Experimental Procedure

Four sets of test were made, each with different exhaust system. The first one was made on the original system supplied with the engine.

A 1" coupling was fitted to the exhaust opening be mean of a set of screws and a tee was joined with the coupling via a union. The pressure transducer and its watercooled adaptor were fitted on the tee and the expansion chamber which was supplied with the engine attached to the tee with the help of a special union. The drawing of the assembly is shown in Fig.14 along with dimensions.

The second and the third arrangement were almost the same, the difference came from the unequal length of the pipe screwed into the tee. The length of the long exhaust pipe was 10 feet while of the short one was 49 inches. The expansion chamber was attached at the end of each pipe and the exhaust gas conveyed out of the laboratory by means of a blower. The drawing and the dimensions of the rigs are show in Fig. 14b and the photograph of the short exhaust pipe arrangement is shown in Fig. 15.

The last test was made with the pipes and the expansion chamber left out, exhaust gases went directly

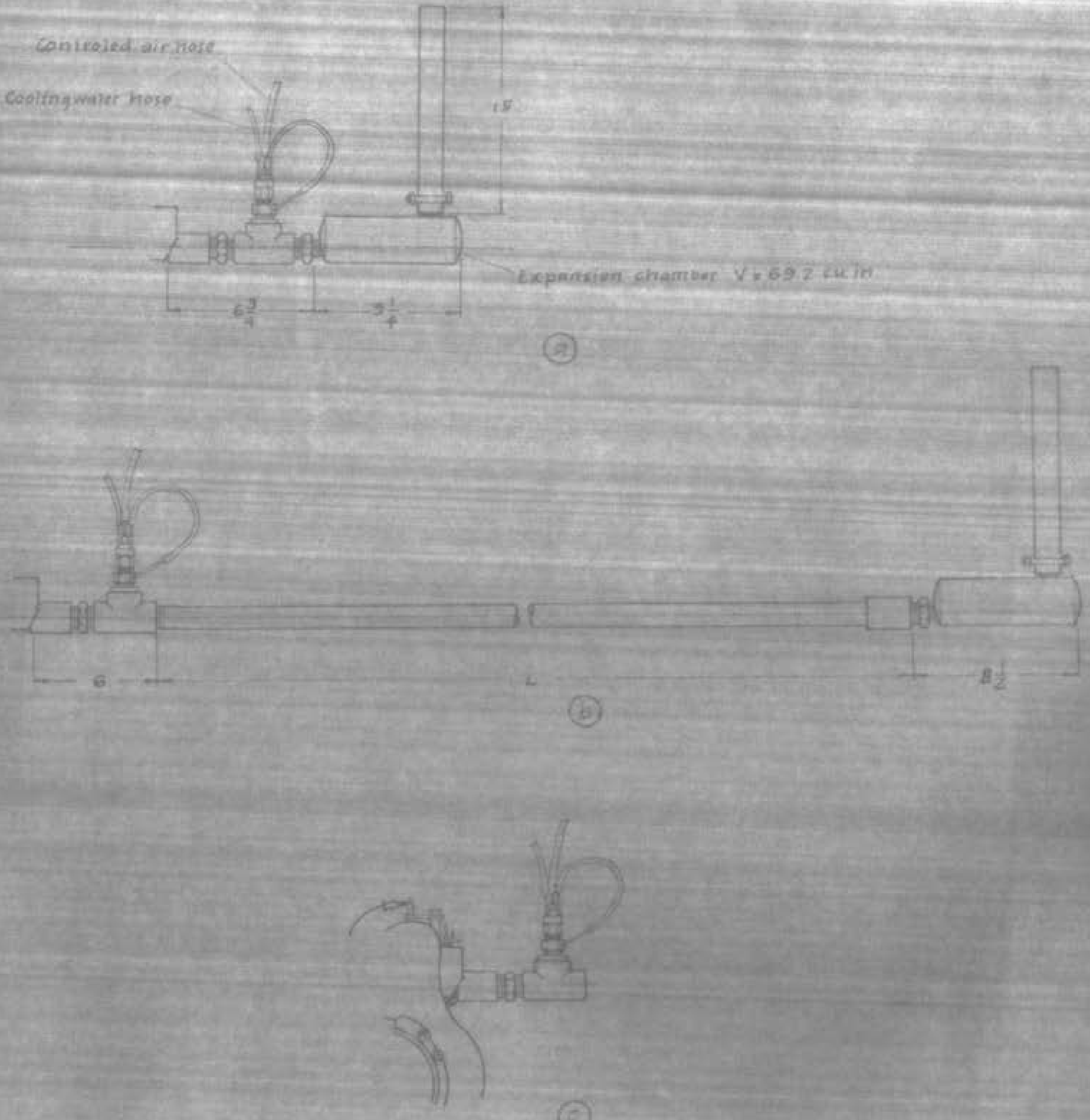


Fig. 14 Exhaust System Arrangements.

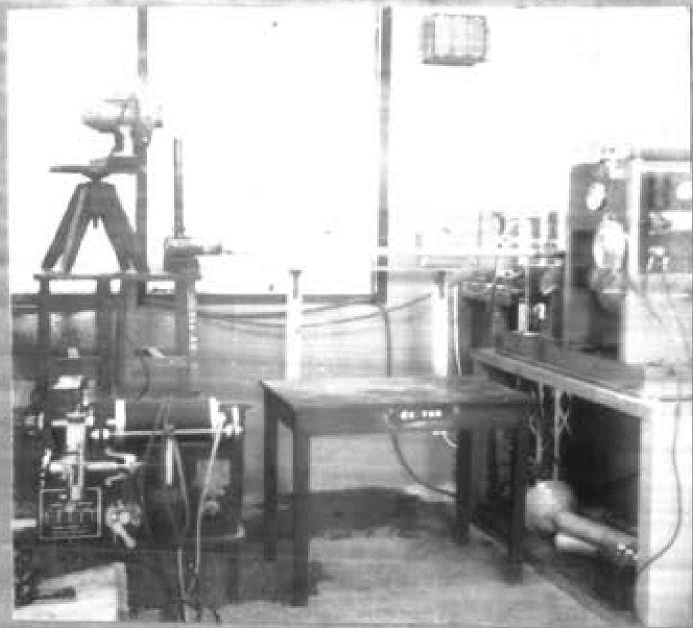


Fig. 15
Short Exhaust Pipe Arrangement

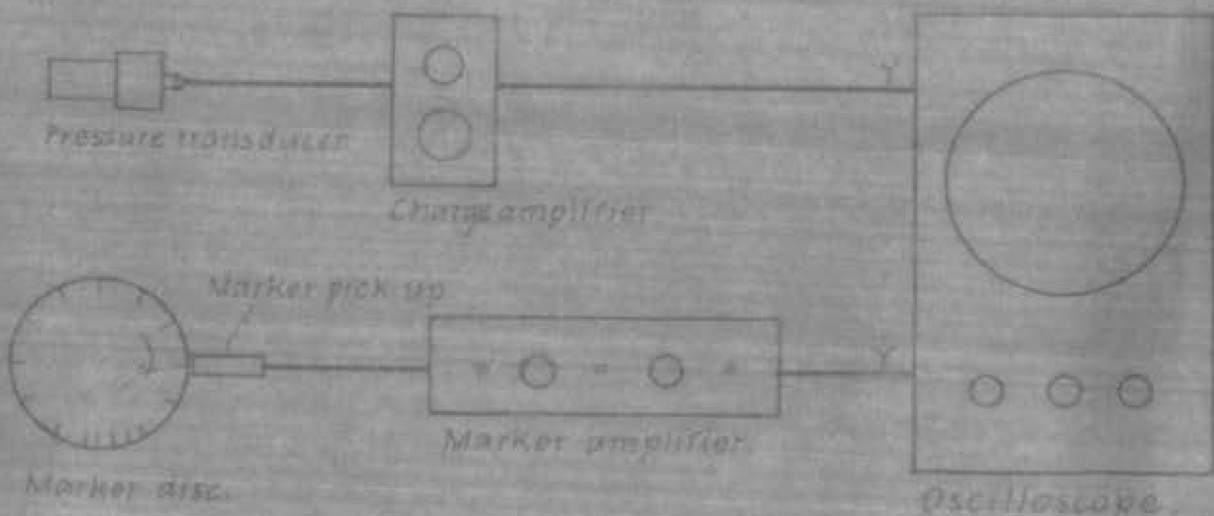


Fig 16 Schematic Diagram of Electronic Equipment.

through the tee into atmosphere. The drawing of the setting is shown in Fig. 14 c.

At each of the arrangement three pressure traces were photographed,they are:cylinder, exhaust and crankcase pressure traces. On every arrangement the tests were made at three speed. The lowest selected speed was 2000 rpm, the other two were 2500 rpm and 3000 rpm at the upper limit. Electronic equipment were connected as shown in schematic diagram of Fig. 16.

Ideal Scavenging Analysis

To enable the evaluation of the effectiveness of each exhaust arrangement, an ideal scavenging process is established. Since the accurate analysis of the process is a complex problem, simplifying assumptions are made at this stage. This implies discharge of exhaust gas directly into atmosphere while the engine is running at a very low speed and no chemical reaction occurs during the process. In this way the effect of a pressure pulse which is reflected at the open end of the exhaust pipe and returns to the ports as a rarefaction wave as it happens in the conventional arrangement is avoided. Fig 18 shows the variations of the pressures in the exhaust port and the crankcase of the engine operating on an ideal process.

Beginning at TDC and follows the air standard cycle is the expansion of the products of combustion. It is assumed to be adiabatic and reversible. The expansion continues until the pressure reaches the atmospheric pressure just before the exhaust port is uncovered by the moving of the piston downwards. Before this the exhaust port is in connection with the external air only, thus the pressure in the port is at atmospheric and remains to be atmospheric after the exhaust port is put into connection with atmosphere. As the piston moves further and uncovers

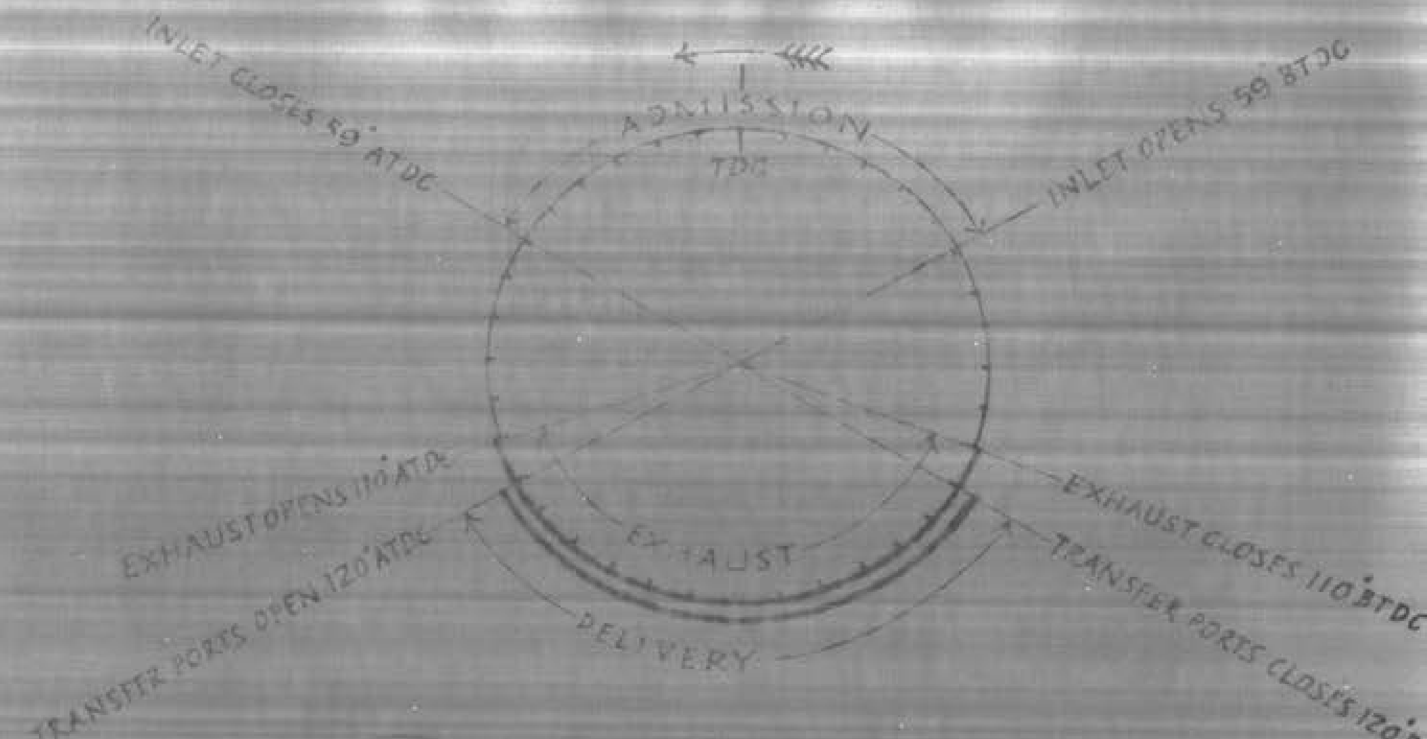


Fig. 17

Engine Timing Diagram

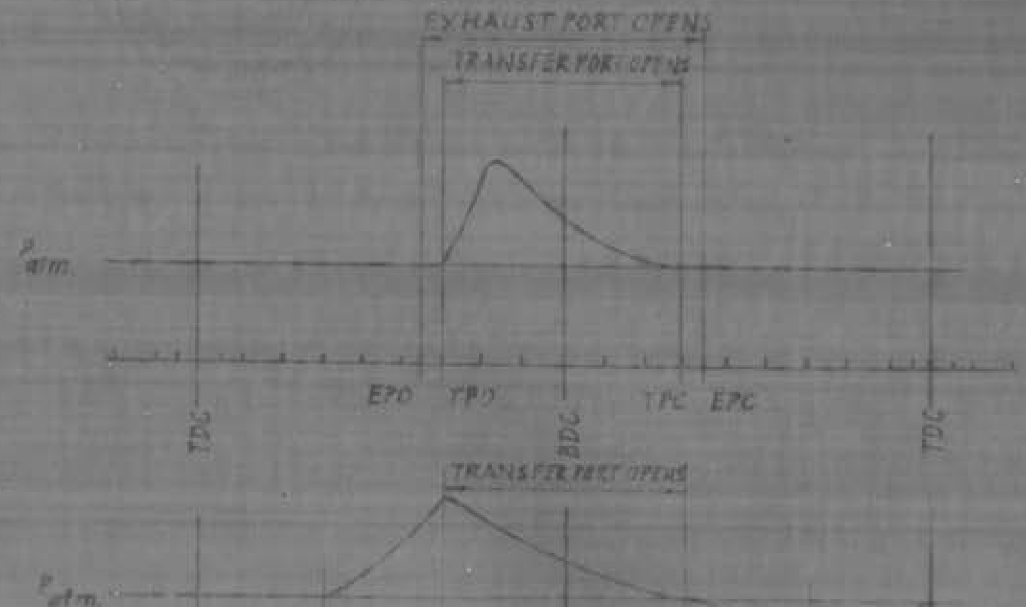


Fig. 18

Ideal Scavenging Diagram

the transfer ports the exhaust pressure begins to rise due to the rushing of fresh charge into the cylinder. At this moment the exhaust port, the cylinder and the crankcase are put in communication, hence, after the pressure in the exhaust port and the crankcase are in equilibrium, they begin to fall down to atmospheric again. The decrease of the pressures going on while the piston commences to move upwards from BDC. This two pressures reach atmospheric just before the transfer port closes. After this the persure in the exhaust port remains at atmospheric until the transfer port opens again. The opening and closing of the exhaust port do not effect the pressure variation in the exhaust port at all because the pressure inside the port is atmospheric at both instances.

The lower diagram shows the corresponding crankcase pressure variation. The piston covers the intake port after it moves for some times from TDC. The mixture induced in the crankcase is then compressed as the piston continues to move downwards. From this moment on the pressure in the crankcase will be increasing until the transfer port is uncovered. The compressed charge then flows into the cylinder. The crankcase pressure will be decreasing to atmosheric while the piston is moving up from BDC. At this period the exhaust and crankcase pressure are the same. After the transfer port is closed and the piston moving upwards, pressure in the crankcase will drop below atmospheric. The pressure stops decreasing the moment the inlet port opens and then begins to rise. As soon as the piston reaches TDC, the crankcase pressure will be atmospheric and remains there until the inlet port is closed again, hence a cycle is completed.

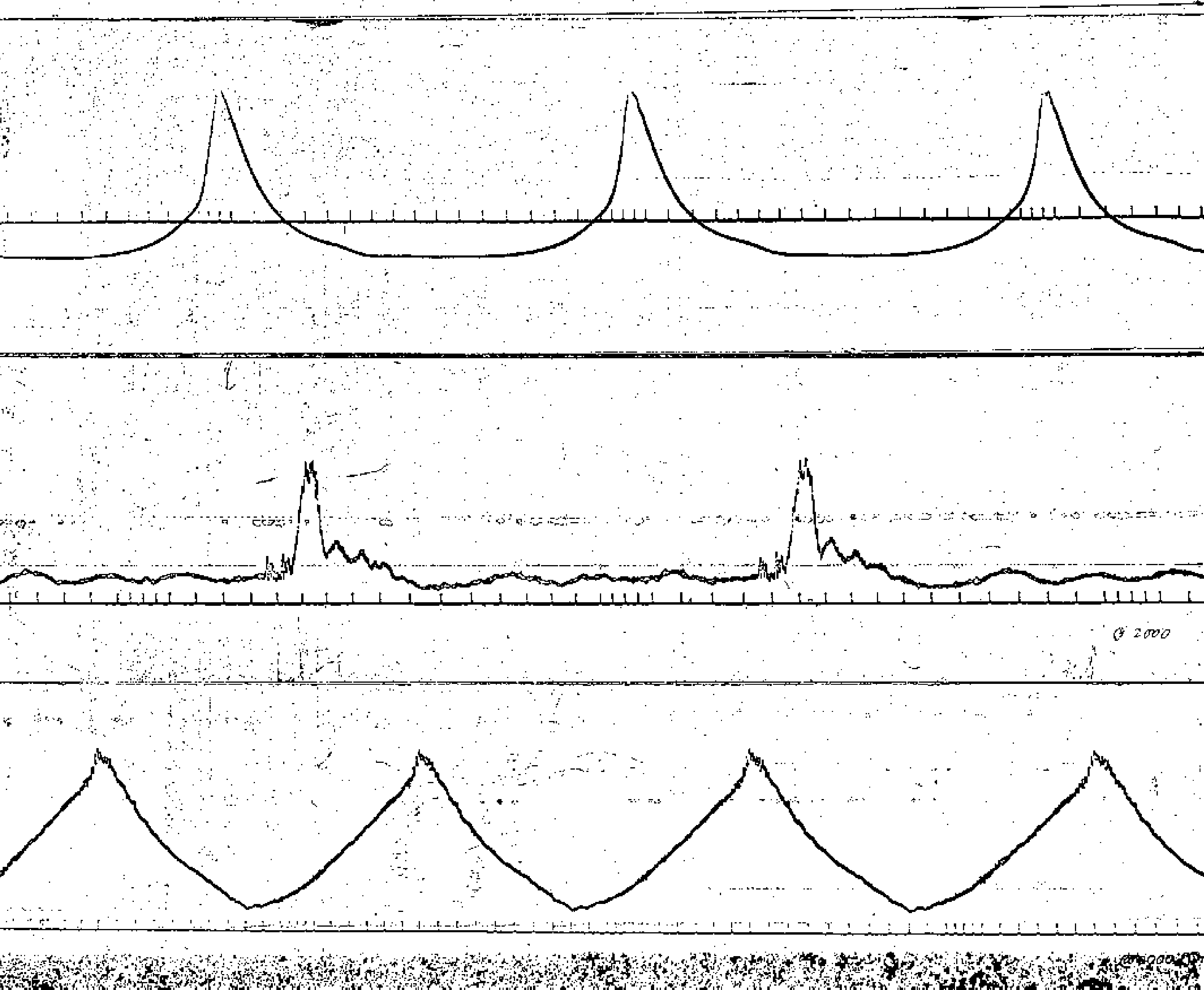
What deviate the actual scavenging process from the ideal one will be the effects of the high speed and the reflected waves from the open end of the exhaust pipe as well as the combustion of the unburned gas in the exhaust system.

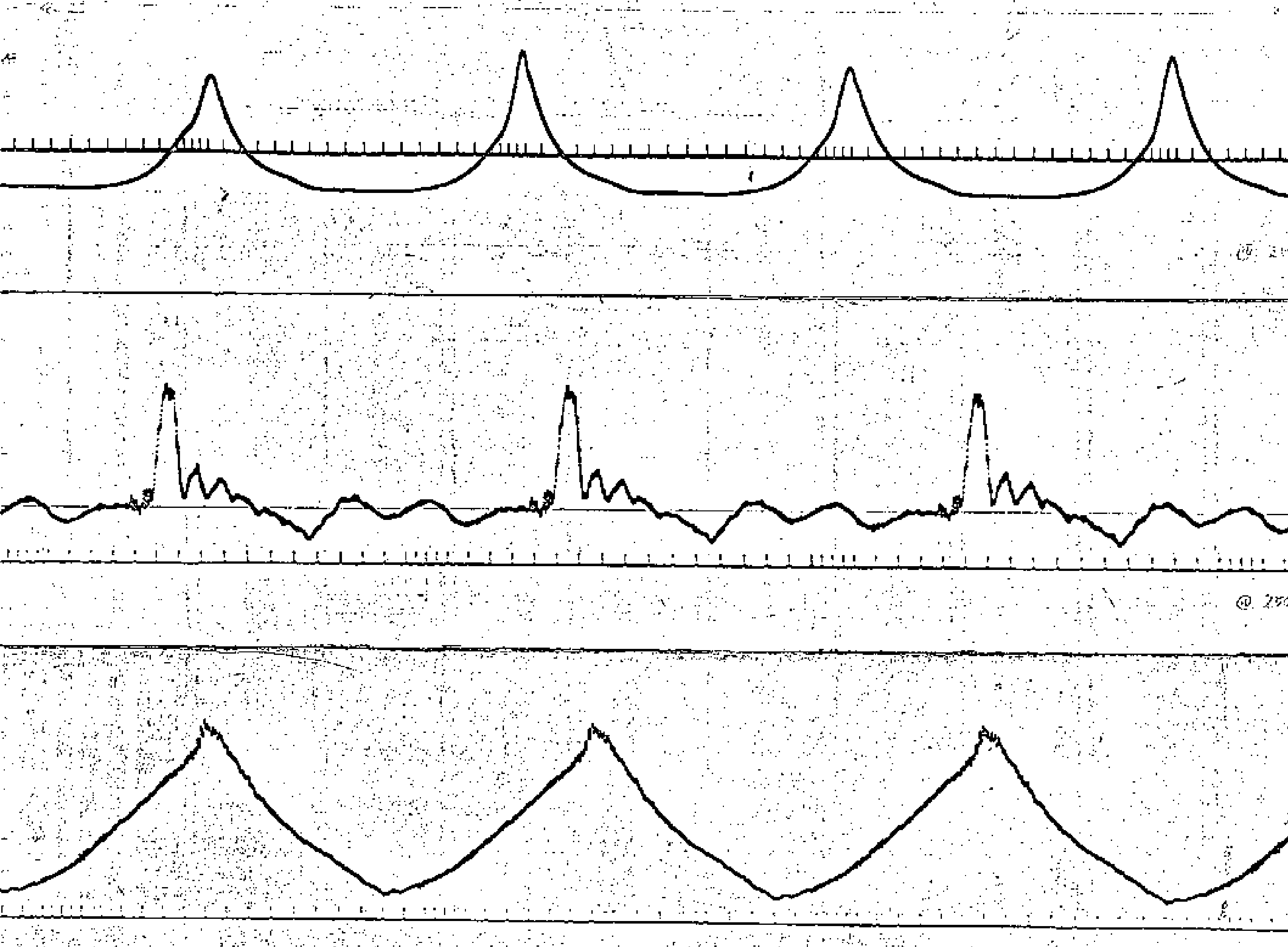
Photographic Traces of Variations of Pressure in the cylinder, Exhaust systems and Crankcase of a Two - stroke Crankcase Scavenge Engine at Different Speeds.

Four sets of photograph were taken which corresponded to four arrangements of exhaust system. They are shown in Fig 19, 20, 21, 22, The first represents pressure traces of the engine with the original setting. The second represents pressure traces when the long pipe was fitted in the system. The third is for the short pipe arrangement and the fourth is for the free exhaust arrangement.

Each set of photograph consists of nine traces. The first three traces were taken while the engine running

ORIGINAL SETTING




@ 2500 rpm

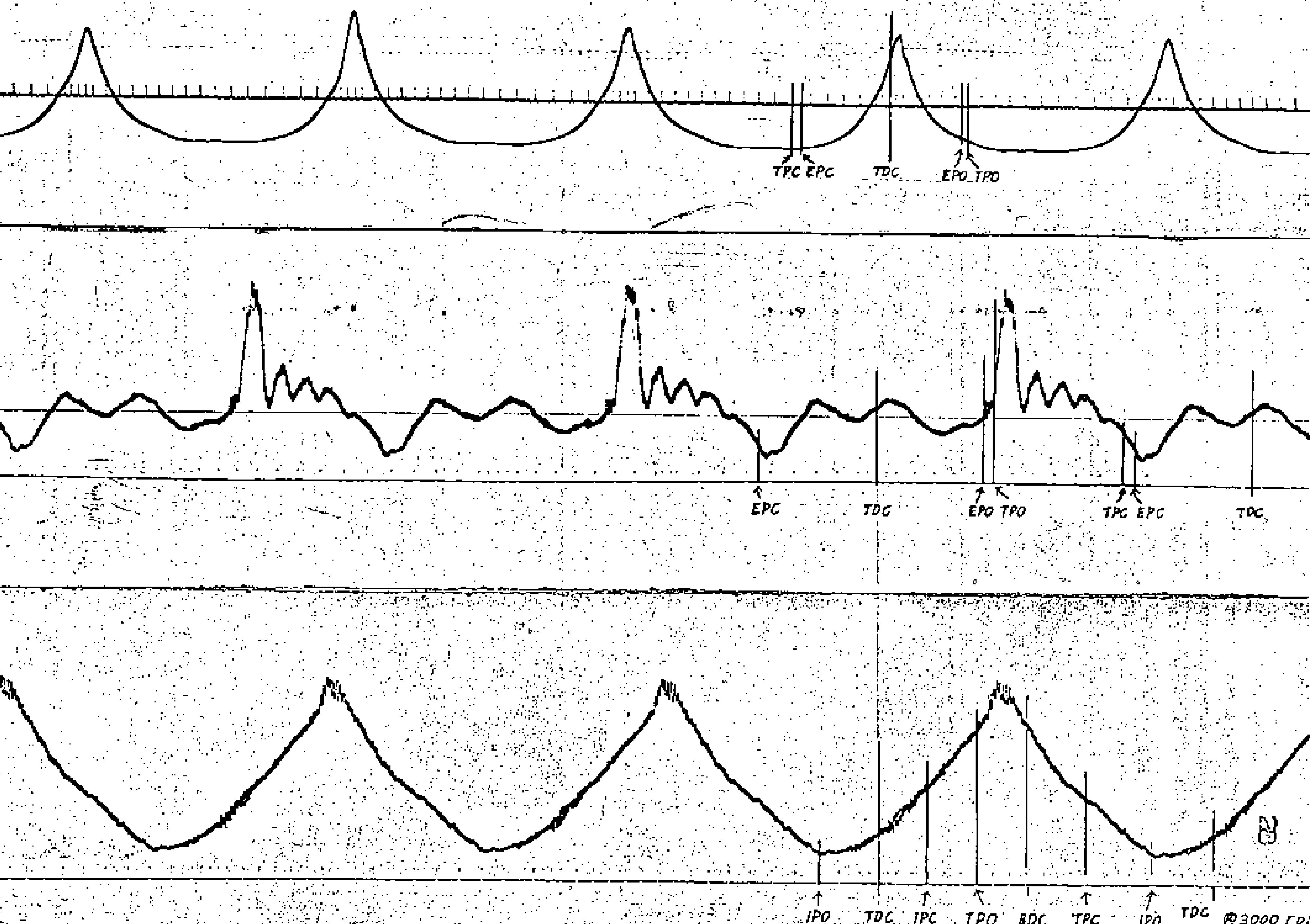
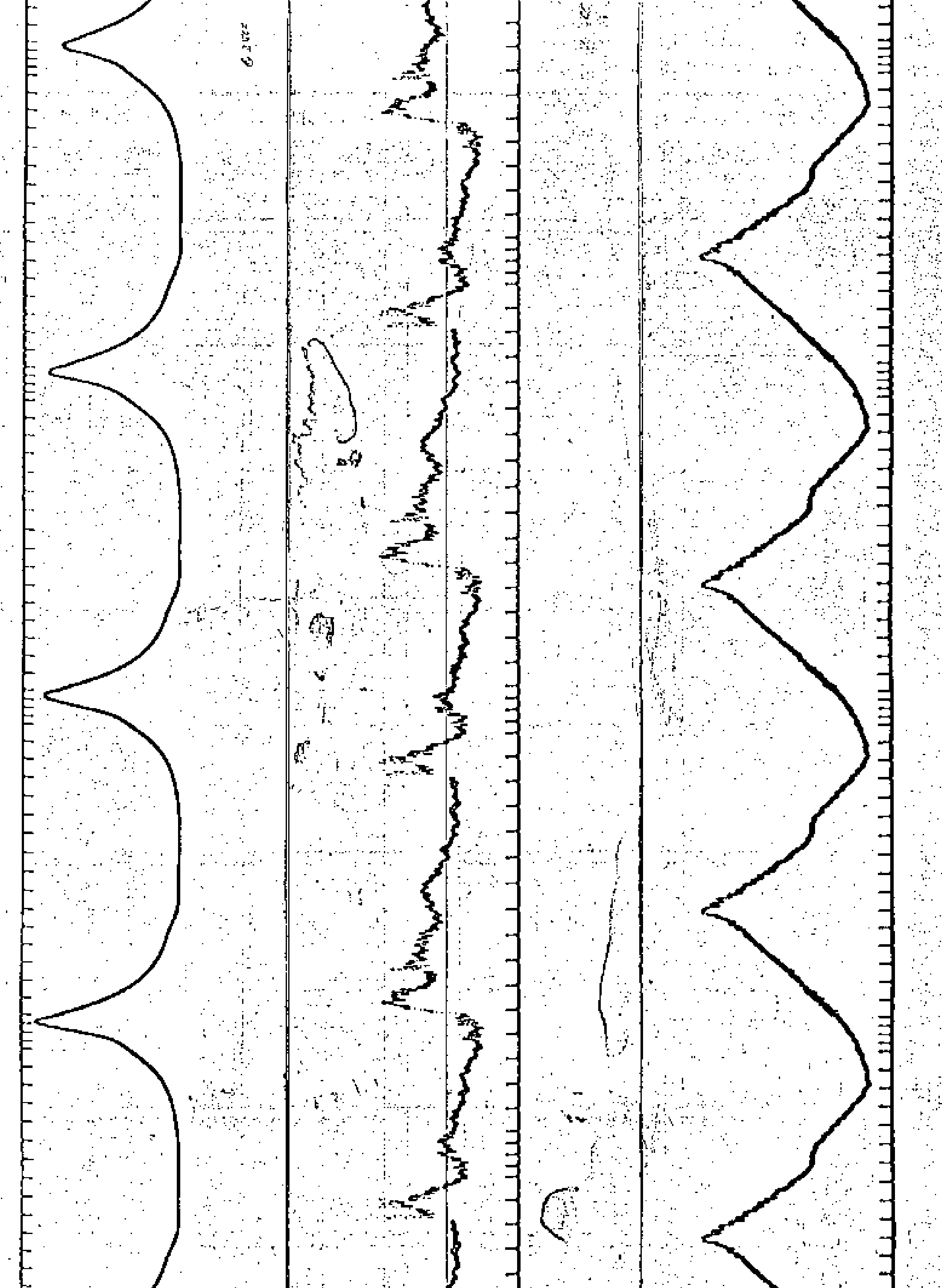
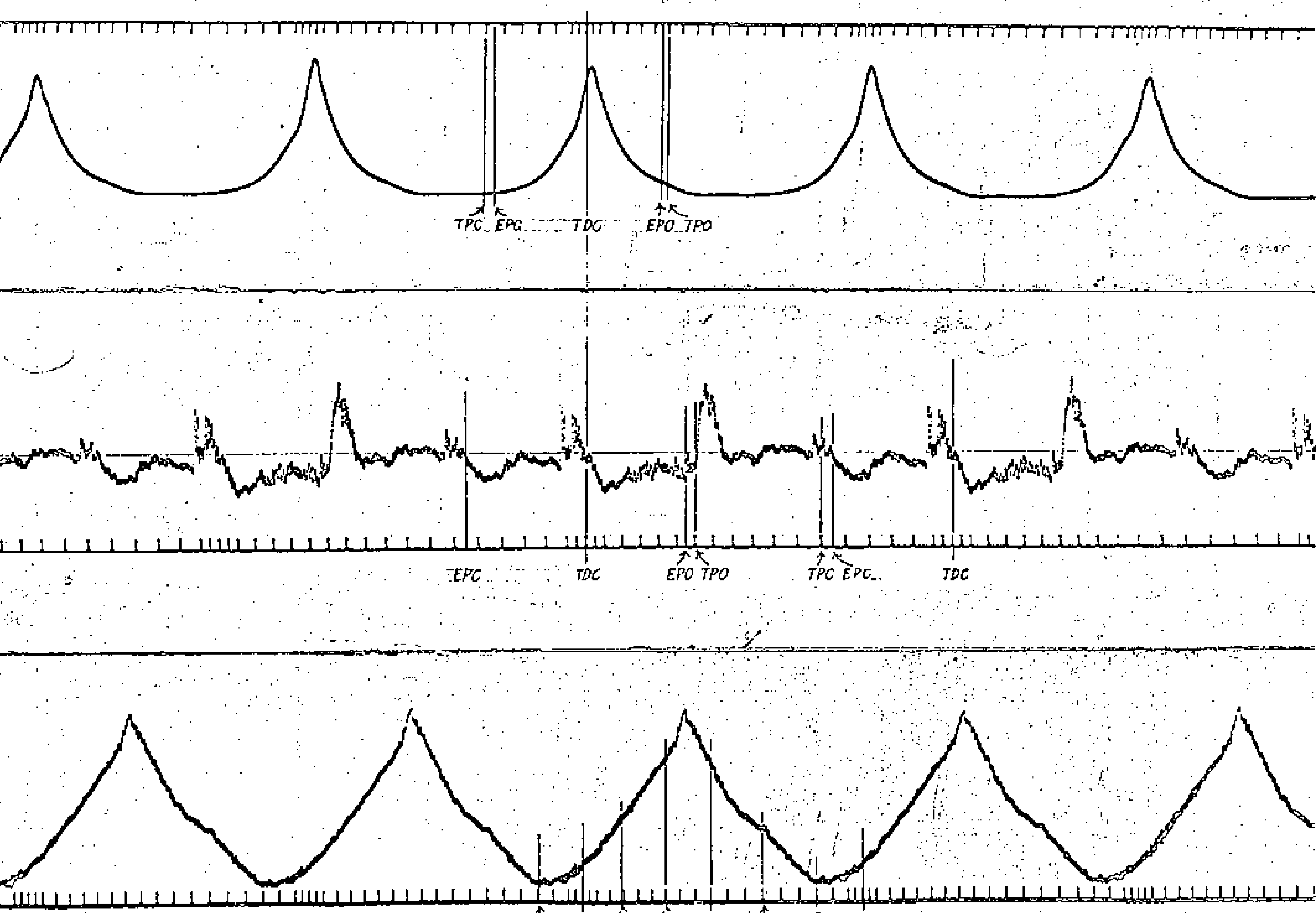


Fig. 19. Oscillograms of Original Setting.

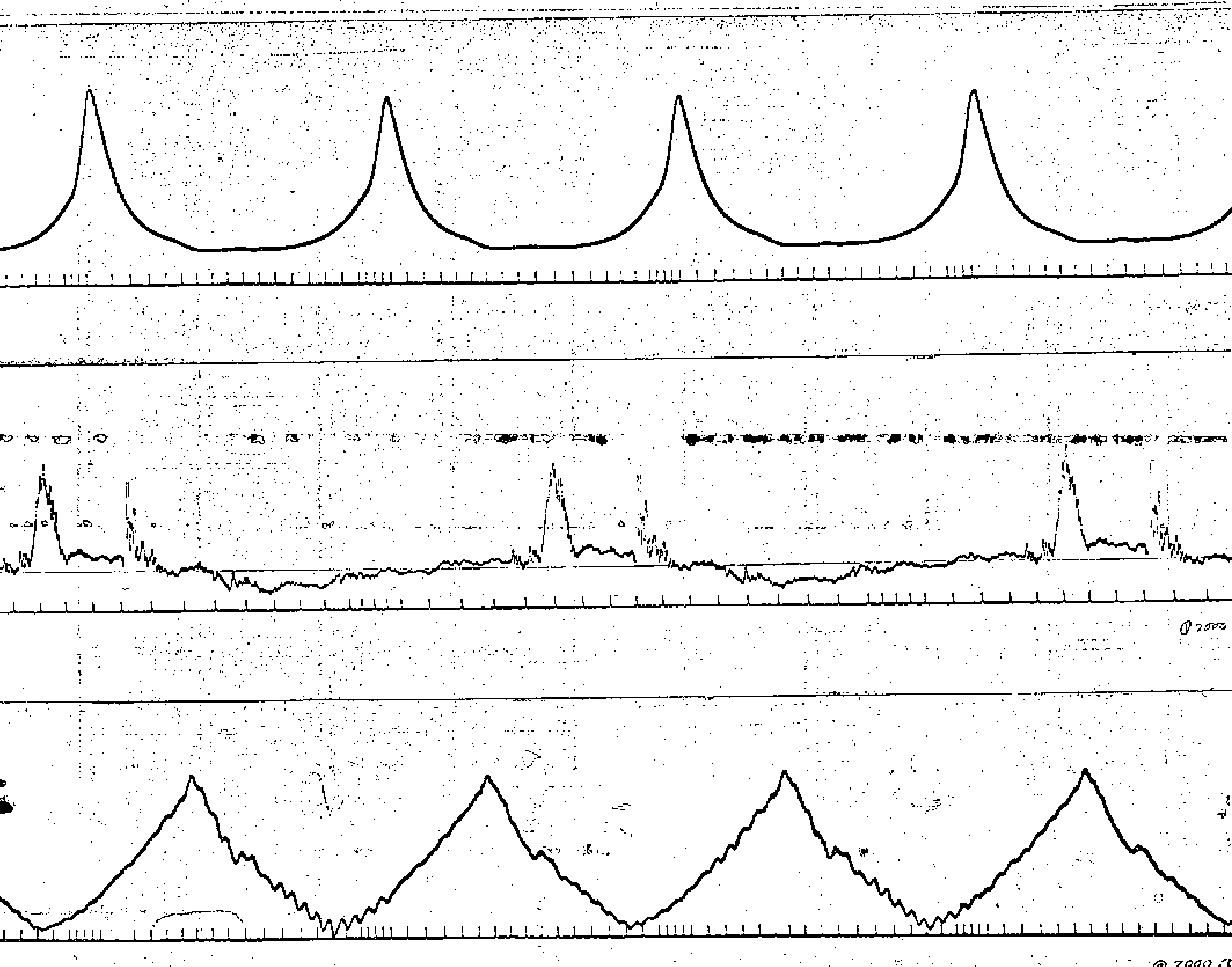


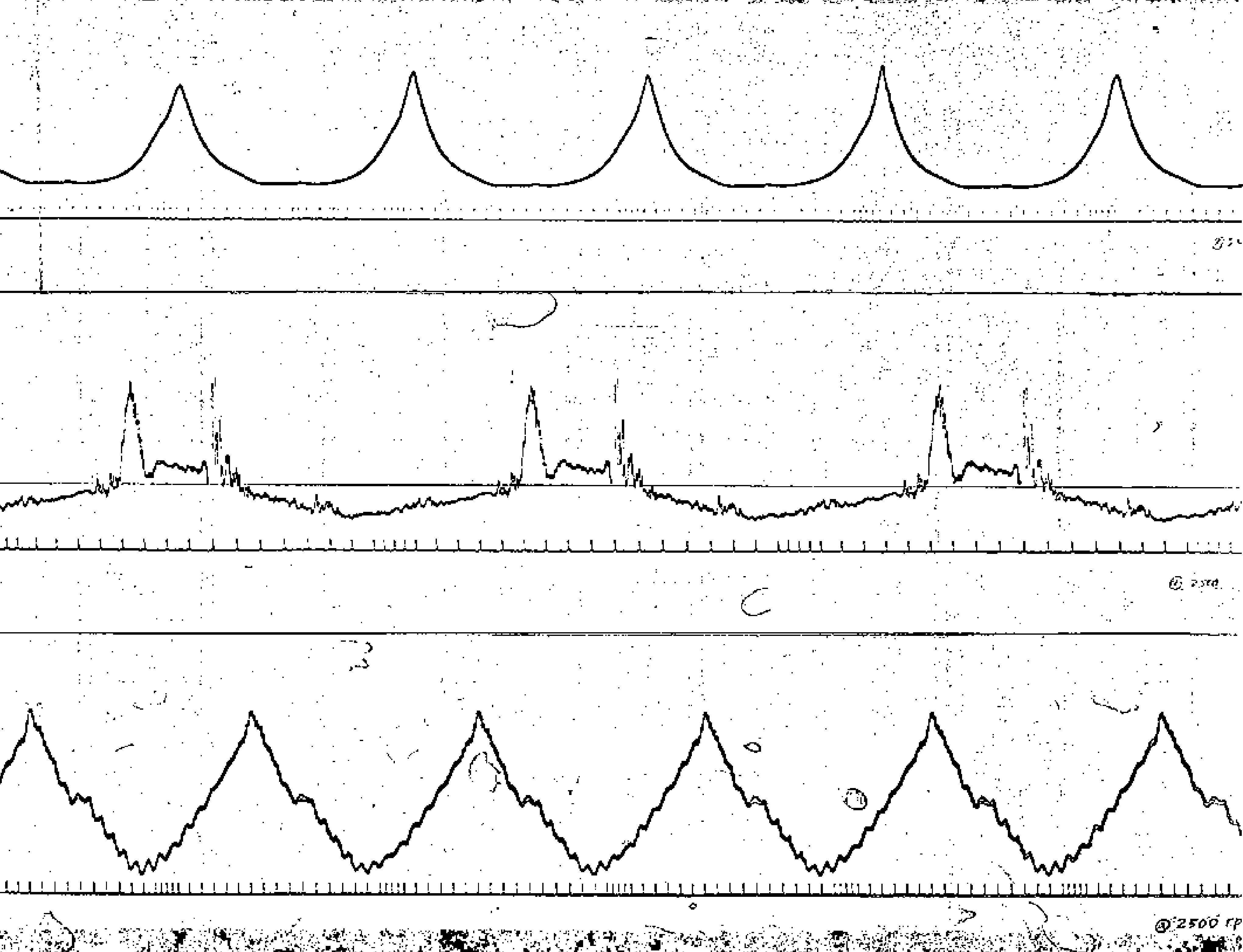


@ 3000 rp.

Fig. 20. Oscillograms of Long Exhaust Pipe Arrangement.

SHORT EXHAUST PIPE




@ 2500 rp

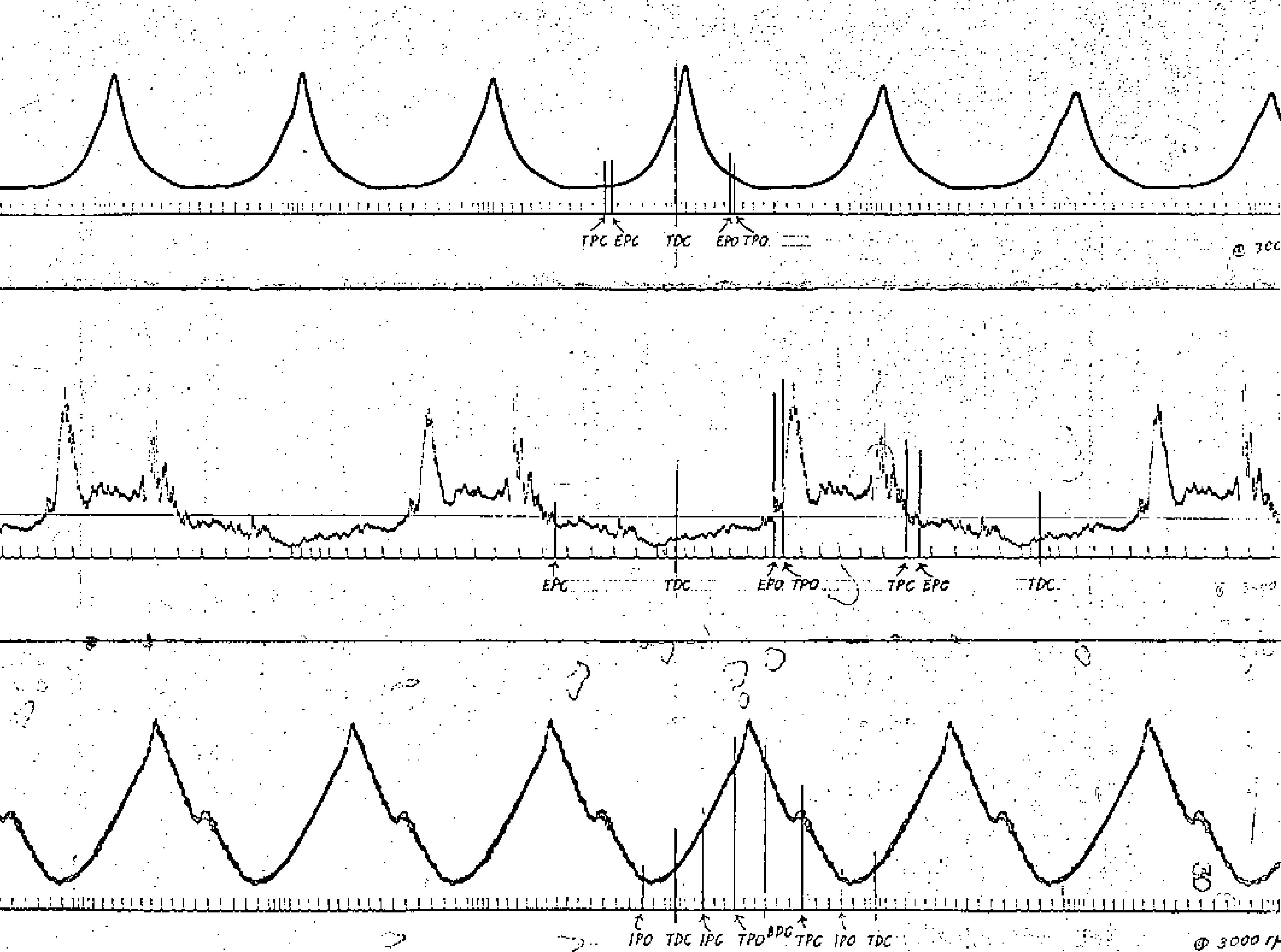
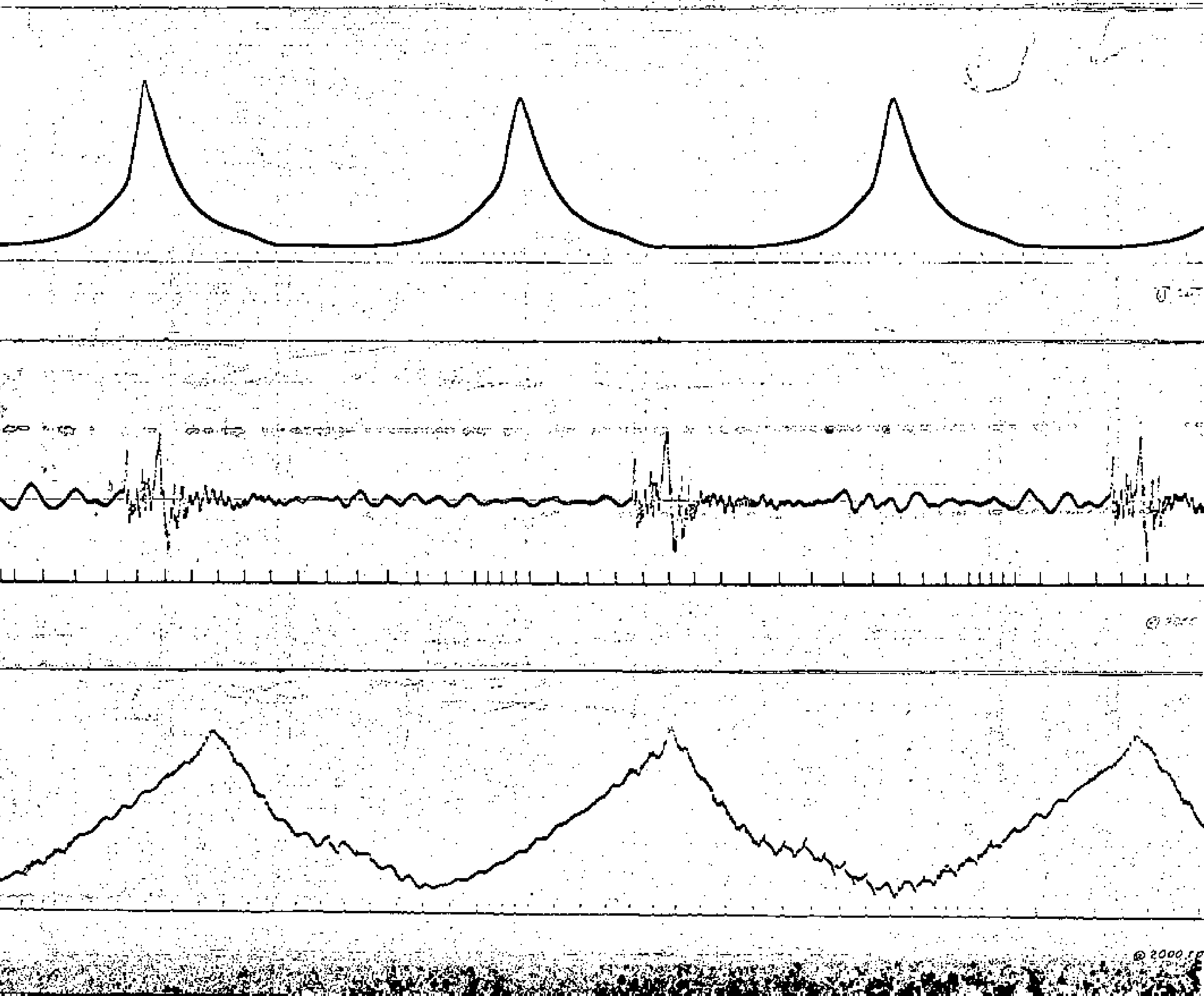
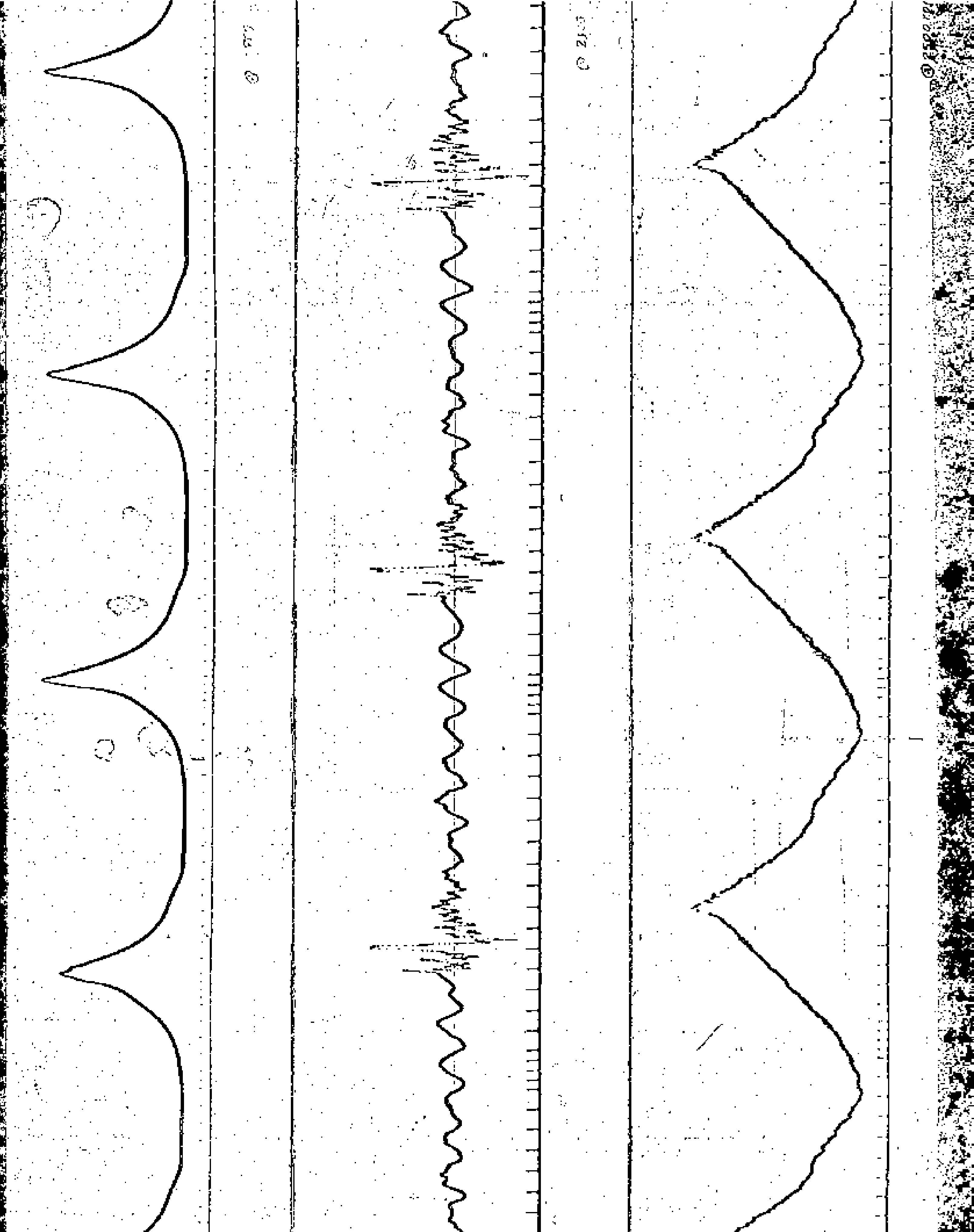


Fig. 21. Oscillograms of Short Exhaust Pipe Arrangement.

FREE EXHAUST





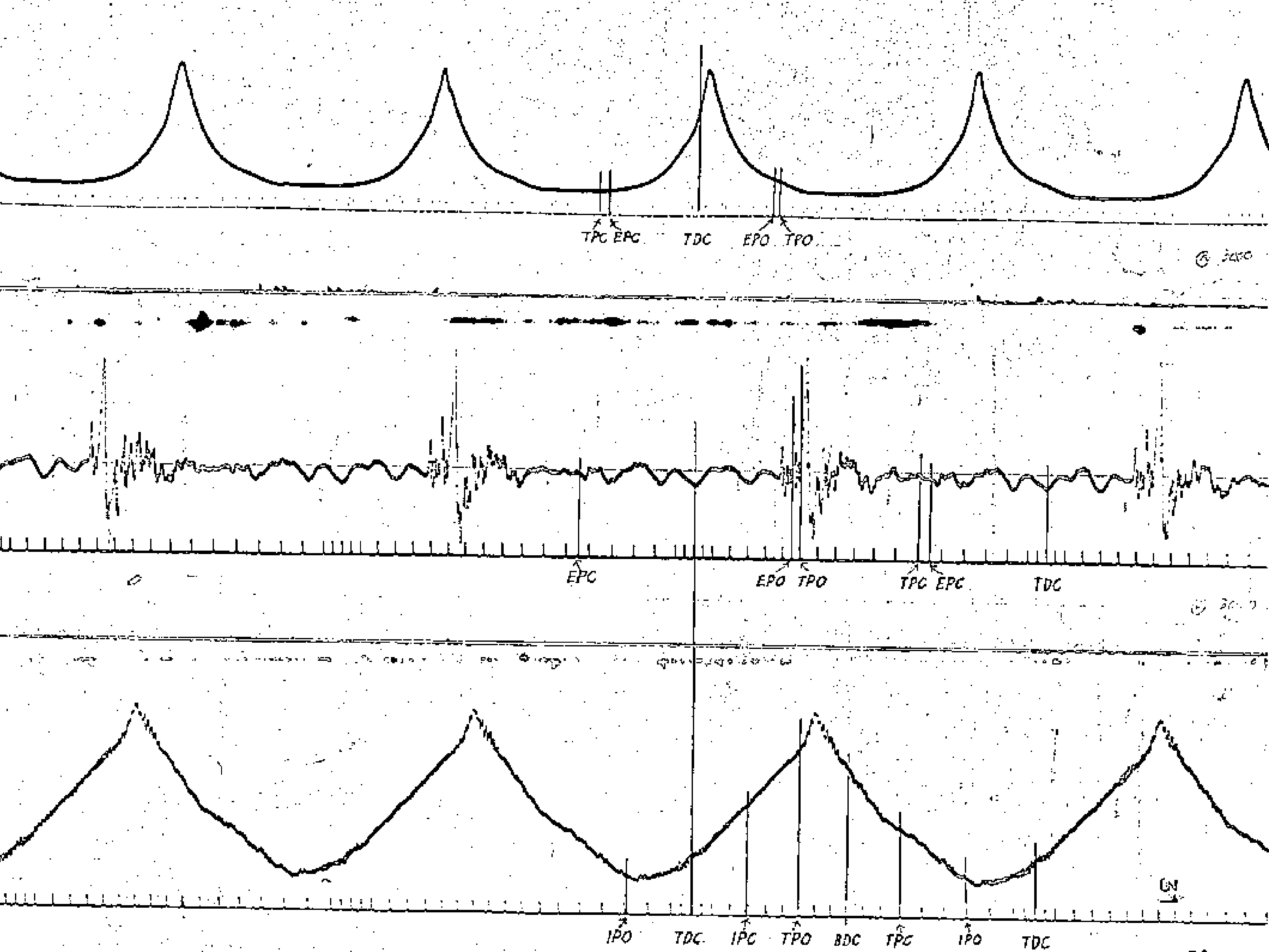


Fig. 22. Oscillograms of Free Exhaust Arrangement.

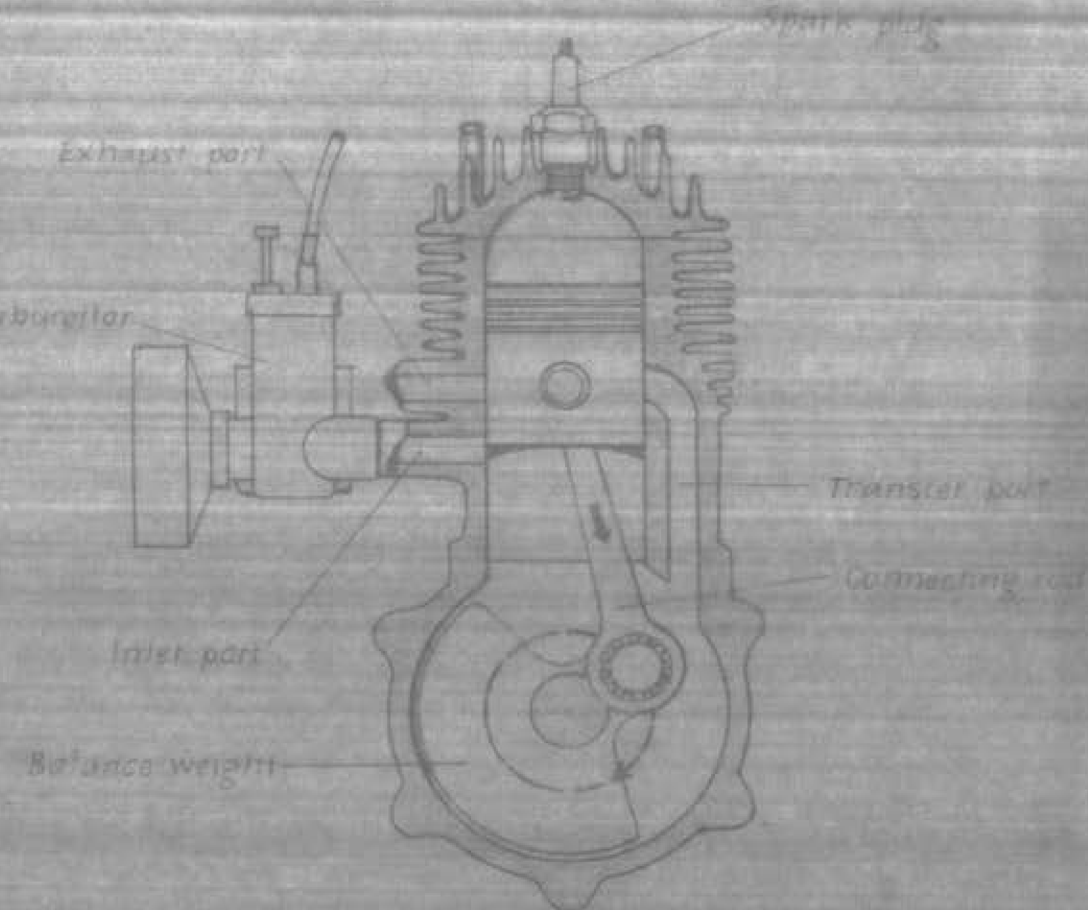


Fig23 Typical 2 stroke Crankcase-Scavenge Engine

at 2000 rpm. The later three and the last three were taken at 2500 and 3000 rpm of engine speed respectively.

The three traces are placed in successive order. The cylinder pressure trace is on the top, below it is the exhaust pressure as it appeared on the oscilloscope screen. At bottom is the corresponding crankcase trace.

Every photograph accommodates the pressure trace and the time base mark. The time base is shown in crankangle degrees, being 20 degrees between successive graduations. At 20 degrees before and after TDC, the time base is marked every 10 degrees. To give added information there are marks showing the ports timing and the positions of the piston shown on the time - base. This is done by the help of the timing diagram showing in Fig. 17. Also, there are atmospheric lines superimposed on every exhaust pressure traces. This is accomplished by the virtue of the Kiestler 737 twoway adaptor. The photographs are best investigated by the aid of the drawings showing positions of ports at each sequence. One of which is shown in Fig. 23.

<u>Investigation the Photographic Traces of Pressure variation</u>

<u>First Arrangement (Original setting)</u>

The first test was made with the system consisted of an expansion chamber mounted immediately after the fitting for adaptor as shown in Fig, 14 a.

The cylinder pressure traces show that the ignition timing was fairly good. The peaks were almost at TDC when the engine was running at low speed, and shifted to about 15 degrees of crank angle after TDC at maximum speed.

Pressure in the exhaust system did not rise rapidly the moment the exhaust port opened. It was when the transfer port opened that the pressure "Kick" appeared. At maximum speed the pressure fluctuated somewhat then it got lower than atmospheric just before the transfer ports closed. At lower speeds the exhaust pressure dropped below atmostpheric faster than at high speed. This make the scavenging more efficient and can be detected readily from the crankcase traces as they were smoother at low speeds. The closing of the transfer ports did not affect the motion of both the wave in the exhaust port and in the crankcase at all. Only an almost unnoticeable hump can be seen on the crankcase pressure trace at maximum speed. Therefore, it was apparent that there was no tendency of the gas to flow from the cylinder back to the crankcase. This indicated a very efficient scavenging and high volumetric efficiency.

Both the exhaust port and crankcase pressure traces were resemble to the ideal oscillogram. This was owing to the effect of the expansion chamber as it damped out the waves from the exhaust port and the reflected aves travelled back from the tailpipe

The crankcase traces were the smoothest of the four, hence consequent pumping loss was small.

Second Arrangement (Long Exhaust Pipe)

Fig. 14b Shows the arrangement of a long pipe of 1 inch nominal diameter joined to the adaptor fitting, The length L was 10 feet and the same expansion chamber was fitted at the end of the pipe. The gas from the tailpipe was conveyed out of the laboratory by a centrifugal blower in the manner shown in Fig. 15.

The exhaust pressure of this arrangement was lower than the first. There were two pressure peaks at the lowest speed (2000 rpm) and three peaks at higher speeds (2500 and 3000 rpm). When the exhaust port opened the pressure hardly reached atmospheric. The kick took place just as the transfer ports opened. Other peaks that occured after the first kick were the reflected wave travelled back from the open end of exhaust system. The transfer ports closed when the pressure in the exhaust pipe was atmospheric at all speed, hence no gas flow back into crankcase.

Since the cylinder pressure did not show any significant difference from the first setting, they will be left uncriticize.

Investigation of the crackcase pressure traces reveals that as the piston moved up from BDC and transfer ports being uncovered, pressure in the crankcase remained constant for a short time. This indicated that the pressure in the crankcase was the same as that in the cylinder and the exhaust port, since at this period both the transfer and the exhaust ports were opened. Therefore a pressure equilibrium existed at this moment. There was no more fresh charge flow into the cylinder, nor exhaust gas flow back from the port. After the piston covered the transfer ports, the pressure in the crankcase continued to descend until it was somewhat lower than atmospheric. By the time the inlet port was opened the pressure began to rise again, and as the piston was moving downwards from TDC, the pressure continued to rise consistently.

The reflected pressure peak in the exhaust system took place after both the exhaust port and the transfer ports were covered. So it has no effect in the scavenging process of the engine.

Crankcase pressure traces of this arrangement were very resemble to the ideal one, especially at low speed.

Consequently the scavenging of this arrangement was better than the first, The exhaust pressure traces differed from the ideal in regard of reflected waves. Thus, in view of scavenging the exhaust gas, this arrangement was better than the first.

Third Arrangement (Short Exhaust Pipe)

The third arrangement was almost the same as the second, but the pipe of 49 inches long was used instead of the 10 feet one. The photograph of this arrangement is shown in Fig. 15, and the length of the pipe is L=49 inches for the drawing shown in Fig. 14 b. From the cylinder pressure traces, the ignition advance was very late. Combustion occured at TDC at all speeds. There were noticable bumps during the scavenging period that were corresponded to the reflected waves in the exhaust pipe at every speed.

There were two peaks of pressure waves appeared per cycle of engine in this arrangement of exhaust system. The amplitude of the reflected waves were almost the same as the fundamental ones, being slightly smaller at low speed (2000 rpm) and a little higher at high speeds (2500 and 3000 rpm). The characteristic of the exhaust pressure at the opening of the exhaust and transfer ports was the same as the previous setting. The pressure in the exhaust system was well above atmospheric all the time after the transfer ports opened. The reflected waves travelled back to the exhaust port just before the transfer ports were covered by the piston, and so there was a tendency for the exhaust gas to flow back into the crankcase. This made volumetric efficiency of the engine drop and was the reason why an improper tuning of the exhaust system caused power drop in the crankcase scavenge engine.

There were evident humps on the crankcase pressure traces at all speeds. Before the transfer ports were uncovered the pressure in the crankcase rose, then fall down after the ports were closed. The waviness of the traces after the ports were closed was due to the shock occured in the crankcase.

The pressure variation inthe exhaust port and the crankcase of this arrangement had nothing in common with the ideal pressure variation. this arrangement gave the worst performance of the three.

Fourth Arrangement (Free exhaust)

The last arrangement was made on the engine exhaust system by taking off the pipe and the expansion chamber, leaving the transducer fitting on the exhaust port only. The performance of this arrangement was expected to be the best.

Oscillograms of the pressure in the cylinder indicate a fairly good spark advance. Combustion took place at about 10 degrees of crankangle before TDC. The peak was at 10 degrees after TDC at all the speeds.

It is very interesting that although the engine exhaust into atmosphere, the amplitude of the exhaust pressure at peak was not higher than that in the third arrangement. Furthermore, after the pressure reached its peak, it dropped sharply below atmospheric. This was different to the ideal trace as there was no time that the ideal exhaust pressure be lower than atmospheric. But this can be considered as an advantage for it tended to draw fresh charge of mixture into the cylinder. After the piston reached BDC, the pressure in the exhaust system was fluctuating at the atmospheric line. This favored the scavenging process because it would not retard the delivering of the fresh mixture from the transfer ports.

The fluctuating of the exhaust pressure had some effect on the pressure in the crankcase at low speed. The crankcase pressure trace at low speed revealed the waviness of the pressure during the period the transfer ports was shut. This was thought to be the effect of the charge rushing into the crankcase. At higher speeds the waviness disappeared.

Comparing the exhaust and the crankcase traces to the ideal oscillogram is very disappoint. Only the crankcase pressure variation at high speeds resemble to the ideal, showing that the effects of the speed are very great to the exhaust pressure.

FUEL CONSUMTION TEST

Description

In order to investigate the performance of the engine with different exhaust arrangement, the rates of fuel consumption were measured. The results were plotted in the form of fuel consumption "hook curves" for comparison between each setting.

To produce such curves, the carburettor was provided with a fixed needle valve and constant throttle opening. The position of the throttle was marked by a pointer and was set in the position that the engine took the load up to maximum B.H.P. with-out difficulties. The rate of consumption of petroil was measured by noting the time the engine took to consume a calibrated quantity in the Flint fuel gauge. The weight of this quantity of fuel was measured by allowing the fuel between the marks to flow out through the test cork into a weighted vessel. The increase of weight corresponding to the stated quantity was then the desired fuel weight being consumed in each test. This fuel gauge and the test bed are shown in Fig. 24. Engine output was measured by means of a self-excited dynamometer coupled to the engine. The data of the test equipment is shown below.

Fig. 24

Fuel Gauge

Engine: JAP 16 H

Bore and Stroke	57.mm. x 62.mm.
Swept Volume	158 c.c.
Carburetter	Amal Fixed Jet Carburetter Type 321
Maximum BHP	2.9 @ 3000 rpm.

Dynamometer: B.K.B. Frame 609

Capacity	4 H.P. @ 3000 rpm.
Maximum speed	3000 rpm.
Brake arm	10.504 in.
BHP	$\frac{WN}{6000}$

The dynamometer is a D.C. swung field dynamometer with a speed range up to 3000 rpm. The machine is shunt wound and self excited. The electrical output is absorbed in a separately wall mounted load unit.

In the experiment, Torque was measured by a spring balance and speed by a tachometer and counter. The tachometer was used to adjust the speed up to the desired range and the counter used to record the exact revolutions in one minute interval.

Operation

The engine was started in the usual way and after a few seconds was brought to load. Then it was kept running for three minutes to ensure warming up.

The field rheo-stat was adjusted to give the loading required. The tachometer was observed that the engine was running steadily, then the spring balance was read and recorded at the start of each run.

After these were done, the cork to the tank was shut off and the air vent at the top of the fuel gauge was open immediattely. The engine was then drawing fuel from the gauge. The timing clock was started when the fuel level passes the top spacer and the counter was clutched in simultaneously. The time taken by the engine to consume the quantity of fuel between the first and the second spacer was recorded while the clock was allowed to run continuously. After a period of one minute the counter was then de-clutched to give the revolutions reading. At this moment the engine was still drawing fuel from the gauge. The clock was stopped the moment fuel passing the third spacer and the time recorded. The spring balance reading was recorded again at the end of the run.

The experiment was repeated with different speeds by opening the cock below the fuel tank, thus allowing fuel to top up the fuel gauge. After the glass vessel was filled the cock was shut and the experiment carried on in the same manner.

Results from the four arrangements are tabulated and shown in the following tables. The graph obtained by plotting the specific fuel consumption on a base of the corresponding values of brake mean effective pressure is also shown in Fig. 25.

The values appeared in the tables are calculated as follow:

B.H.P. The length of the brake arm is 10.504 in. therefore the corresponding value of torque when the spring balance read w lb, is

$$2 \times \frac{10.504\ w}{12} \text{ lb-ft}$$

From the formula $\text{H.P.} = \frac{TN}{33,000}$

where T = torque in lb-ft

N = revolutions per minute

$\therefore$ BHP of the engine $= 2 \times \frac{10.504\ WN}{12 \times 33,000}$

$$= \frac{WN}{6000}$$

BMEP The swept volume of the engine is 158 c.c.

This is equal to $\frac{158 \times 1728}{28,320}$ cu.in.

From BMEP = $\dfrac{\text{brake work of the engine (in-lb)}}{\text{total volume swept by the piston in the same interval}}$

$$= \frac{WN}{6000} \times 12 \times 33,000 \times \frac{1}{\frac{158 \times 1728}{28,320} \times N}$$

$$= 6.85\ W$$

Table a Fuel Consumption Measurement of the J.A.P. 16 H Engine with the Exhaust Arrangement Showing in Fig 14a (Original Setting)

No. of Test	W lbs	N Revs/min	Time taken for consuming 1 graduation of fuel		Duration of test in minutes	BHP	BMEP lb/sq.in	S.F.C. lb/bhp-hr
			min	sec				
1	5.375	808	1	22.30	1.370	0.724	36.80	17.30
2	6.000	845	1	30.80	1.513	0.845	41.10	13.42
3	6.500	1028	1	12.00	1.200	1.113	44.40	12.84
4	6.938	1878	-	41.23	0.687	2.172	47.55	11.51
5	7.125	1952	-	37.88	0.632	2.320	48.80	11.70
6	6.813	1984	-	39.93	0.666	2.255	46.60	11.44
7	5.875	2589	-	36.63	0.611	2.535	40.22	11.08
8	5.250	2845	-	37.13	0.620	2.490	35.90	11.12

Note 1 Graduation = 0.286 lb of fuel

Condition of test : Constant throttle valve setting

Petroil Mixture : 20 parts petrol to one part S A E 30 lubricating oil.

Table b   Fuel Consumption Measurement of the J.A.P. 16 H Engine with the Exhaust Arrangement Showing in Fig14B(Long Exhaust Pipe, L = 120 in)

No. of test	W lbs	N Revs/min	Time taken for consuming 1 graduation of fuel		Duration of test in minutes	BHP	BMEP lb/sq.in	S.F.C. lb/bhp-hr
			min	sec				
1	6.625	814	1	24.25	1.404	0.898	45.3	13.60
2	6.750	1406	1	2.50	1.042	1.580	46.2	10.42
3	7.125	1894	-	50.25	0.838	2.250	48.8	9.10
4	6.413	2532	-	44.25	0.732	2.710	43.9	8.67
5	5.625	2970	-	41.50	0.692	2.780	38.4	8.94

Note 1   Graduation = 0.286 lb of fuel

Condition of test & Petroil Mixture same as table a.

Table c Fuel Consumption Measurement of the J.A.P. 16 H Engine with the Exhaust Arrangment Showing in Fig 14 (Short Exhaust Pipe, L=49 in)

No. of Test	W lbs	N Revs/min	Time taken for consuming 1 grauation of fuel min	 sec	Duration of test in minutes	BHP	BMEP lb/sq.in	S.F.C. lb/bhp-hr
1	6.000	810	1	18.55	1.309	0.810	41.10	16.20
2	6.125	1415	-	48.60	0.810	1.443	41.90	14.69
3	6.375	2004	-	41.30	0.689	2.128	43.60	11.30
4	6.063	2126	-	41.33	0.690	2.150	41.50	11.59
5	5.813	2256	-	43.05	0.718	2.190	39.90	10.90
6	5.438	2364	-	43.73	0.729	2.140	37.10	11.01
7	4.813	2566	-	45.58	0.759	2.060	33.00	10.97

Note 1 Graduation = 0.286 lb of fuel

Condition of test & Petroil Mixture same as table a.

Table d   Fuel Consumption Measurement of the J.A.P. 16 H Engine with the Exhaust Arrangement Showing in Fig.14e(Free Exhaust)

No. of Test	W lbs	N Revs/min	Time taken for consuming 1 graduation of fuel min	sec	Duration of test in minutes	BHP	BMEP lb/sq.in	S.F.C. lb/bhp-hr
1	5.875	802	1	25.90	1.432	0.785	40.20	15.27
2	5.813	824	1	28.45	1.474	0.798	39.80	14.58
3	7.562	1400	-	53.28	0.889	1.768	51.70	10.92
4	7.875	1924	-	39.43	0.657	2.528	54.00	10.34
5	6.625	2456	-	39.95	0.666	2.710	45.30	9.50
6	5.875	2958	-	36.00	0.600	2.895	40.25	9.88

Note 1   Graduation = 0.286 lb of fuel

Condition of test & Petroil Mixture same as table a.

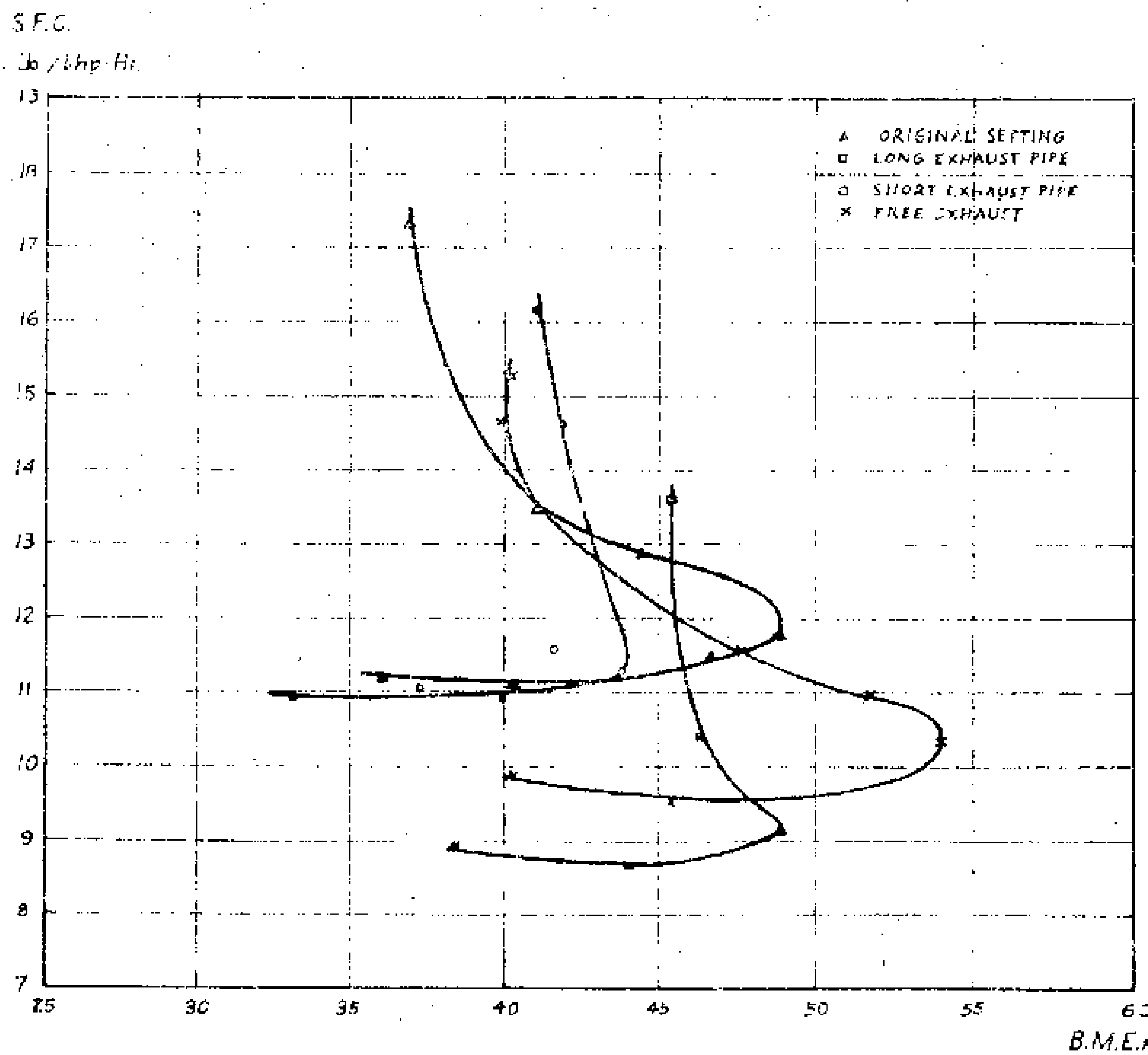


Fig. 25

Fuel Consumption Curves in Comparison

Interpretation of Curves

The four hook curves shown in Fig 25 reveal the effect of exhaust tuning. It is expected that the free exhaust arrangement should give the best BMEP and the lowest specific consumption. This is partly true, as the BMEP of the free exhaust setting is the maximum while the consumption is well above the long pipe arrangement. The higher consumption indicates that some fresh charges of mixture are lost through the exhaust part. As for the long exhaust pipe the reflected waves restrict this flow of fresh charge, pushing it back into the cylinder and so higher volumetric efficiency is obtained.

The original arrangement of the exhaust system gives the highest consumption but almost the same maximum BMEP as the long pipe setting. This again shows the loss of charge into the expansion chamber.

The lowest maximum BMEP is obtained in the short pipe arrangement while the consumption is slightly below the original setting. This is due to the reflected waves which occur before both the transfer ports and the exhaust port close. These waves of very high amplitudes drove the charge back into the crankcase. The power delivered from the engine was the least as it hardly reached 2.2 BHP at maximum output.

The above interpretation confirms the result from the oscillograms. It emphazises that the length of the pipe plays a great part in the performance of the engine. The long pipe gives a satisfactory performance for the lowest consumption of fuel while the short pipe gives the contrary.

<u>Engine Noise</u>

Taking the oscillograms of the atmospheric pressure at different setting of exhaust systsm into consideration gives a cluc of sound level of various arrangements. The upper four traces of Fig. 26. Show almospheric pressure variations while the engine was running at 2000 rpm. The oscillogram were recorded using the transducer equipped with the two-way adaptor as described before. The bottom trace is the pressure fluctuation recorded while the engine was being driven to the speed of 1000 rpm by the dynamometer. Of all the four upper traces the free exhaust trace is the most wary one. It was noticed that the sound intensity was very high with this type of exhaust system filted to the engine. The most steady trace is that obtained from the short exhaust pipe arrangement. The first two oscillogramsare almost the same.

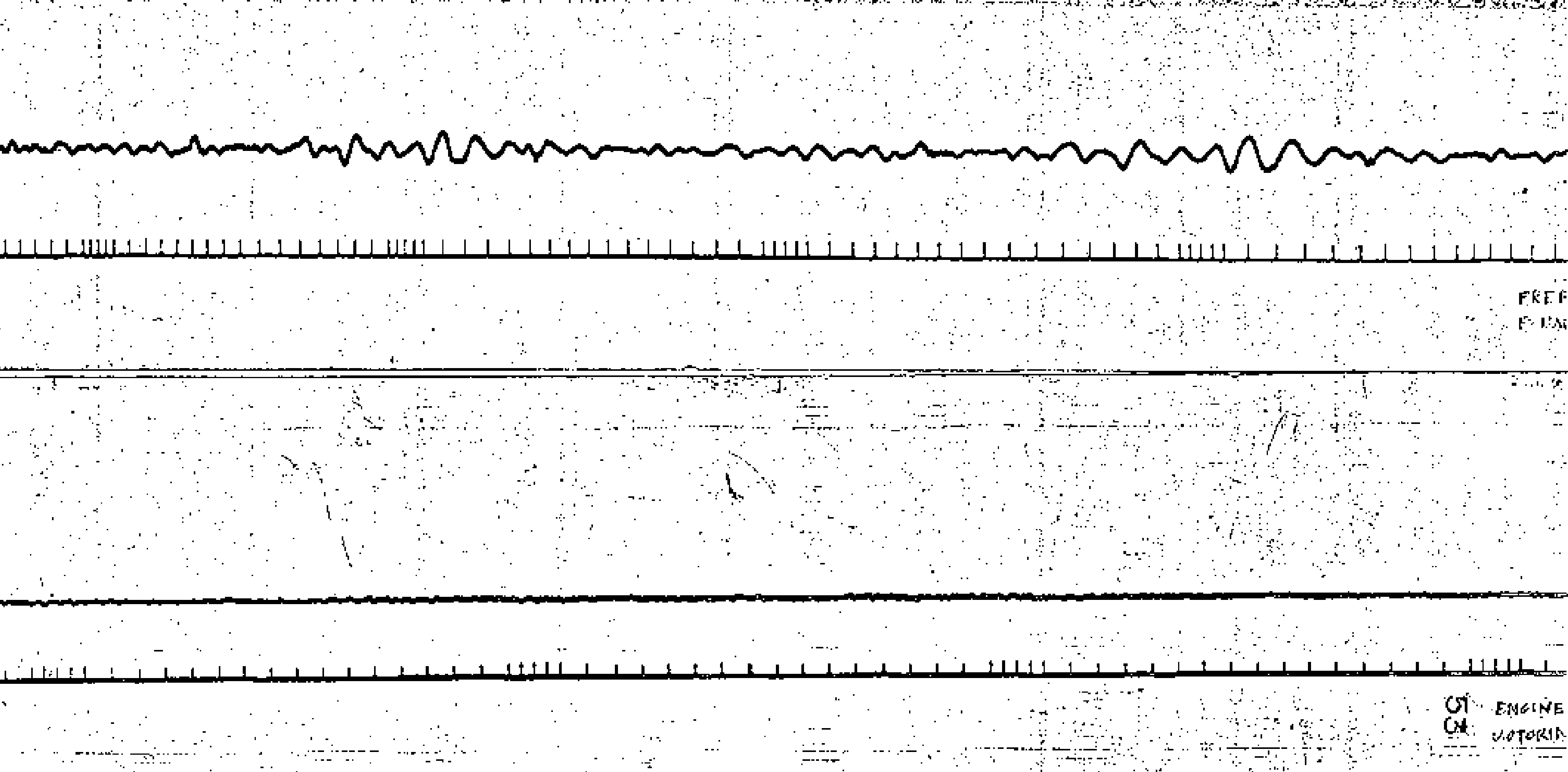
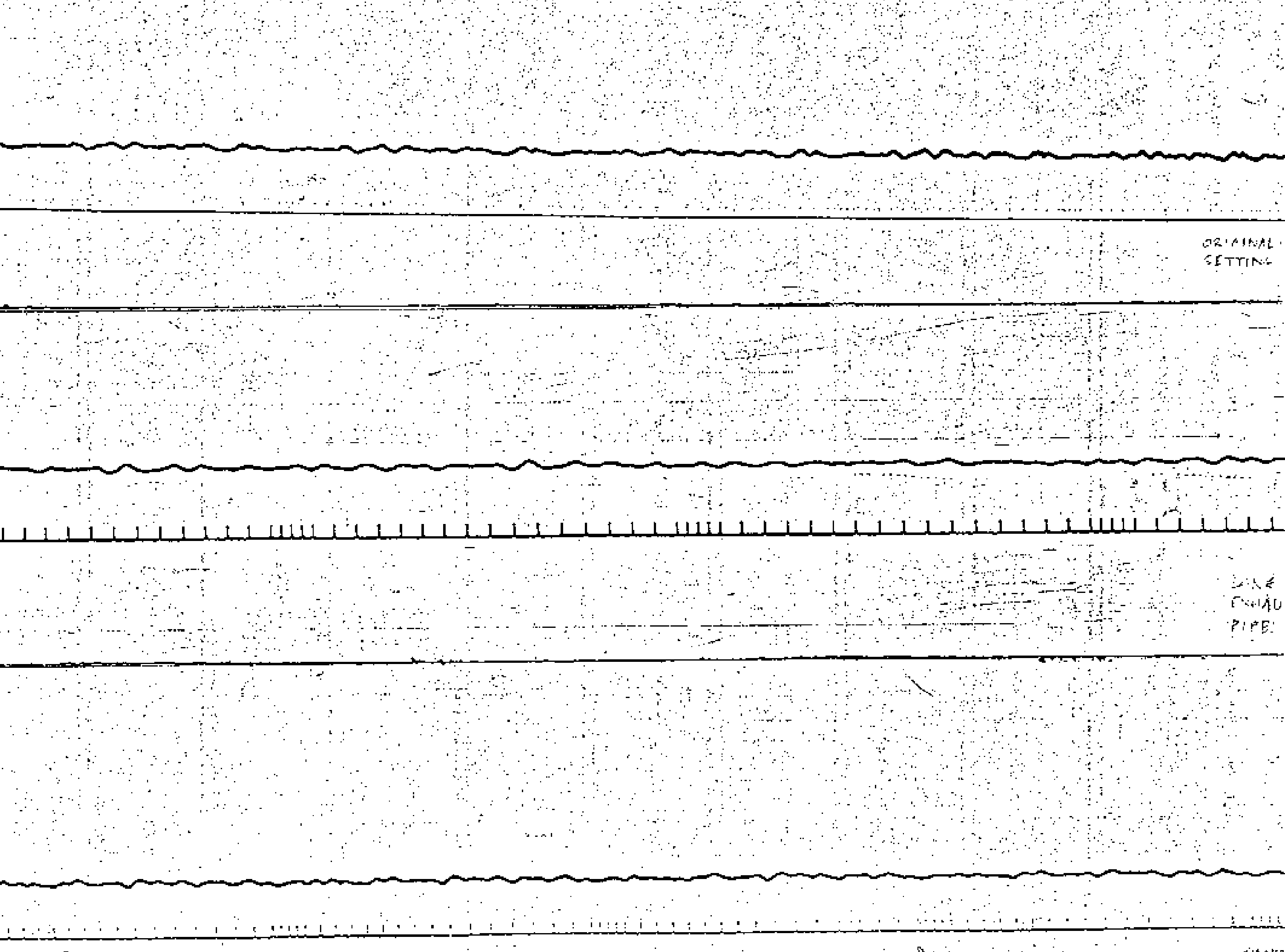


Fig. 26. Atmospheric pressure traces at different settings


ORIGINAL
SETTING
PIPE

As a comparison the atmospheric pressure was recorded in the same way but with the engine at rest. It is a straightline which indicates no pressure disturbance on the transducer. Fig. 27 shows such a trace on the top. The lower two trace are the sound of a blow, on a table and the sound of an electric grinder respectively. The amplitude of this two traces were magnified ten times. The trace of the blow is very complicated, combining of very unruly waves. For the electric grinder, the pressure wave shows to be regular with the fundamental waves of 50 cps. and overtones. Since the recording was made when the engine was not running, the time base was switch to be driven by 50 cps. mains frequency.

From the foregoing investigation, the atmospheric pressure display on the oscilloscope can be considered as a rough indication of the sound intensity each exhaust system produces.

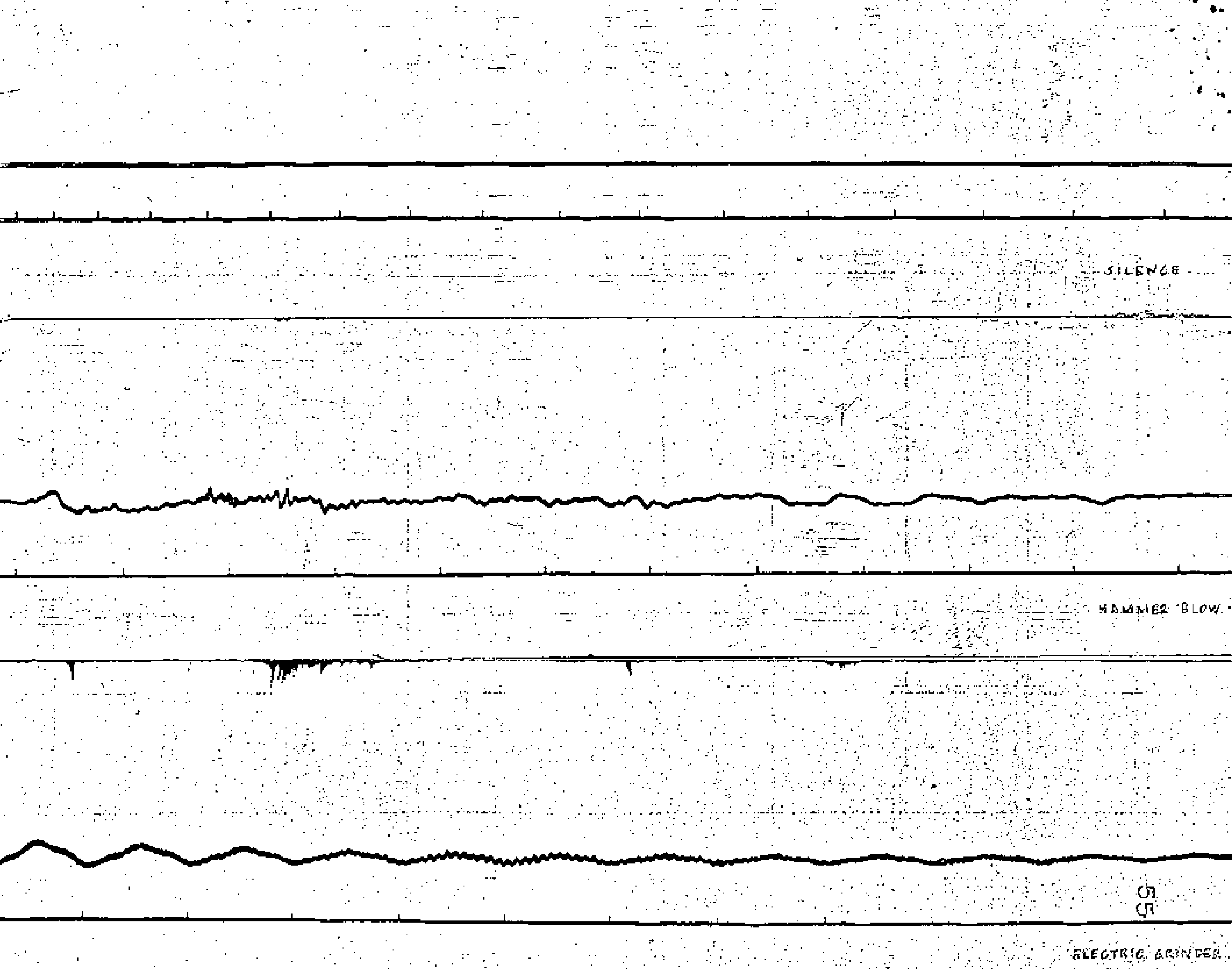


Fig. 27. Oscillograms of noise caused by various sources

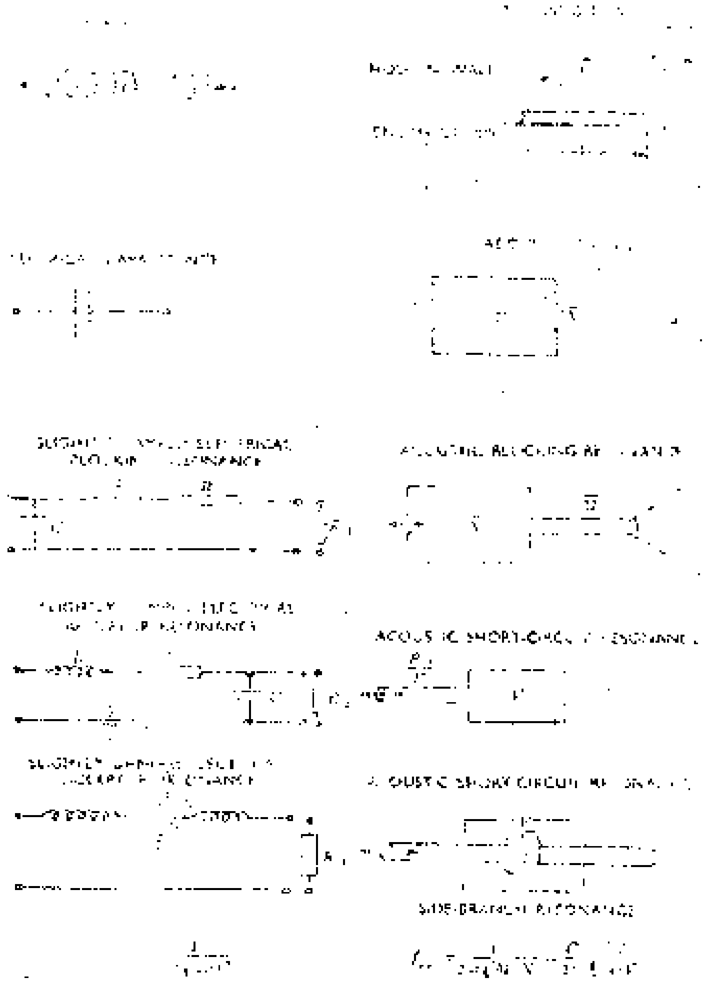
Electrical Analogy

The duty of the silencer is to smooth out the pulsating continuous flow which contains the audible alternating flow. At their exit only the inaudible purely continuous flow should emerge. Therefore, the silencer must possess a high alternating reactance for all the exhaust frequencies. Furthermore, in order to avoid the loss of engine power, the whole exhaust passage including the silencer must not restrict the leaving gasses but let the exhaust escape in perfect freedom.

In old practice, the silencing of exhaust noise is done by means of severe throttling resistances. This is replaced by the modern method of inserting reactances instead of frictional resistances which increase the restriction. These are comparable with electrical inductances and capacitances. Table 8. shows the acoustic components required for the silencing together with their electrical analogies and their resonance posibilities.

It would facilitate the design of exhaust system if, by an electrical analogy, the performance could be estimated. It is an impossible design task to test every possible arrangement on a test bed. The following is a discussion of the settings in the experiment with their electrical analogy.

Table [illegible]



Reproduced from [illegible] "Measurement of the Noise of Motor Vehicles" Reference no. 4 in Bibliography.

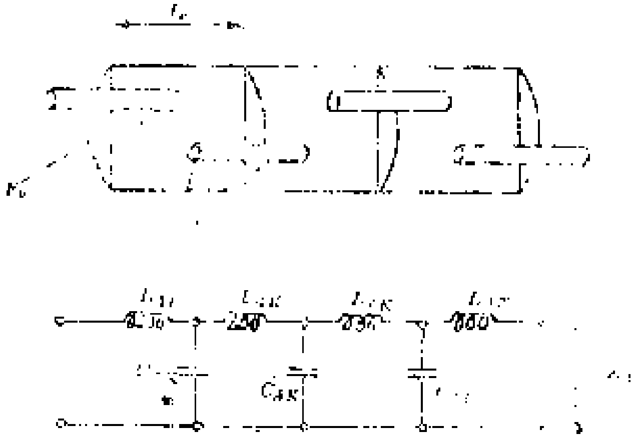


Fig. 28 Rejection-type Filter and its Equivalent Electrical Circuit

The first arrangement can be compared to the third item in table 8. It is a reflection filter which passes the lowest frequencies and blocks all oscillations above a certain limiting frequency. This arrangement is good for a particular speed of engine and if the engine is to be operated at a wide range of speeds, this type of silencer should be made in series forming a reflection-series filter such as shown in Fig 28 with its equivalent electrical circuit. The series should be made by graduated construction of the individual filter stages, in which the longitudinal dimensions of all the chamber and tubes are different.

The second and third arrangement can be compared to the fourth item of the table. These arrangements are also effective at one particular speed only and to use it on the engine operating at a wide speed range should require a reflection series filter to damp out the harmonics as well.

Because of the lack of any acoustic mass or spring in the last setting, it cannot be related to the item in the table. The only thing that dampen the pulse of exhaust gases is the room acoustic, the effect of which can readily be seen from the oscillogram of Fig. 22.