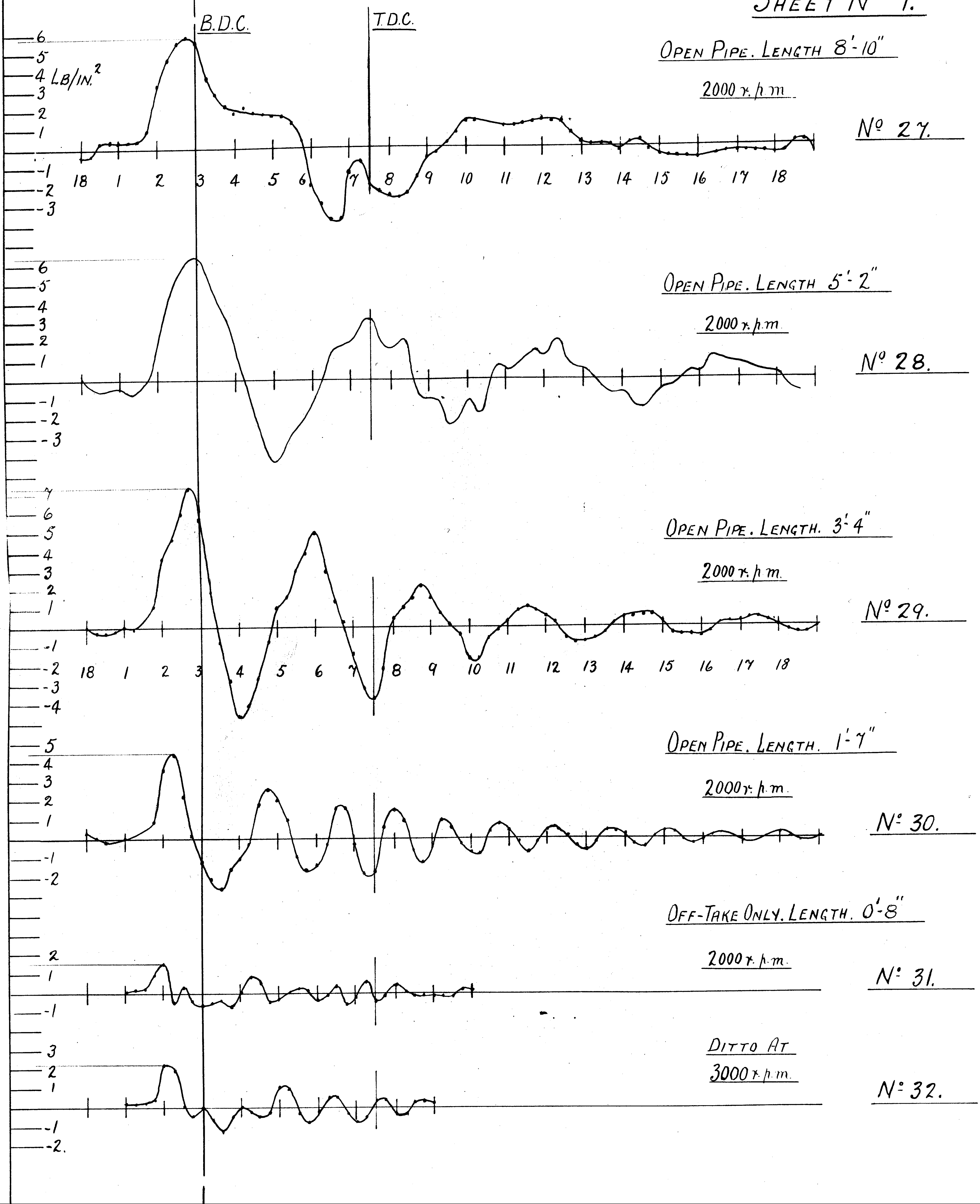


SHEET Nº 8.


Sheet No. 8.
Open Pipe 5'-2" Long
At 3400 r.p.m.
No. 25.
LB/IN²
Over
Atmospheric
18 1 2 B.D.C. 5 6 7 T.D.C 8 9 10 11 12 13 14 15 16 17 18
Ditto
At 4000 r.p.m.
No. 26.
Howarth Silencer On Pipe 3'-4" Long At 4000 r.p.m.
Note Similarity Between First Part Of This Curve
And " " " Above "
This Curve Is
No. 48 Replotted
Open Pipe As Above
At 5000 r.p.m.
Note:- This Curve Was Taken After
The Thesis Was Completed, Merely
As A Matter Of Interest.
Observe How The Initial Max. Press.
Has Been Decreased By The Negative
Residual Wave.
Conditions Were Not Very Steady.

SHEET Nº 9.

Sheet Nº 9.



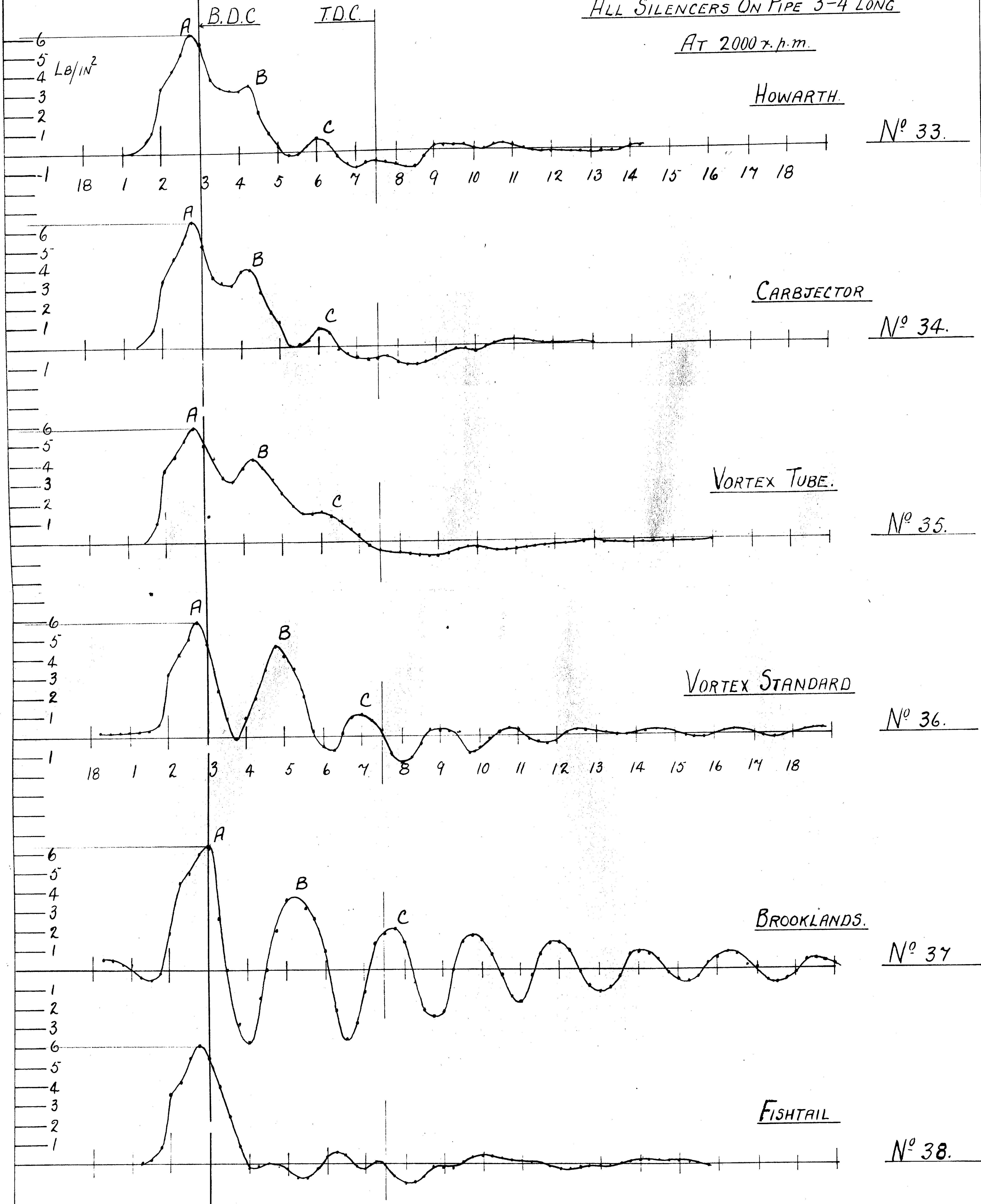
B.D.C.
T.D.C.
Lb/in.²
Open Pipe. Length 8'-10"
2000 r.p.m.
Nº 27.
Open Pipe. Length 5'-2"
2000 r.p.m.
Nº 28.
Open Pipe. Length. 3'-4"
2000 r.p.m.
Nº 29.
Open Pipe. Length. 1'-7"
2000 r.p.m.
Nº 30.
Off-Take Only. Length. 0'-8"
2000 r.p.m.
Nº 31.
Ditto At
3000 r.p.m.
Nº 32.

SHEET Nº 11.

SHEET Nº 10.

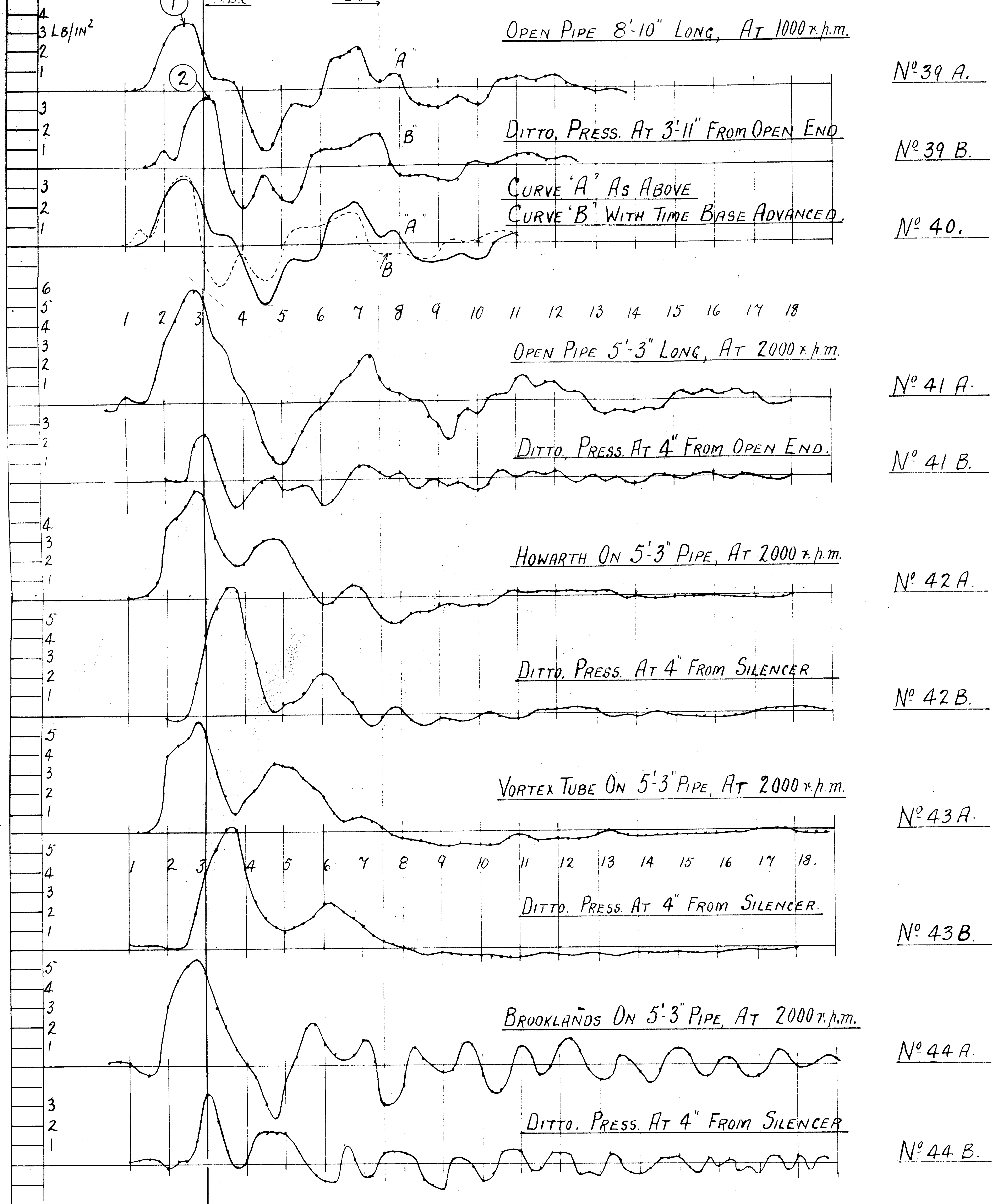
ALL SILENCERS ON PIPE 3'-4" LONG

AT 2000 r.p.m.



SHEET Nº 11.

SHEET Nº 11.

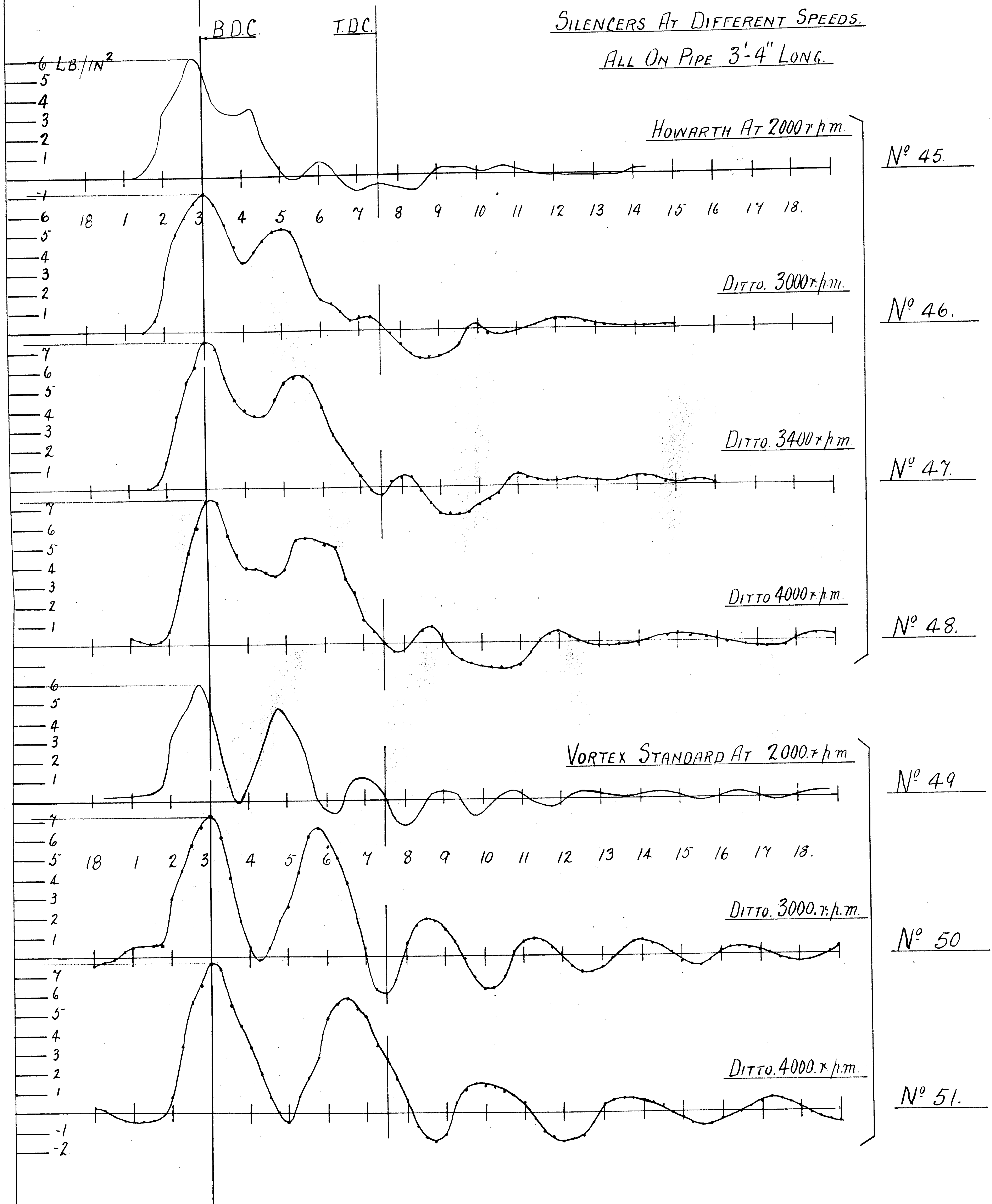

B.D.C
T.D.C
3 LB/IN²
OPEN PIPE 8'-10" LONG, AT 1000 r.p.m.
Nº 39 A.
DITTO, PRESS. AT 3'-11" FROM OPEN END
Nº 39 B.
CURVE 'A' AS ABOVE
CURVE 'B' WITH TIME BASE ADVANCED.
Nº 40.
OPEN PIPE 5'-3" LONG, AT 2000 r.p.m.
Nº 41 A.
DITTO, PRESS. AT 4" FROM OPEN END.
Nº 41 B.
HOWARTH ON 5'-3" PIPE, AT 2000 r.p.m.
Nº 42 A.
DITTO. PRESS. AT 4" FROM SILENCER
Nº 42 B.
VORTEX TUBE ON 5'-3" PIPE, AT 2000 r.p.m.
Nº 43 A.
DITTO. PRESS. AT 4" FROM SILENCER.
Nº 43 B.
BROOKLANDS ON 5'-3" PIPE, AT 2000 r.p.m.
Nº 44 A.
DITTO. PRESS. AT 4" FROM SILENCER
Nº 44 B.

SHEET Nº 12.

SHEET No. 12.

SILENCERS AT DIFFERENT SPEEDS.

ALL ON PIPE 3'-4" LONG.

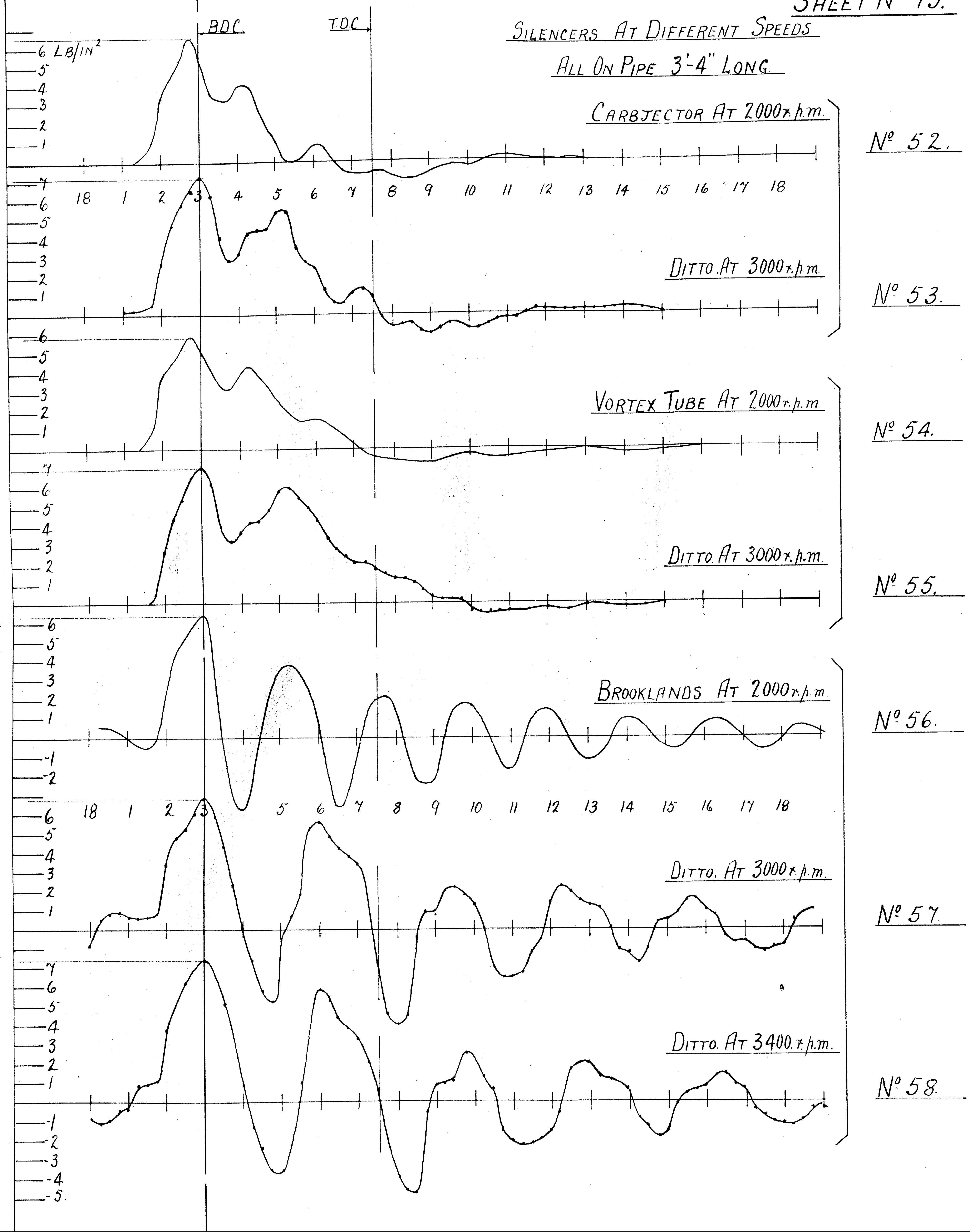


SHEET No 13.

SHEET Nº 13.

SILENCERS AT DIFFERENT SPEEDS

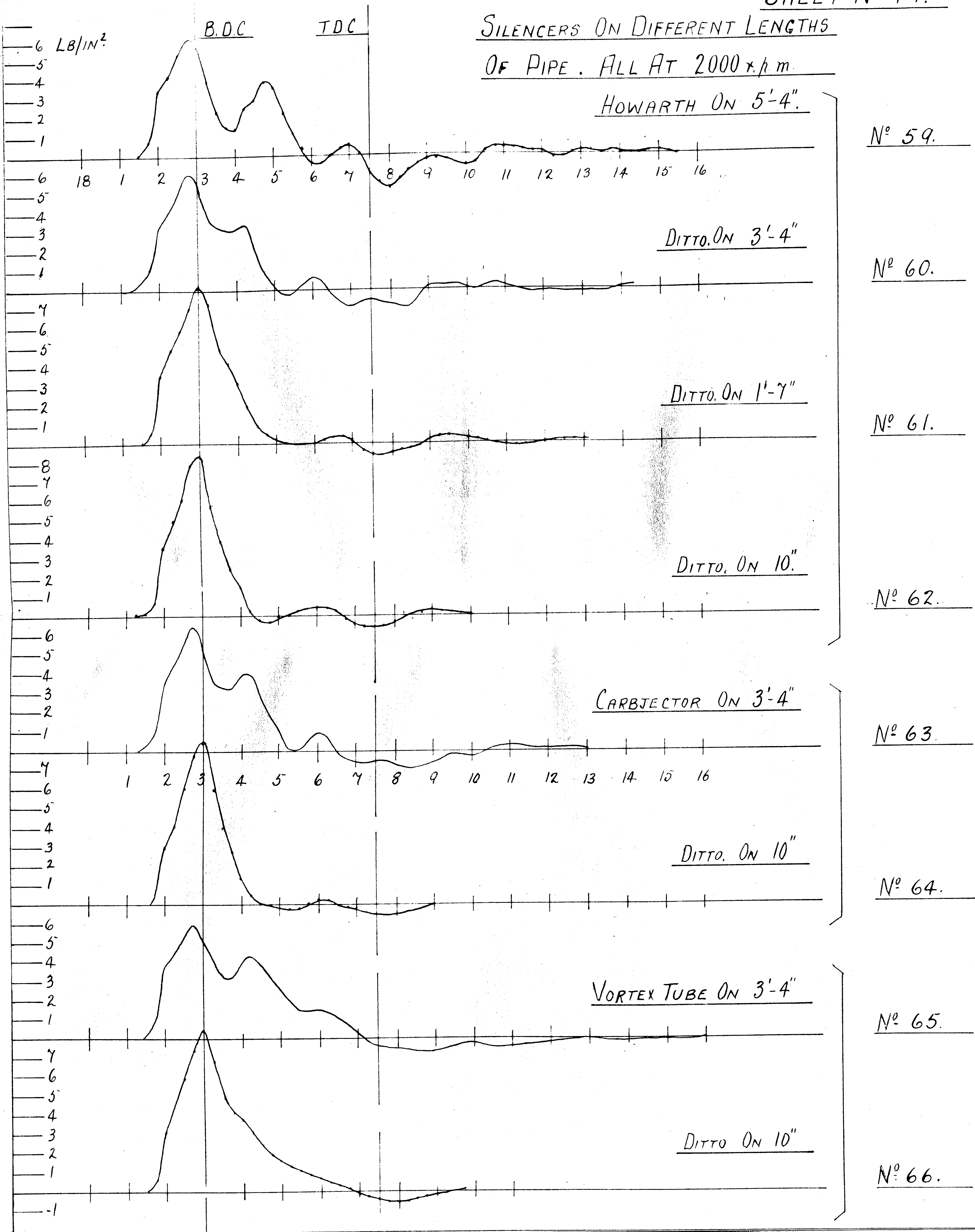
ALL ON PIPE 3'-4" LONG.



SHEET Nº 14.

SHEET No. 14.

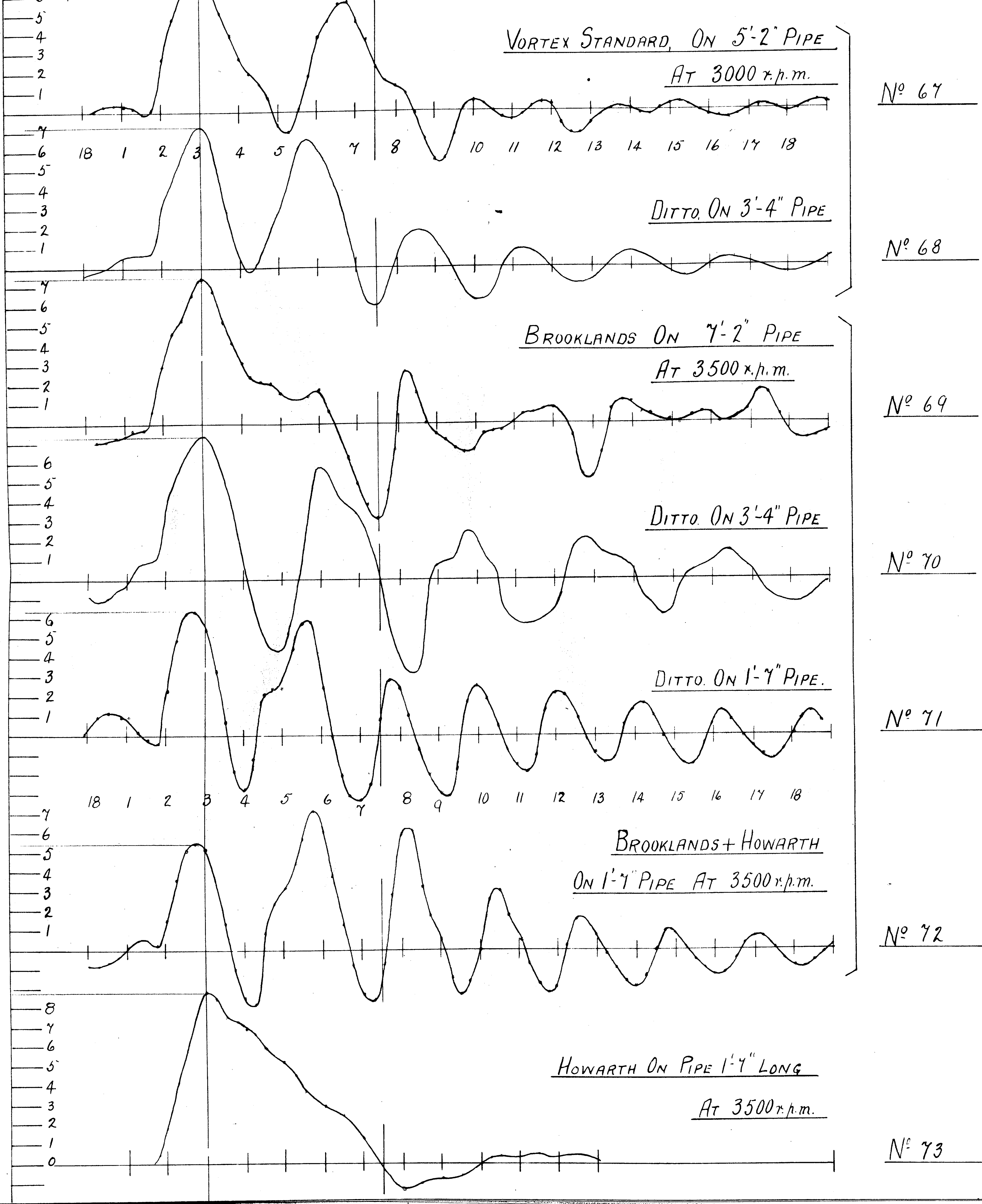
SILENCERS ON DIFFERENT LENGTHS OF PIPE. ALL AT 2000 r.p.m.



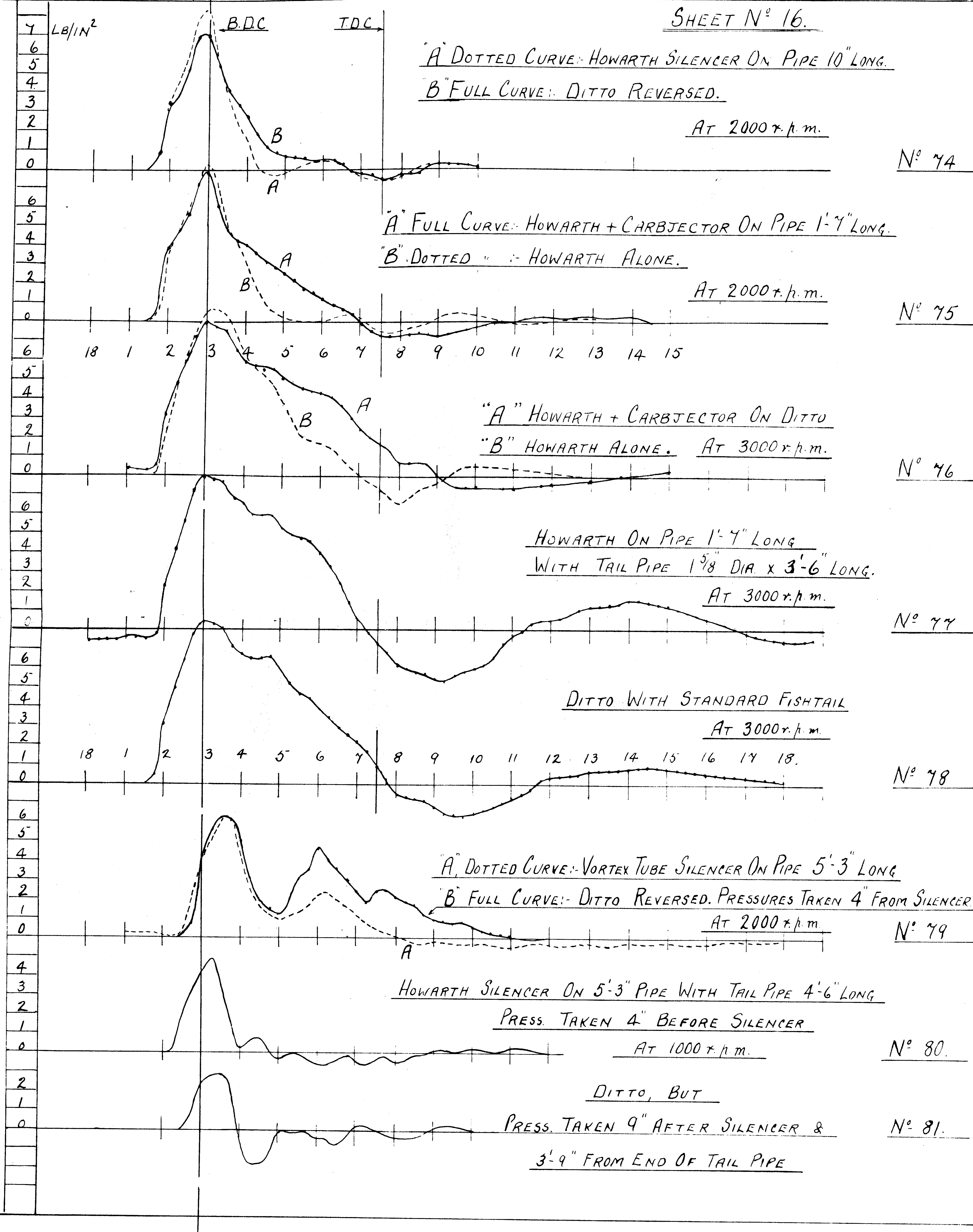
SHEET Nº 15.

Sheet No. 15.

Silencers On Different Lengths Of Pipe



SHEET Nº 16.



SHEET Nº 16.
LB/IN²
B.D.C
T.D.C
"A" DOTTED CURVE:- HOWARTH SILENCER ON PIPE 10" LONG.
"B" FULL CURVE:- DITTO REVERSED.
AT 2000 r.p.m.
Nº 74
"A" FULL CURVE:- HOWARTH + CARBJECTOR ON PIPE 1'-7" LONG.
"B" DOTTED " :- HOWARTH ALONE.
AT 2000 r.p.m.
Nº 75
"A" HOWARTH + CARBJECTOR ON DITTO
"B" HOWARTH ALONE.
AT 3000 r.p.m.
Nº 76
HOWARTH ON PIPE 1'-7" LONG
WITH TAIL PIPE 1 5/8" DIA x 3'-6" LONG.
AT 3000 r.p.m.
Nº 77
DITTO WITH STANDARD FISHTAIL
AT 3000 r.p.m.
Nº 78
"A" DOTTED CURVE:- VORTEX TUBE SILENCER ON PIPE 5'-3" LONG
"B" FULL CURVE:- DITTO REVERSED. PRESSURES TAKEN 4" FROM SILENCER
AT 2000 r.p.m.
Nº 79
HOWARTH SILENCER ON 5'-3" PIPE WITH TAIL PIPE 4'-6" LONG
PRESS. TAKEN 4" BEFORE SILENCER
AT 1000 r.p.m.
Nº 80.
DITTO, BUT
PRESS. TAKEN 9" AFTER SILENCER &
3'-9" FROM END OF TAIL PIPE
Nº 81.