CHAPTER IV

RESULTS AND DISCUSSION

In this chapter, results obtained experimentally and theoretically from two 4-cylinder in-line engines, one is conventional type and the other equipped with second-mode balancer system, are presented mostly in graphical form as well as discussion of results attempting to raise out the interesting points of this experiment.

In Fig. 4-1 to Fig.4-10, the acceleration of vibration for conventional and counter-balancing shaft engines at different measuring points are plotted against the octave filter centre frequencies. Generally, these figures exhibit same trend such that the attempt to discuss each figure in details would be exhaustive and give no further benefit. However representative figures are treated in details and points of interest raised for others.

Fig. 4-1 to Fig. 4-5 are acceleration of vibration at various locations of conventional engine.

When the speed of engine is 1000 rpm., in Fig. 4-1, first mode vibration could not be measured due to octave filter centre frequency starting at 31.5 Hz. corresponding to second mode at this speed. The curves of different loadings indicate second mode vibration level is the highest followed by fourth mode at 63 Hz., sixth mode at 125 Hz. and twelvth mode at 250 Hz.

At 2000 rpm, vibration at first, second, fourth, eight and sixteenth modes are shown at 31.5, 63, 125,250 and 500 Hz. respectively. For all loadings, second mode vibration are the highest followed by fourth mode.

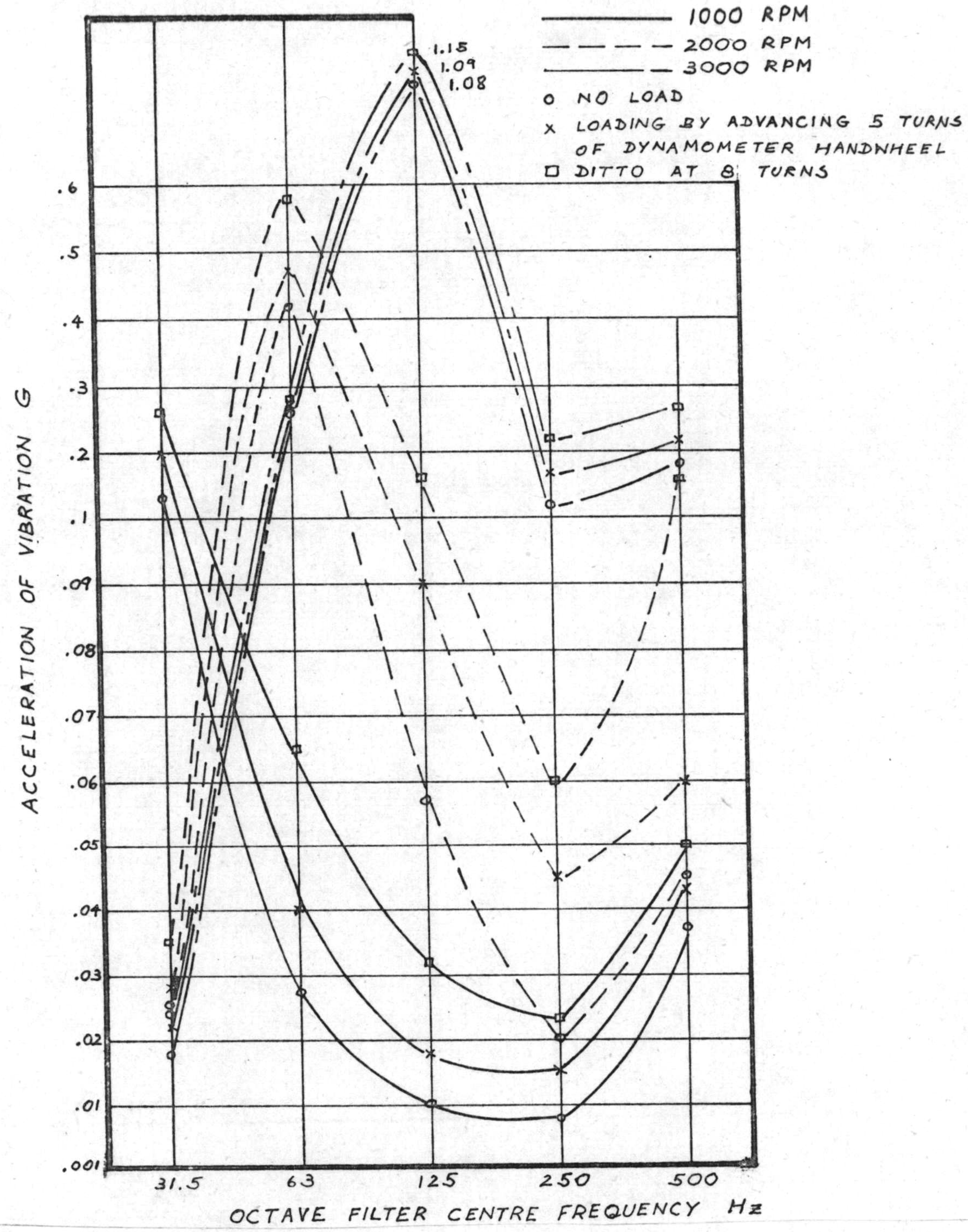


Fig.4-1 Acceleration of vibration at left support bracket plotted against octave filter centre frequency for conventional engine

At 3000 rpm; vibration at half, first, second, fourth and eight modes are recorded at 31.5, 63, 125, 250 and 500 Hz. respectively. Predominant values prevail at 125 Hz. coinciding with second mode followed by first mode and fourth mode.

It can be said from the results of Fig. 4-1 that at engine support bracket, the engine vibration mainly arises from second mode vibration. All the curves at 500 Hz. corresponding to twenty-fourth, sixteenth and eight modes for speeds of 1000, 2000 and 3000 rpm. respectively indicate rising of vibration which contradicts to the theory stating that subsequent modes of vibration would become smaller as the orders become bigger. Also as shown in Appendix Fig. A-1 to Fig. A-2, vibration in the range 500 Hz. up to 8000 Hz. show abnormally high level. This can be explained by noting that the method of measuring vibration by probe with handheld would give false response in this range resulting in high level of vibration due to natural frequency of the measuring instrument. Thus, these figures cannot represent the engine vibration. Fig. 4-11 shows the natural resonant frequency of probe measurement(8).

Fig. 4-2 shows the acceleration of vibration at valve rocker cover which is similar to Fig. 4-1 but the acceleration values are different.

In Fig. 4-3, the curves for carburetor vibration suggest that the carburetor has a natural frequency at 250 Hz. which is well away from normal operating speed of engine to ensure no excessive vibration existing in normal driving.

Vibration in Fig. 4-4 and Fig. 4-5 is measured at engine test bed closed to engine rubber isolation. The magnitudes of vibration are small since well designed rubber mounting act as a good isolator permitting very little vibration transmitting to the automotive body. Comparing Fig. 4-1, vibration at engine support bracket, to Fig. 4-4 and Fig. 4-5 indicates that rubber isolator

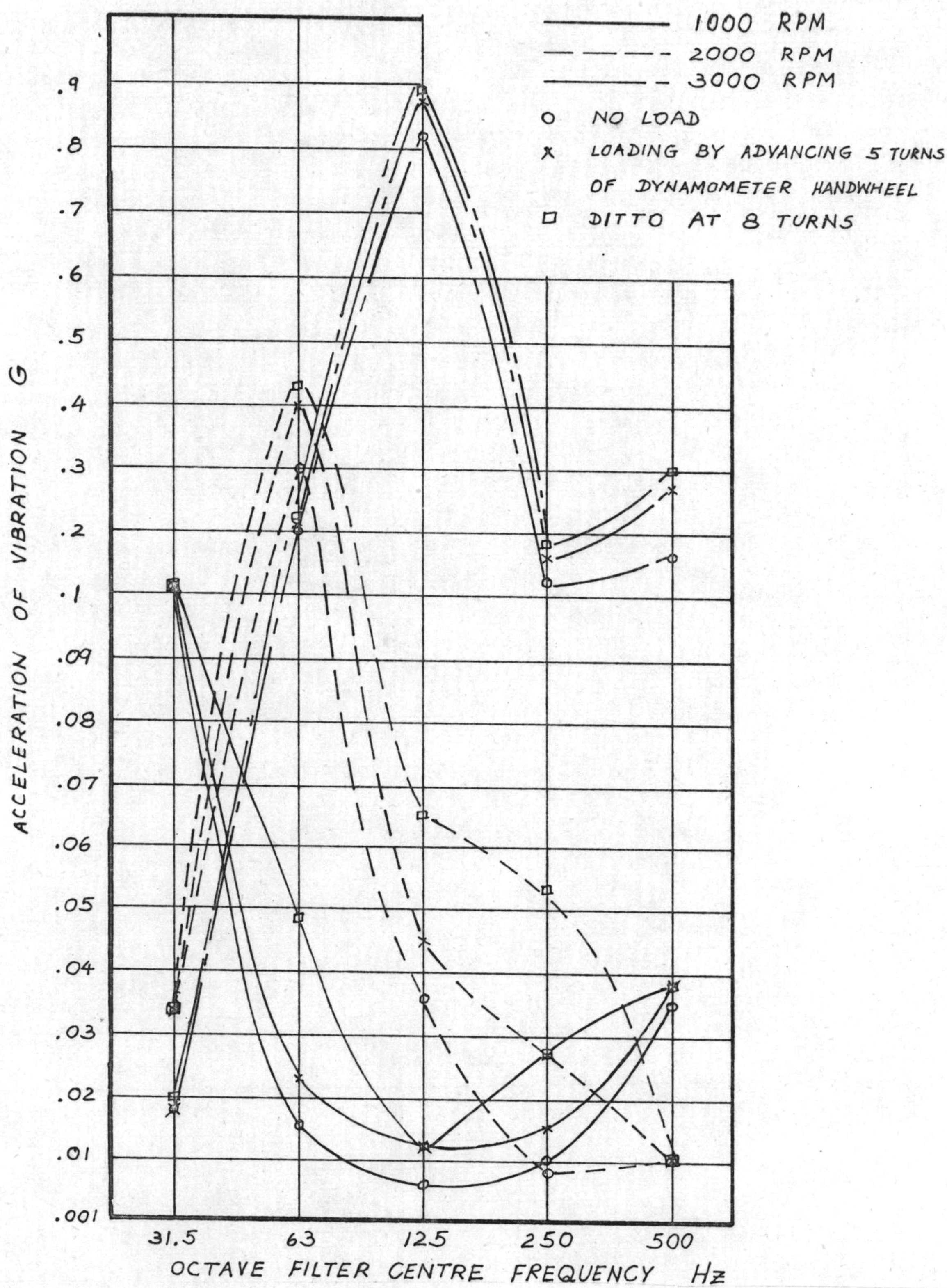


Fig.4-2 Acceleration of vibration at valve rocker cover plotted against octave filter centre frequency for conventional engine

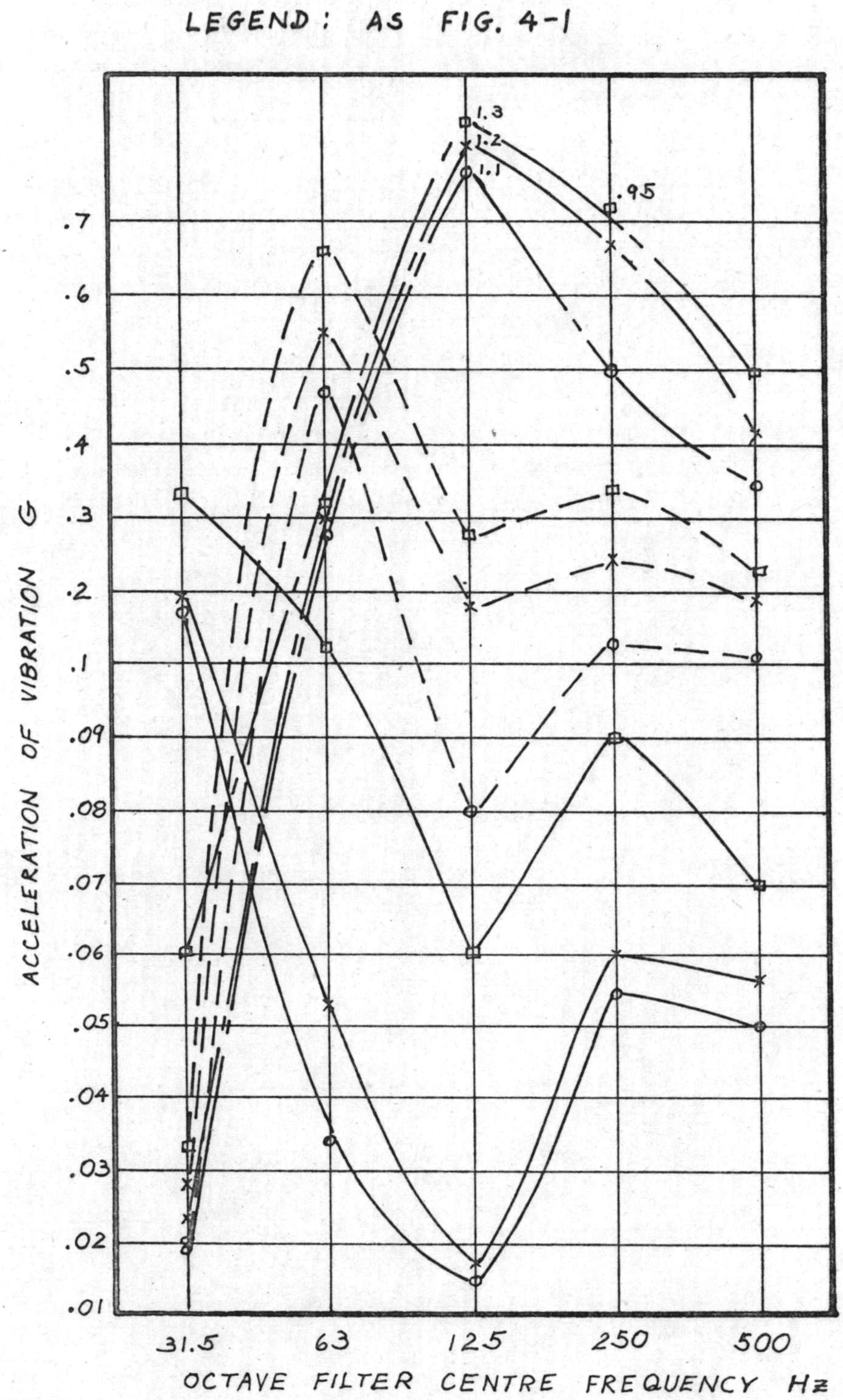


Fig.4-3 Acceleration of vibration at carburetor plotted against octave filter centre frequency for conventional engine

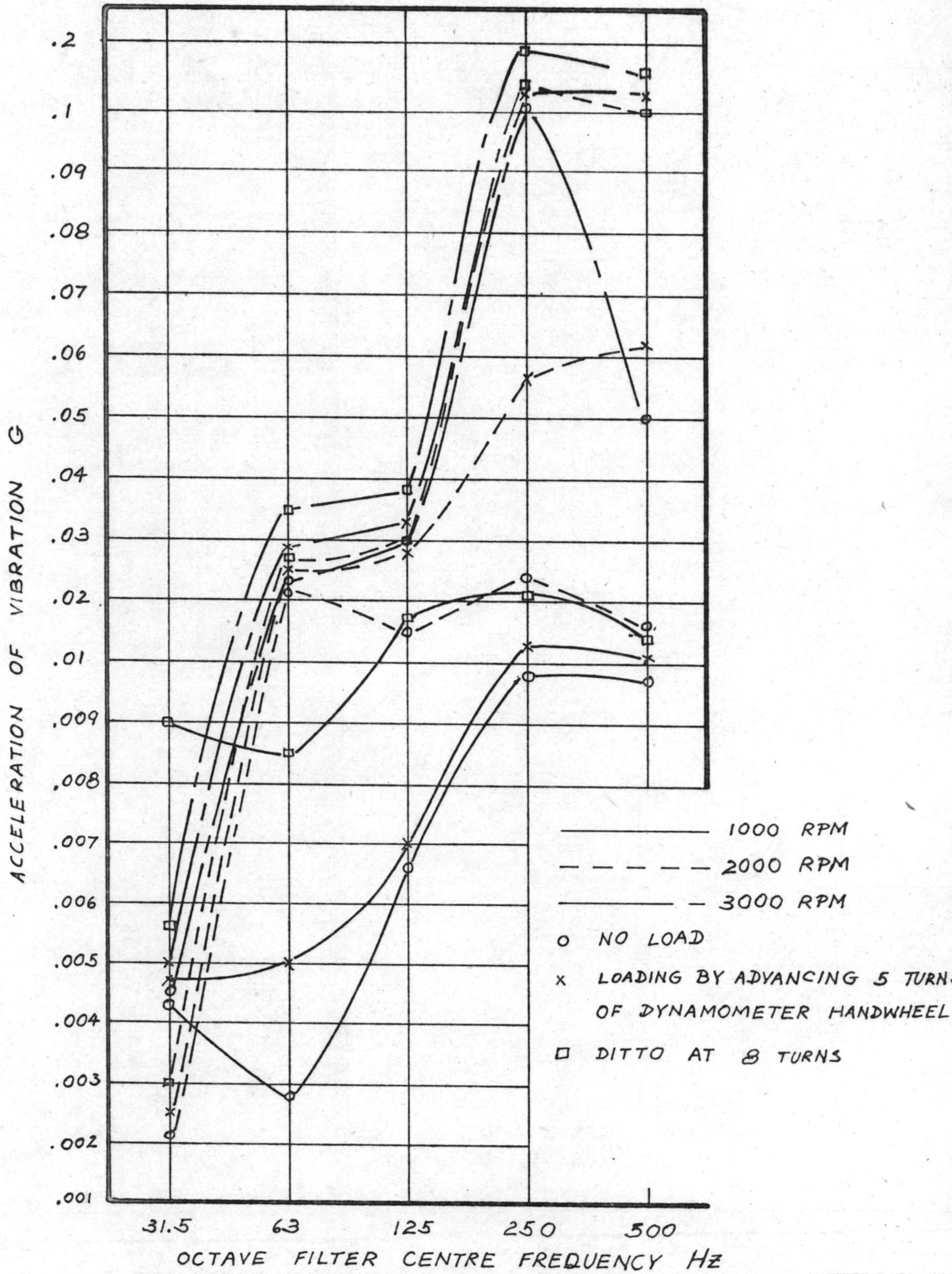


Fig.4-4 Acceleration of vibration at left engine test bed plotted against octave filter centre frequency for conventional engine

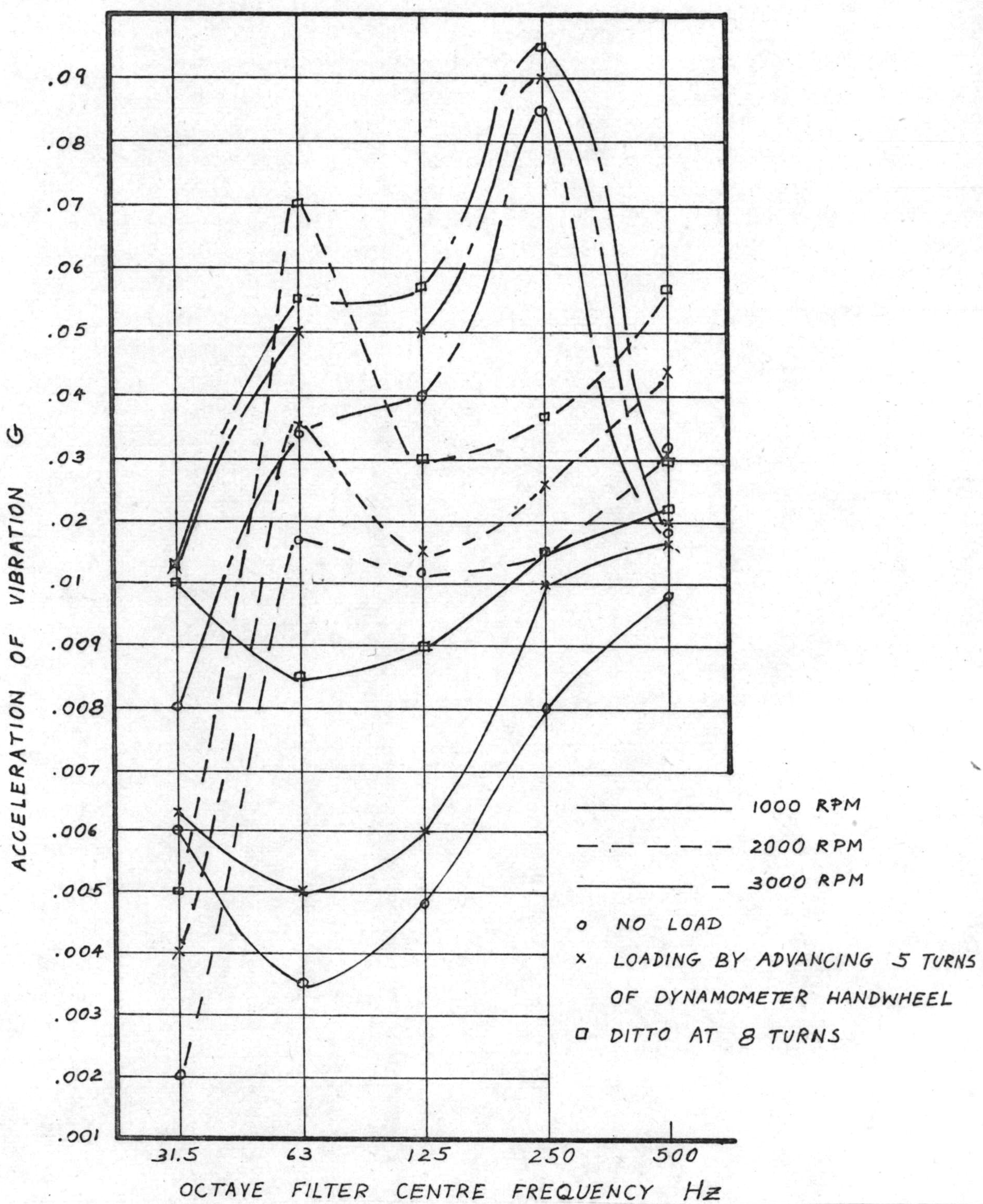


Fig.4-5 Acceleration of vibration at right engine test bed plotted against octave filter centre frequency for conventional engine

could reduce vibration as much as ten times.

Fig. 4-6 to Fig. 4-10 are the values measured from counter-balancing engine showing the results of vibration at various modes at different measuring points.

The vibration in Fig. 4-6 and Fig. 4-7 for right and left engine support brackets respectively exhibit same trend as in conventional engine namely second mode vibration is predominant at test speeds even though the engine is equipped with second mode balancer system. Vibration values at both supports are not equal and tend to be higher on the left side. This is so because weight distribution of the engine is lighter at this side resulting in bigger amplitudes.

Fig. 4-8, at valve rocker cover acceleration of vibration is less than at engine support brackets but as shown previously for conventional engine the values at both locations nearly equal.

Vibration curves for carburetor in Fig. 4-9 suggest there exists a resonant frequency at 250 Hz. similarly to the one on conventional engine. This might be an inherent design feature of Solex carburetor incorporated on both engines.

In Fig. 4-10, the vibration of engine test bed closed to engine rubber isolator shows resonant frequency at 125 Hz. Comparing Fig. 4-6 with Fig. 4-10 indicates vibration level has diminished about ten times passing through engine rubber isolator, the same as found in conventional engine.

Results of Fig. 4-1 to Fig.4-10 altogether indicate that in 4-cycle 4-cylinder in-line engine there are many modes of vibration arising from various exciting forces and moments.
By proper choice of instrument any mode of vibration can be measured. At all different measuring locations, second mode acceleration levels are highest with some exceptions where resonants occur. Figures show

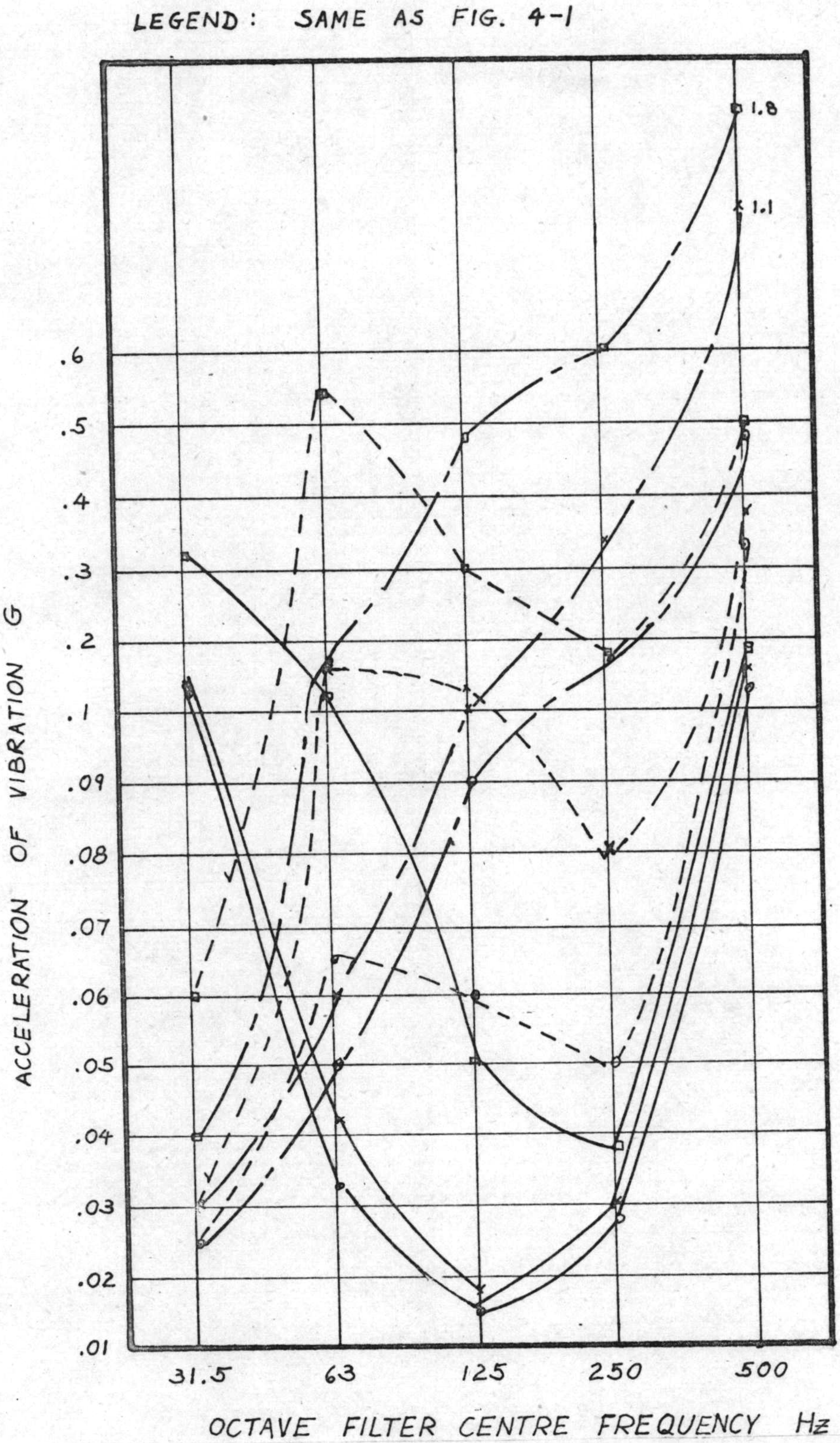


Fig.4-6 Acceleration of vibration at right support bracket plotted against octave filter centre frequency for balanced engine

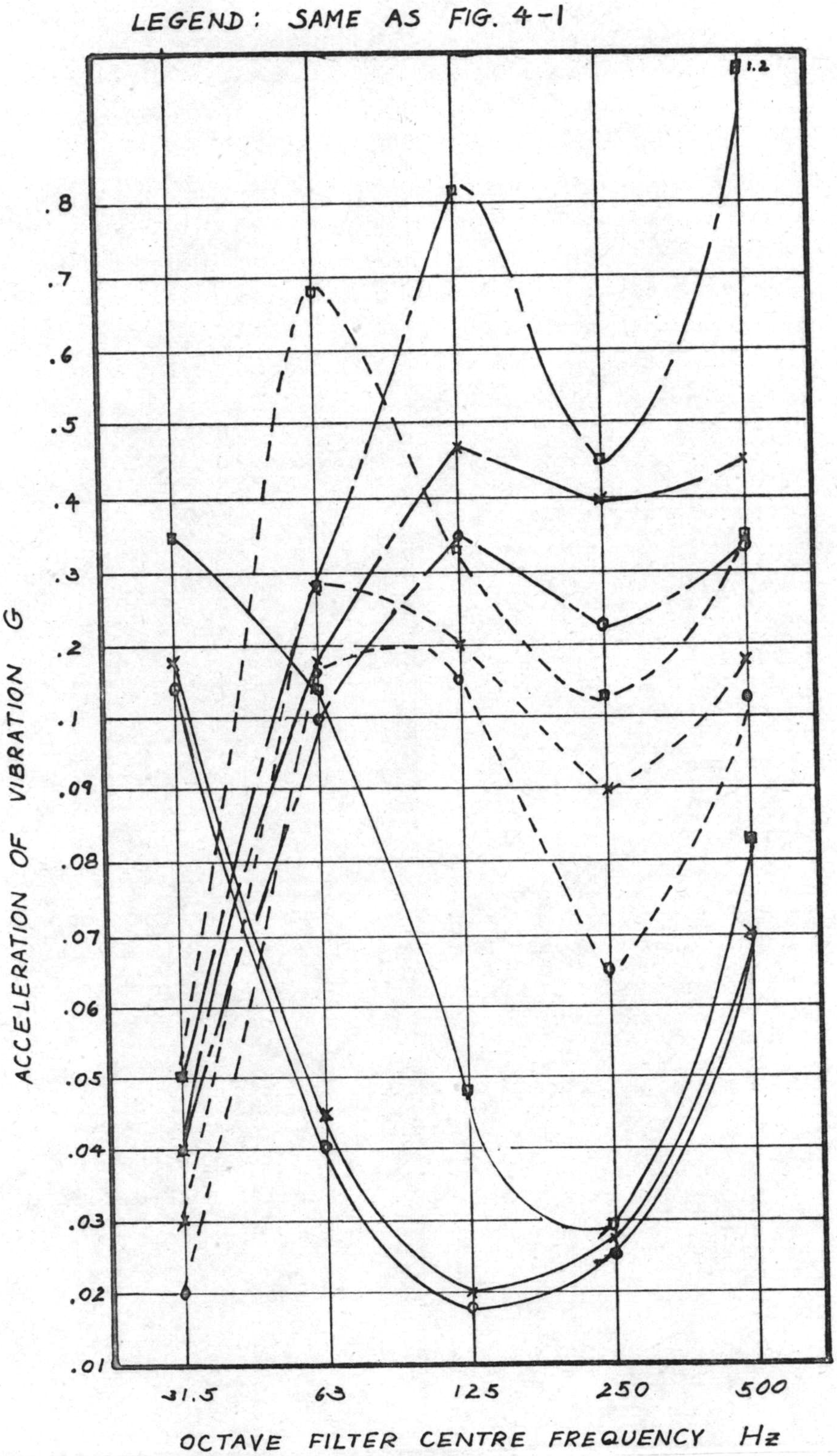


Fig.4-7 Acceleration of vibration at left support bracket plotted against octave filter centre frequency for balanced engine

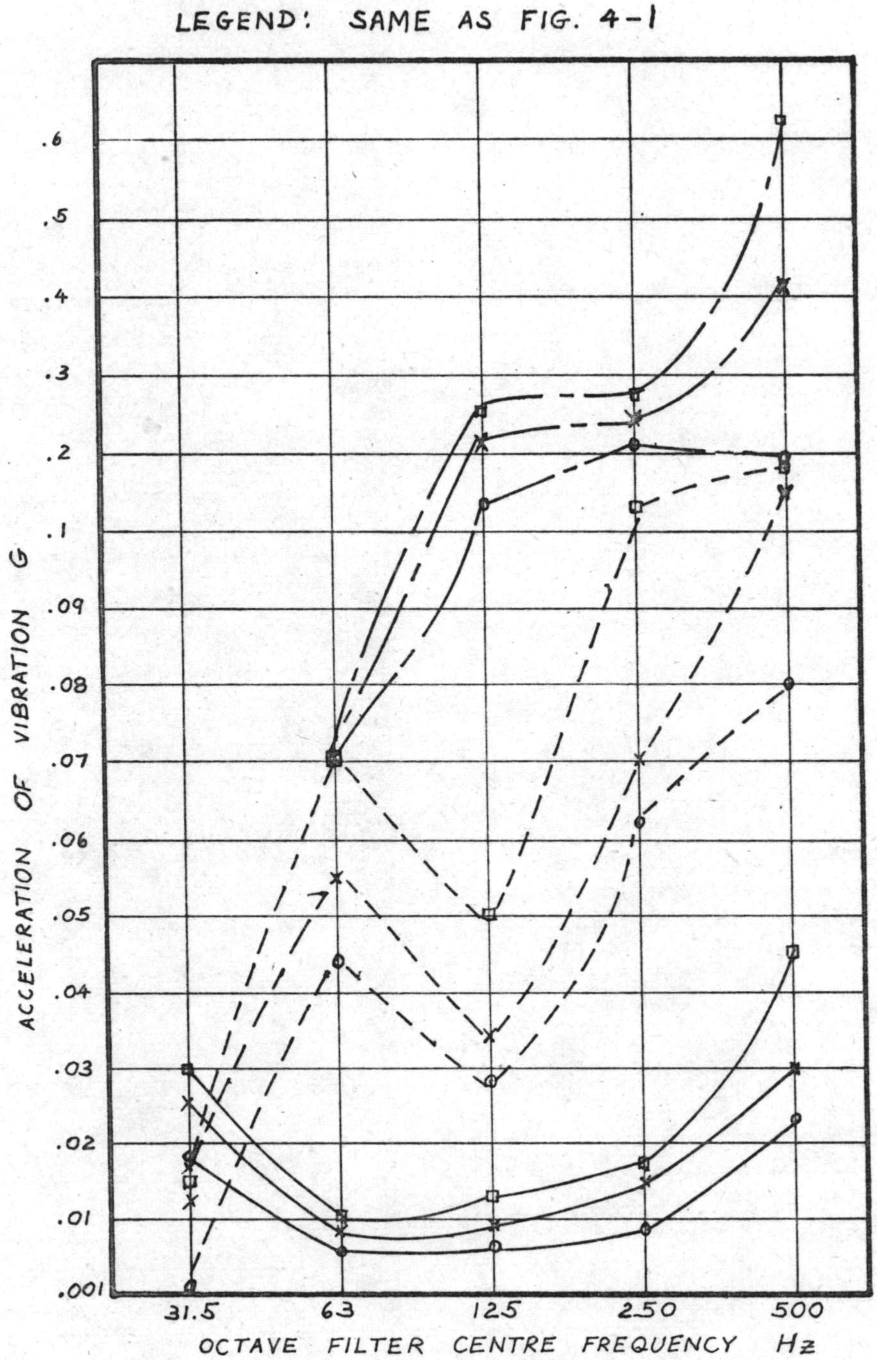


Fig.4-8 Acceleration of vibration at valve rocker cover plotted against octave filter centre frequency for balanced engine

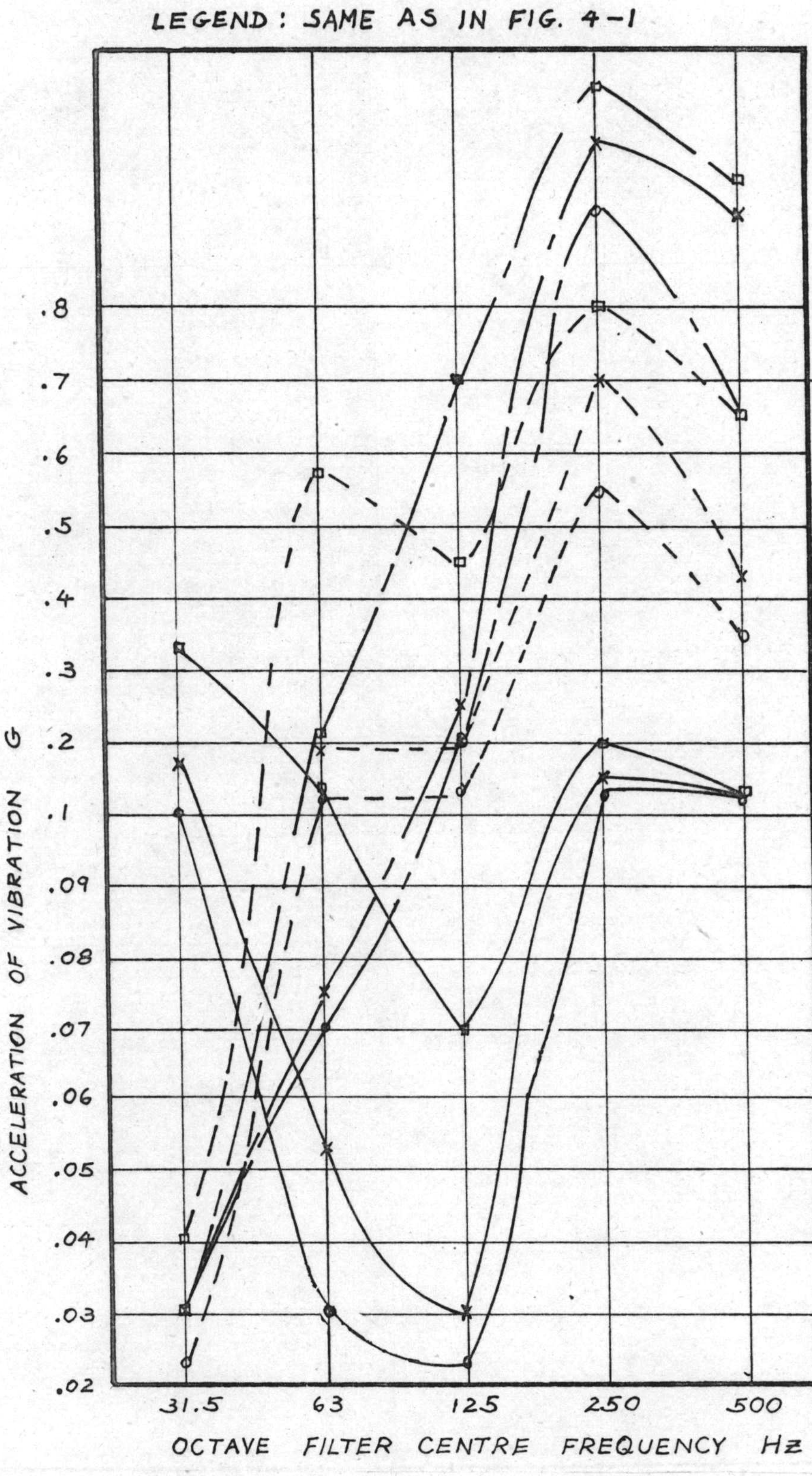


Fig.4-9 Acceleration of vibration at carburetor plotted against octave filter centre frequency for balanced engine

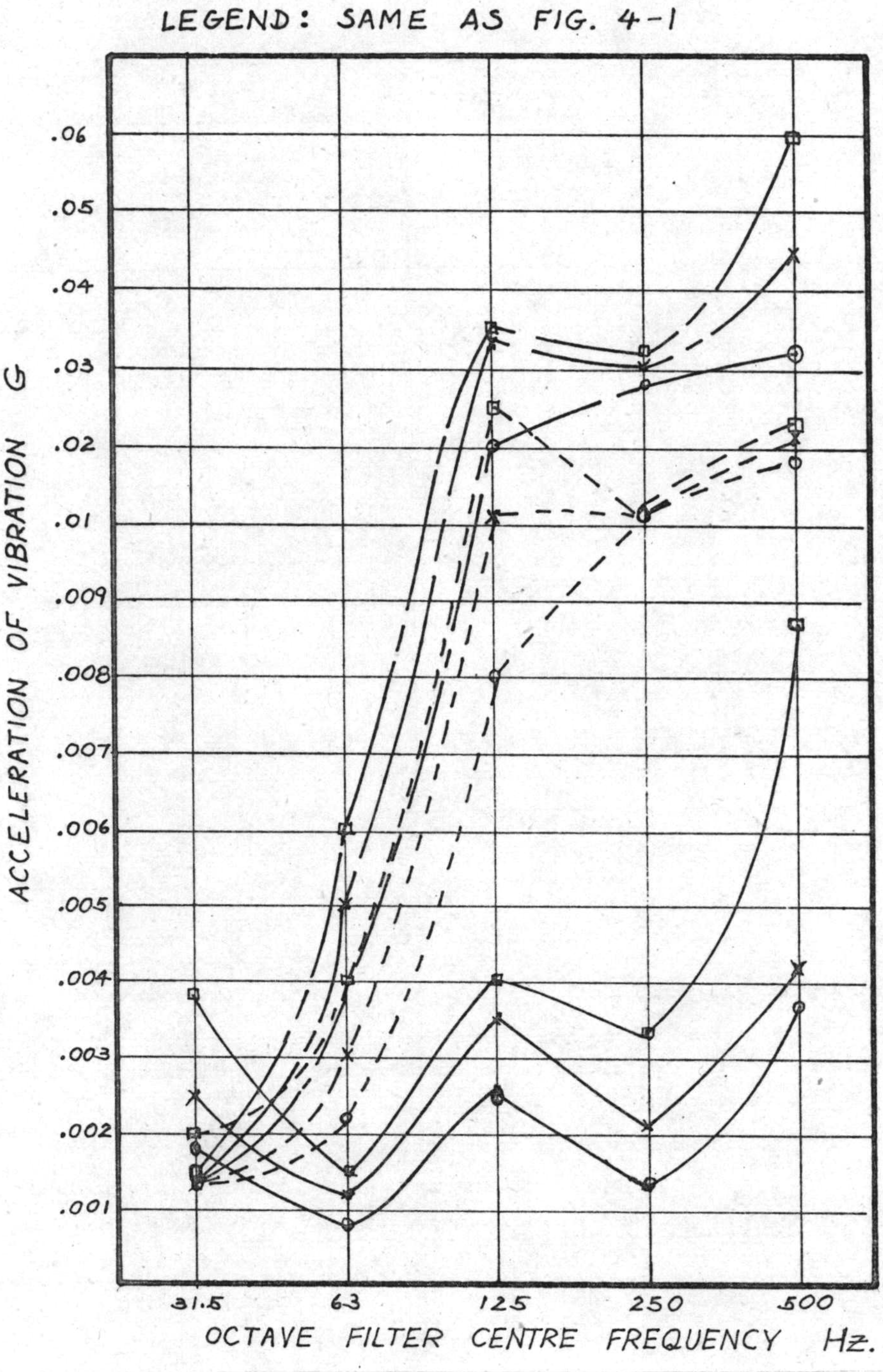


Fig.4-10 Acceleration of vibration at engine test bed plotted against octave filter centre frequency for balanced engine

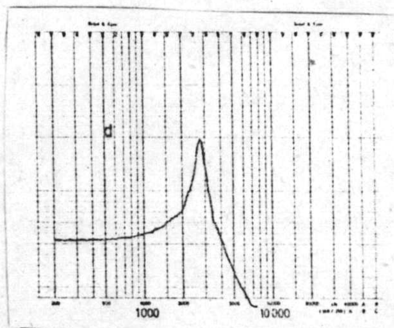


Fig.4-11   Frequency response of accelerometer with probe

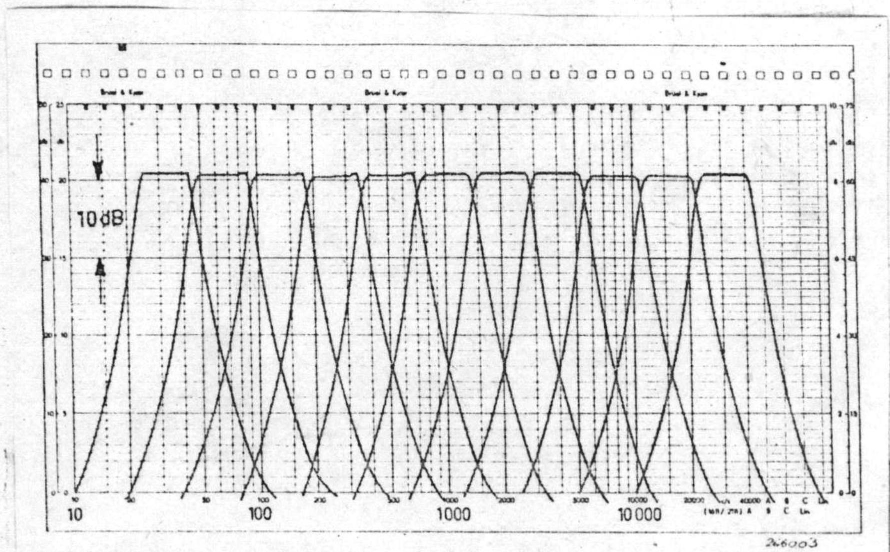


Fig.4-12   Frequency characteristics of octave filter

that as the speed increases, the acceleration of vibration also increases because the inertia force varies with the square of angular velocity. At the same speed grator load imposed on the engine yields higher level of vibration. Acceleration of vibration at different locations subjected to same exciting force is different depending on spring and damping effects of each components.

As shown previously in Fig. 4-1 to Fig. 4-10 and Fig. A-1 to Fig. A-2 that vibration levels worth of consideration due to significant magnitudes, as detected by octave filter with frequency characteristics shown in Fig. 4-12,[9] confine to first, second and fourth modes only. The higher order unbalances would be only small fractions of three modes mentioned.

Fig. 4-13 and Fig. 4-14 compare first, second and fourth mode unbalance forces of both engines at engine support brackets and carburetors. The comparison of vibration characteristics at varying speeds with no load condition indicates that engine with balancing shafts has smaller unbalance forces at first and second modes but higher at fourth mode.

Tables. 4-1 and 4-2 show acceleration of first, second and fourth modes for conventional and balancing engines respectively. Figures in parenthesis are percentages of first and fourth mode magnitudes compared to those of second mode.

According to the theory of vibration; 4-cycle engine having crank angle spacing at 0°-180°-180°-0°, its first mode vibration should be completely balanced but experimentally small vibration is detected . The existence of first mode but with small magnitudes ranging between 3 and 11 per cent of second mode for conventional engine is inevitable since in mass production all reciprocating parts are not exactly alike, they can differ within tolerance limits allowed by manufacturer, thus the force produced by each reciprocating part would not have the same magnitude and cannot be self balancing as predicted by theory which assumes all reciprocating masses are

OPERATING CONDITION : NO LOAD

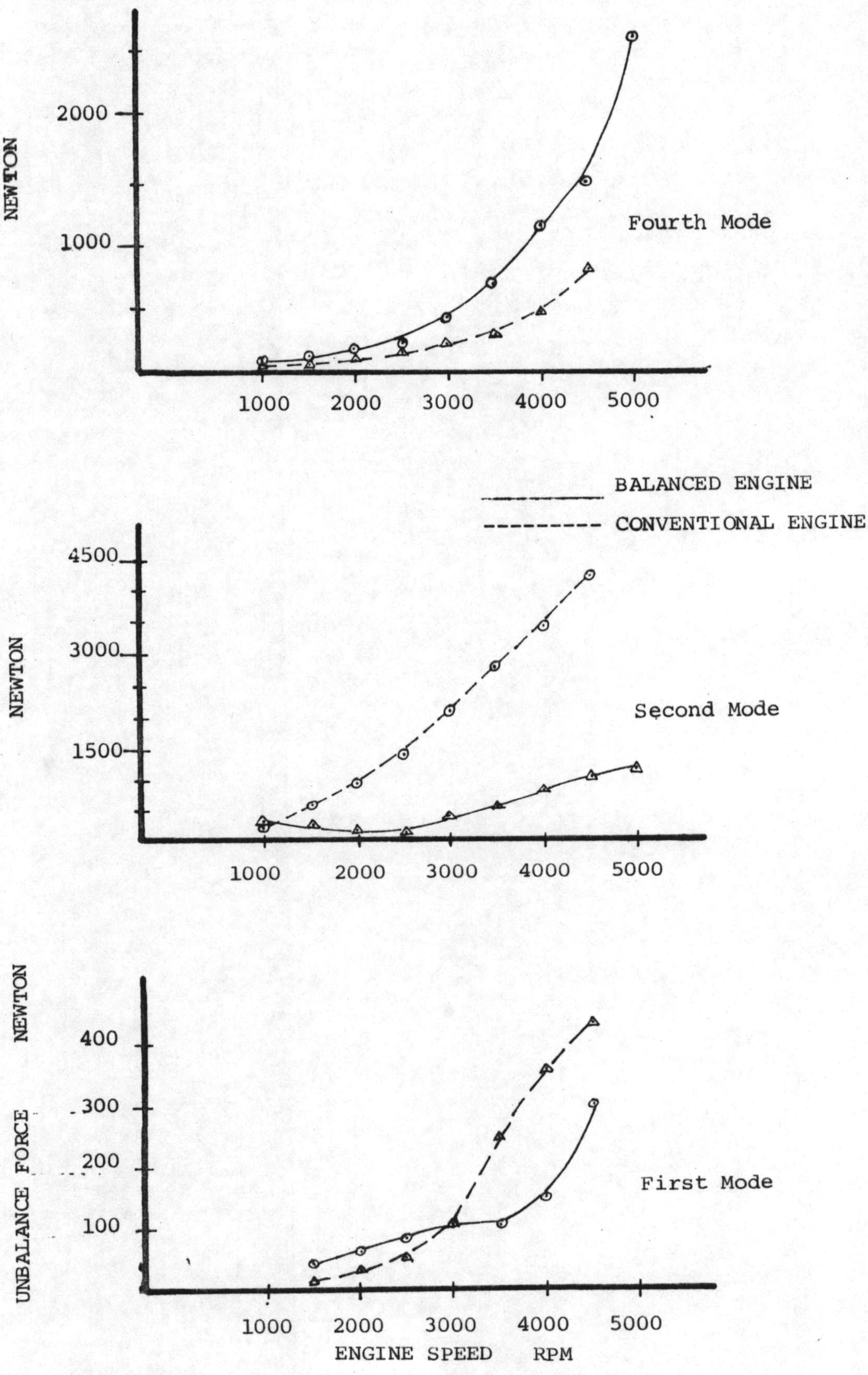


Fig. 4-13 Comparison of vibration characteristics at no load as measured at engine support bracket

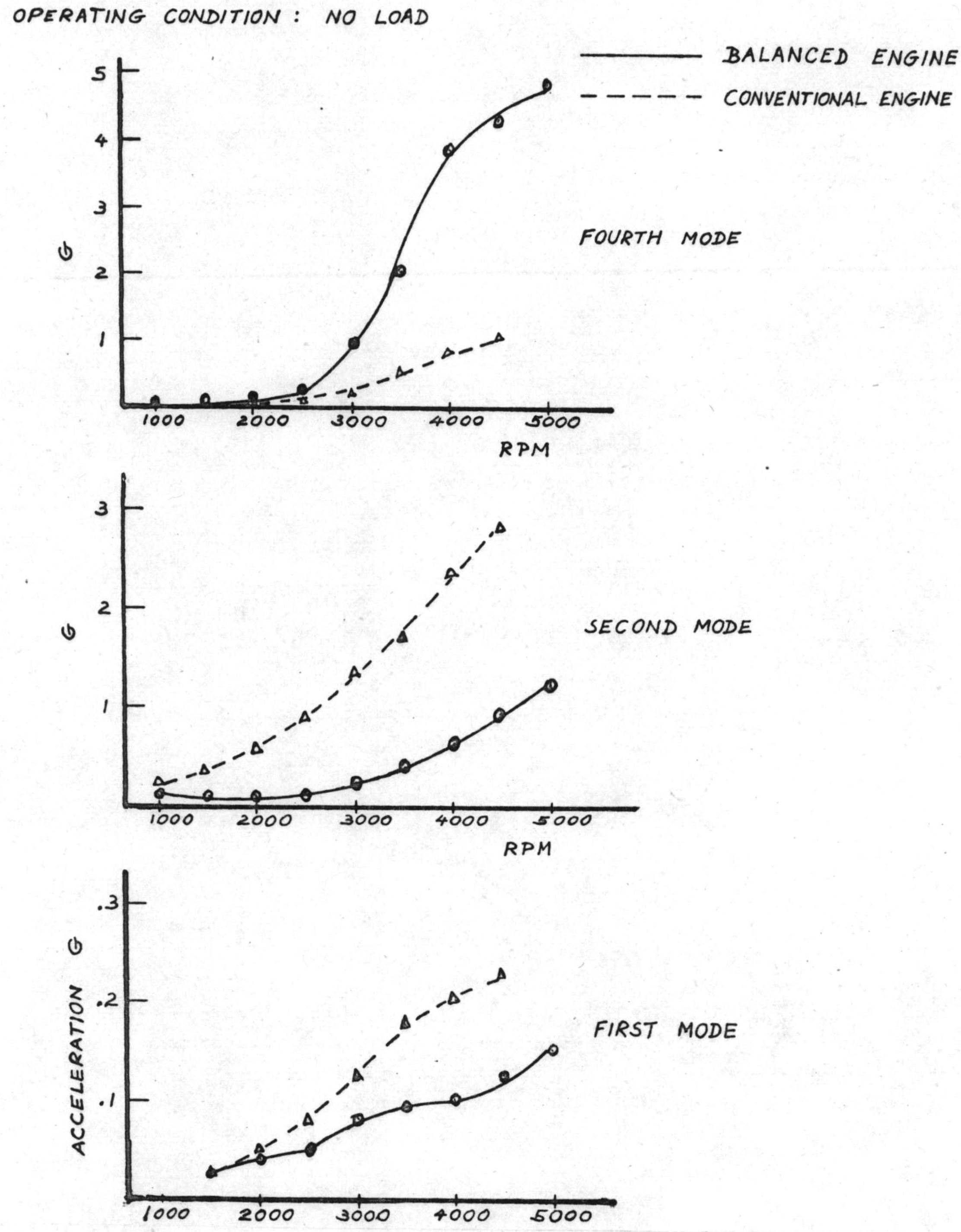


Fig.4-14 Comparison of vibration characteristics at no load as measured at engine carburetor

the same. However, first mode of balanced engine is lower than that of conventional engine possibly due to small tolerance allowed in manufacturing process and better selection of reciprocating parts so as to assemble almost identical parts to approach first mode balance.

RPM	First Mode	Second Mode	Fourth Mode
1000	-	.13	.03 (23.1%)
1500	.01 (3%)	.32	.03 (9.4%)
2000	.02 (3.7%)	.54	.05 (9.3%)
2500	.033 (4.2%)	.79	.08 (10.1%)
3000	.075 (6.4%)	1.17	.12 (10.3%)
3500	.14 (8.8%)	1.6	.16 (10%)
4000	.2 (10.5%)	1.9	.27 (14.2%)
4500	.24 (9.9%)	2.42	.45 (18.6%)

Table 4-1: Acceleration of Vibration at Engine Support Bracket for Conventional Engine in Unit of Acceleration g and Percentage of Second Mode Magnitude

For conventional engine, second mode vibration is predominant but considerably lower in balancing engine. Theoretically, second mode counter balancing shafts should eliminate second mode unbalance completely, however, in actual performance the design objective of the system is partly fullfilled by remarkably reducing its magnitude.

RPM	First Mode	Second Mode	Fourth Mode
1000	-	.13	.033 (25.4%)
1500	.028 (24.4%)	.115	.048 (41.7%)
2000	.04 (51.3%)	.078	.09 (115.3%)
2500	.045 (90.0%)	.05	.1 (200.0%)
3000	.06 (33.3%)	.18	.2 (111.0%)
3500	.065 (21.7%)	.3	.32 (106.7%)
4000	.07 (20.0%)	.35	.53 (151.4%)
4500	.09 (18.0%)	.5	.7 (140.0%)
5000	.15 (27.3%)	.55	1.2 (218.2%)

Table 4-2: Acceleration of Vibration at Engine Support Bracket for Balanced Engine in Unit of Acceleration g and Percentage of Second Mode Magnitude

Fourth mode vibration of balanced engine is larger than conventional engine as expected from theoretical formula given as:

For fourth mode $F_z = 4mr\omega^2 A_4 \cos 4\theta$

Maximum force at fourth mode $F_z = 4mr\omega^2 A_4$

Substituting appropriate values into above formula yields

For conventional engine $F_z(Th) = 3.57 \div 10^6\ \omega^2$ Newton

For balanced engine $F_z(Th) = 8.40 \div 10^6\ \omega^2$ Newton

Experimentally, fourth mode vibromotive force can be calculated from Newton Second Law F=ma where a is the acceleration of vibration at fourth mode. Knowing the masses of both engines results in two equations as follows:

For conventional engine $F_z(Exp) = 1803.27\ g$ Newton

For balanced engine $F_z(Exp) = 2163.92\ g$ Newton

Both pairs of formulae indicate fourth mode vibromotive force for balanced engine is bigger than that of conventional and this is connfirmed by experimental results.

As shown in Table 4-3, comparison between experimental and theoretical values of fourth mode vibration at no load for both engines shows that experimental values can range from 4 to 15 times of theoretical prediction. The factors which possibly contribute to these large discrepancies are:

(a) Mass m and crank radius r are assumed to be identical for 4-cylinders, the assumption that is unlikely to achieve in practice.

(b) The resolution capability of octave filter set cannot be set at any particular frequency but rather as a range of bandwidth as shown in Fig. 4-14,(9) Hence the inffuence of second mode will be detected along with fourth mode measurement . This problem is not critical if the mode to be measured is relatively large as compared to other modes but in this case theoretically fourth mode acceleration levels are rather small so the values are influenced greatly by second mode.

(c) The natural frequencies of the system may be in the region of forcing frequencies contributing to large discrepancies. For furthur investigation, the natural frequencies of damped spring-mass system should be determined.

	Conventional Engine			Balancing Engine		
RPM	F_z(Exp)	F_z(Th)	$\frac{F_z(Exp)}{F_z(Th)}$	F_z(Exp)	F_z(Th)	$\frac{F_z(Exp)}{F_z(Th)}$
1000	54.1	3.57	15.0	71.06	8.4	8.46
1500	54.1	8.04	6.7	103.87	18.9	5.49
2000	90.2	14.29	6.3	173.11	33.6	5.15
2500	144.26	22.32	6.5	216.39	52.5	4.12
3000	216.39	32.14	6.7	432.78	75.6	5.72
3500	288.52	43.75	6.6	692.45	102.9	6.73
4000	486.88	57.14	8.5	1146.88	134.4	8.53
4500	811.47	72.32	11.22	1514.74	170.1	8.90
5000	-	-	-	2596.70	210.0	12.37

Table 4-3: Comparison Between Experimental and Theoretical Values of Fourth Mode Vibromotive Forces at No Load

In Table 4-1, fourth mode magnitudes range from 9 to 20 per cent of second mode but theoretically it should be only 1.6 per cent. The errors arise as discussed above.

In Table 4-2, magnitudes in percentage of first and fourth modes compared to second mode values are relatively large since second mode are low in magnitudes because of counter balancing shafts.

Fig. 4-15 shows the performance curve at full throttle for conventional engine. Each characteristic has two curves, one with mixture closed to stoichiometric air/fuel ratio by weight and the other with mixture too rich. The latter one is presented here to make the performance curve more complete since the curve for correct mixture ratio cannot go below 2000 rpm. For both cases it is found experimentally that vibration characteristic agree closely.

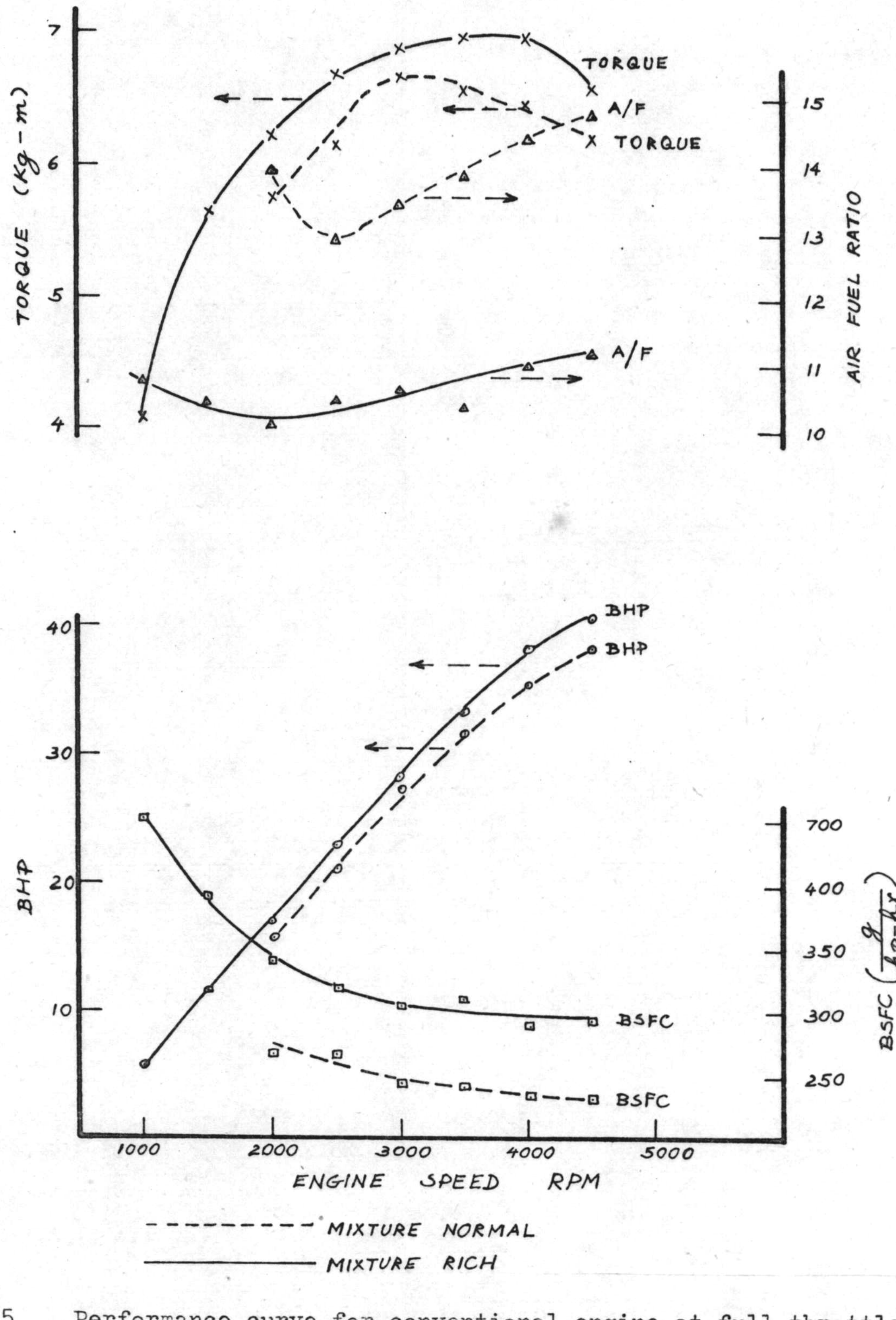


Fig.4-15 Performance curve for conventional engine at full throttle

Fig. 4-16 shows the performance curve at full throttle for balanced engine. According to its specifications published by the manufacturer[10] claim that engine testing based on DIN standard maximum power is 98 PS or 96.7 BHP at 5500 rpm and maximum torque is 15.3 kg-m at 3500 rpm while the experimental figures are 71 BHP at 4500 rpm and 14.5 kg-m at 2500 rpm. The discrepancies may due to additional equipment used in laboratory for measuring air inlet to carburetor causing greater drop of inlet manifold pressure and also additional exhaust pipe length to carry away the poisonuous exhaust from the laboratory causing greater exhaust manifold pressure. Both effects of decreasing inlet pressure and increasing exhaust pressure lower the power output[11] due to air flow rate into engine is less affecting the engine performance.

Comparing the performances of both engines indicate larger capacity engine has greater output in terms of power and torque which are good to meet the demand of greater load and acceleration pick up but it is at disadvantage from economical point of view since it consumes more fuel as shown by brake specific fuel consumption curves.

Fig. 4-17 shows characteristics of first and second mode vibration at full load for both engines measuring at engine support brackets. This figure is similar to vibration characteristic at no load in Fig. 4-12. However, at full load, the second mode acceleration of balanced engine is slightly higher than that of conventional engine in low speed range up to 2000 rpm. but beyond this speed the balanced engine exhibits remarkably low vibration.

Fig. 4-18 shows second mode vibromotive force versus engine speed. For conventional engine, experimental results agree closely to theoretical results up to 2500 rpm. as shown in curve A for no load condition and thereafter either no load (curve A) or full load values (curve B) vary within 10 per cent of theoretical values (dashed curve).

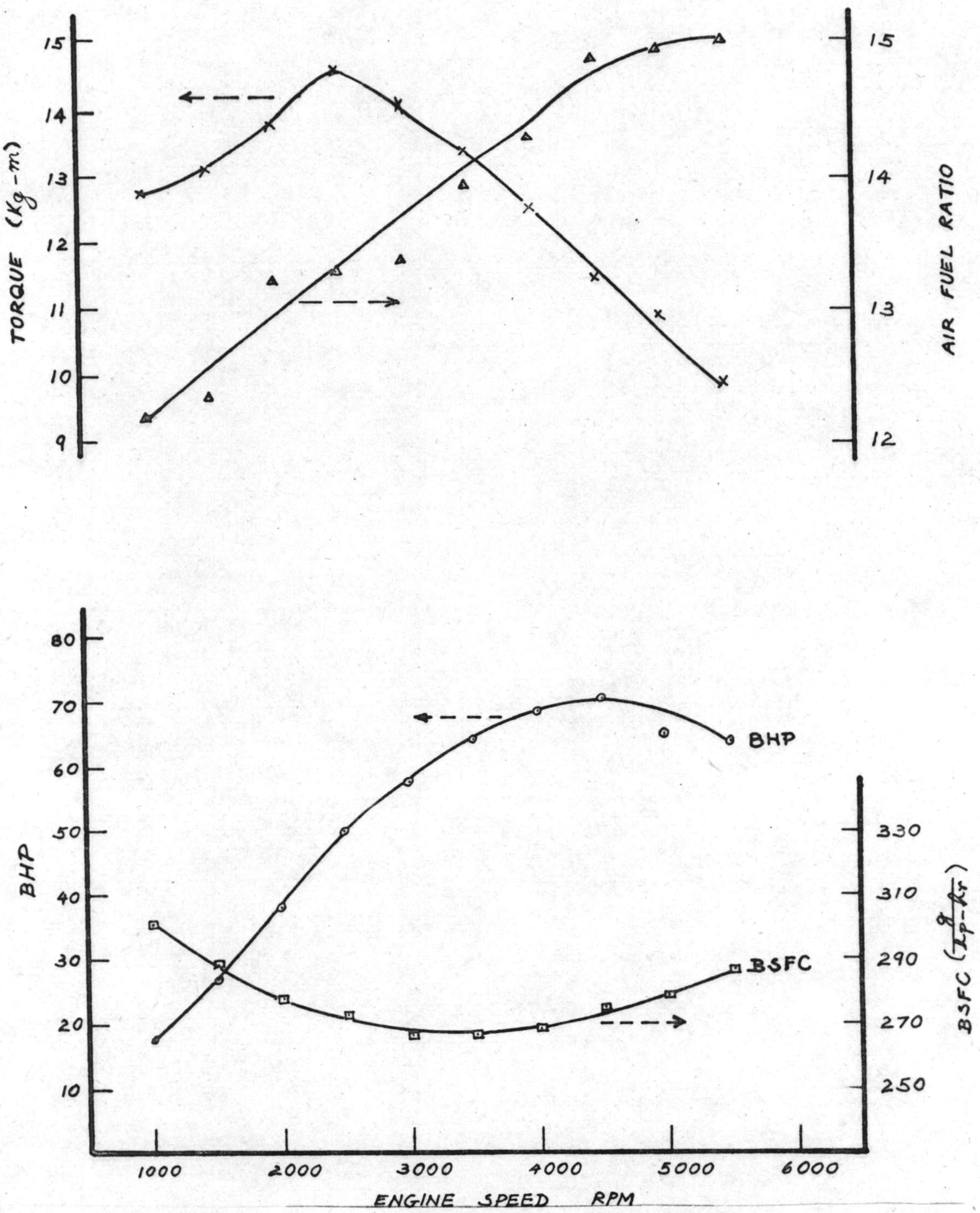


Fig.4-16 Performance curve for balanced engine at full throttle

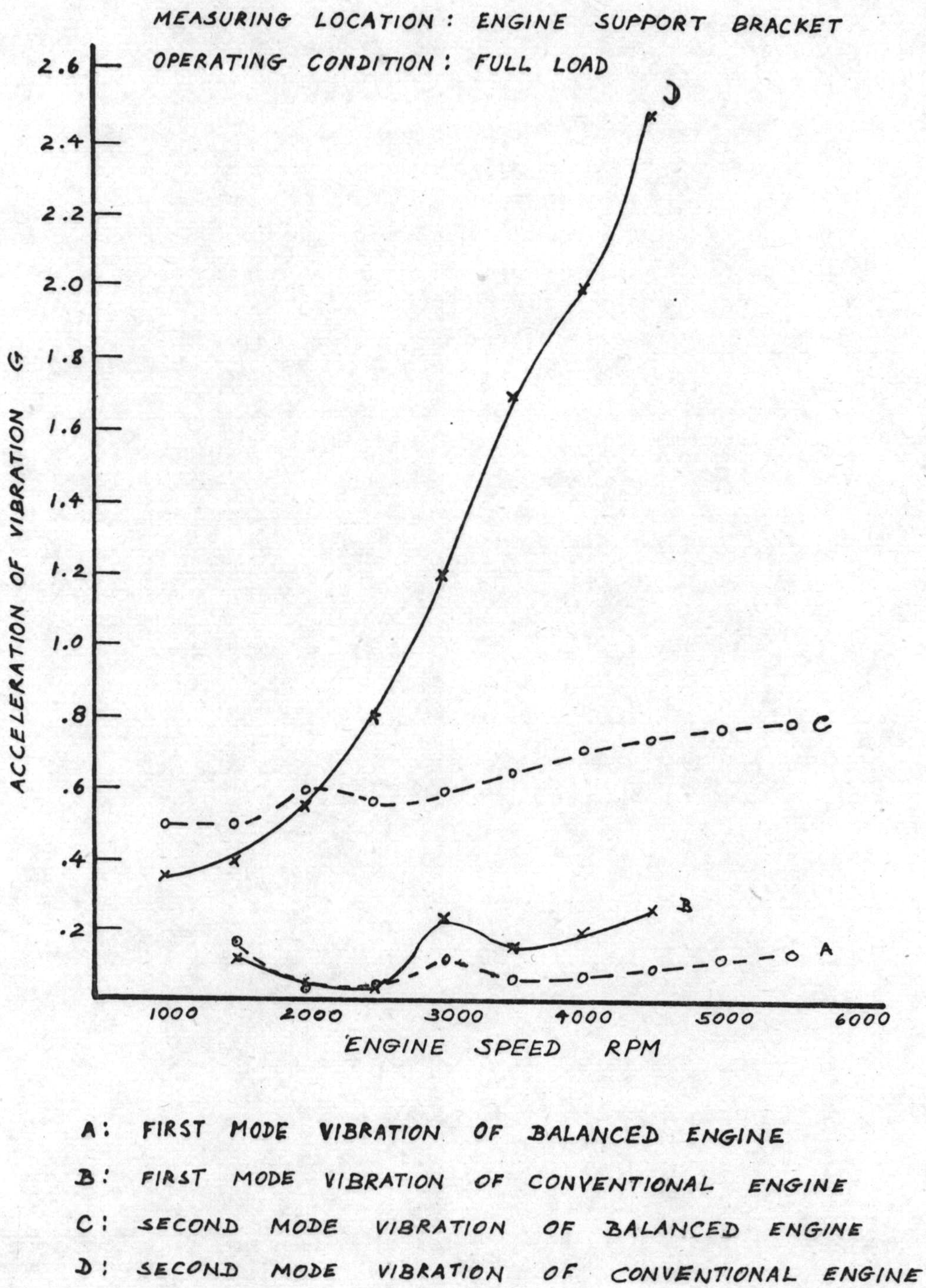
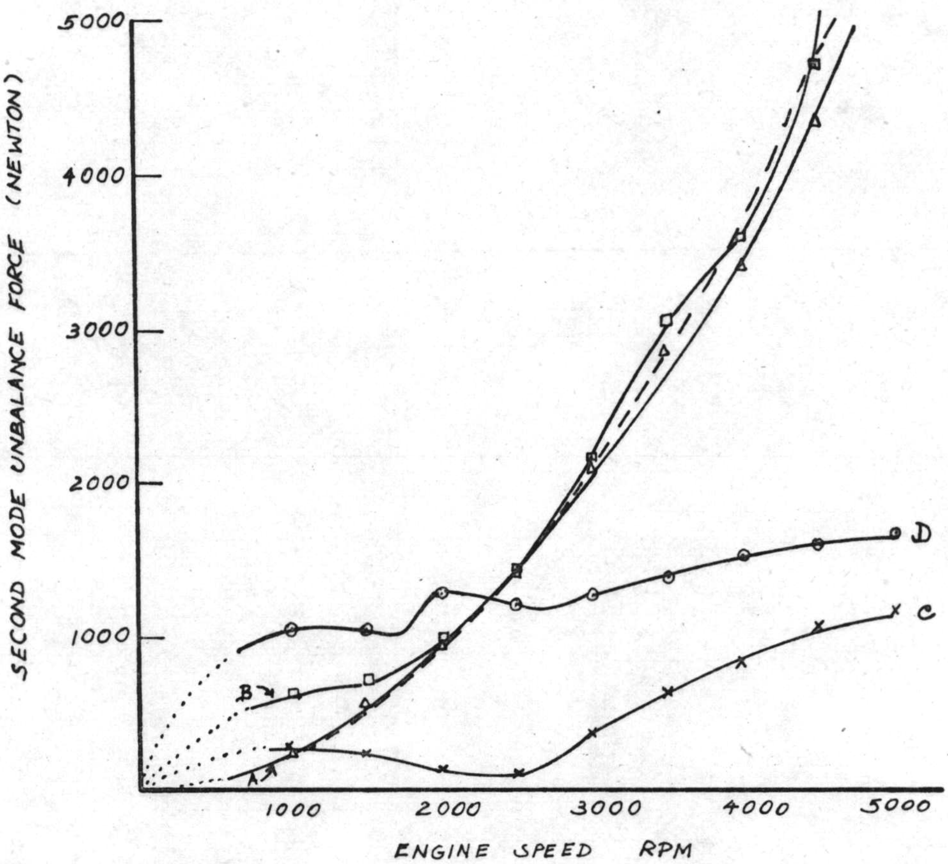


Fig.4-17 Comparison of vibration characteristics at full load as measured at engine support bracket



A : EXPERIMENTAL VALUES FOR CONVENTIONAL ENGINE AT NO LOAD

B : DITTO AT FULL LOAD

C : EXPERIMENTAL VALUES FOR BALANCED ENGINE AT NO LOAD

D : DITTO AT FULL LOAD

– – – – – THEORETICAL VALUES FOR CONVENTIONAL ENGINE

* : THEORETICAL VALUES FOR BALANCED ENGINE ARE NIL

Fig.4-18 Second mode vibration force plotted against engine speed

In the case of balanced engine, the theoretical values for second mode vibromotive force should be nil but experimentally the vibromotive force cannot be wholly eliminated. The magnitudes of vibromotive force at full load (curve D) are larger than at no load (curve C) probably due to larger gas force.