C.P. No. 34
12677
A.R.C. Technical Report

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Torsional Vibration Investigations on the Musketeer Engine

By

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Winifred Davidson, B.Sc.
and M. O. Wolfe, D.I.C.

LONDON: HIS MAJESTY'S STATIONERY OFFICE
1951
Price 2s. 6d. net.

C.P. No. 34

Report No. Structures 43

June, 1949

ROYAL AIRCRAFT ESTABLISHMENT

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SUMMARY

The torsional vibrational characteristics of the crankshaft of the Blackburn Musketeer Engine are investigated by experimental and theoretical analyses. Good agreement between theory and experiment is obtained for values of critical frequencies but not for stresses due to inadequate knowledge of the damping factors. The stresses are not considered to be sufficiently high to explain a crankshaft failure occurring in the course of development tests.

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1 Introduction

During the development of the Blackburn 'Musketeer' engine a failure of a crankshaft occurred. A theoretical vibration analysis made by the firm indicated that the failure might be attributed to a $4\frac{1}{2}$ engine order critical at 2,300 R.P.M.

The firm's calculations were examined and checked by R.A.E. and torsiograph experiments were carried out on the engine when running on the dynamometer test bed and when coupled to a 2-bladed 7'6" diameter Rotol wooden propeller. In each case repeat tests were made with an experimental rubber damper fitted to the crankshaft free end.

For the critical frequencies of the systems considered good agreement was obtained between theory and experiment, but a large discrepancy was found between the calculated and observed maximum stresses, which is due to our present inadequate knowledge of damping in engines and propellers.

The greatest measured crankshaft stress was found to be ± 2.2 tons/sq.in. and was produced by a $4\frac{1}{2}$ engine order critical at 2,000 R.P.M. This is not considered to be excessive and the crankshaft failures are not attributed to it. It is not therefore recommended that a damper should be fitted.

2 Engine Details

The musketeer engine is of the inverted in line type normally aspirated, operating on the four stroke cycle. There are six cylinders 4.8" bore and $5\frac{1}{2}$" stroke, the firing order is 1, 5, 3, 6, 2, 4. The maximum B.H.P. is 263 at 2,500 R.P.M.

3 Details of Crankshaft Failure

The failure of the crankshaft occurred after about 10 hours running and took place in the fillet between the centre main bearing and the adjacent crankweb towards the propeller shaft end.

4 Details of Tests

Two sets of tests were carried out, one with the engine mounted on a dynamometer test bed and one with the engine mounted on a propeller test stand and fitted with a Rotol 2-blade 7'6" wooden propeller.

In each series torsional vibrations were measured on the standard engine and also with a rubber vibration damper fitted to the free end of the crankshaft.

Vibration records were taken at 50 R.P.M. intervals in the range 1,800 to 2,500 R.P.M. during power curve tests on the engine and also during throttle curve tests in the range 800 to 1,800 R.P.M. Further records were taken on the dynamometer test bed during a constant boost curve test at -2 lb/sq.in. boost pressure in the range 1,800 to 2,300 R.P.M.

The Sperry torsiograph and four channel recording equipment were used throughout the tests. Fifty cycle timing marks were obtained from a transformer connected to the A.C. mains and engine revolution marks were also recorded.

5 Experimental Results

Figs.3 to 12 have been plotted showing the predominant components of torsional vibration for the various tests. Table I shows the main resonant speeds and amplitudes. The maximum crankshaft stresses have been calculated using measured amplitudes and calculated natural frequencies and are as follow:-

5.1 Standard engine running on the dynamometer test bed

The maximum crankshaft stress is due to a $4\frac{1}{2}$ engine order resonance at 2,000 R.P.M. which has an amplitude of ± 0.7 degrees at the crankshaft free end (Fig.4). The calculated stress due to this resonance is ± 2.2 tons/sq.in.

5.2 Engine fitted with damper running on the dynamometer test bed

The maximum stress is again found to be due to a $4\frac{1}{2}$ engine order resonance at 2,190 R.P.M. of amplitude ± 0.24 degrees (Fig.7). This gives a reduced crankshaft stress of ± 0.64 tons/sq.in.

5.3 Standard engine fitted with Rotol propeller

The $4\frac{1}{2}$ engine order resonance is again the greatest having an amplitude of $\pm 0.83^{\circ}$ at 2,020 R.P.M. (Fig.10). The maximum crankshaft stress due to this order is 2.1 tons/sq.in. and the propeller shaft stress is 1.8 tons/sq.in.

Stresses due to the $3\frac{1}{2}$ engine order at 2,500 R.P.M. are approximately the same as for the $4\frac{1}{2}$ engine order and stresses due to the 4th engine order at 2,160 R.P.M. are slightly less.

5.4 Engine fitted with damper and Rotol propeller

The $1\frac{1}{2}$ engine order resonance at 2,500 R.P.M. gives the maximum stresses having an amplitude of 1.9° at the crankshaft free end (Fig.12). The crankshaft stress due to this order is ± 1.0 tons/sq.in. and the propeller shaft stress is ± 0.9 tons/sq.in.

6 Theoretical Results

Since the crankshaft failure occurred when the engine was running on a test bed coupled to a Heenan and Froude type hydraulic dynamometer by means of a Hardy Spicer flexible coupling, it was first necessary to consider the vibrations of the dynamic system appropriate to these conditions. The vibrations of the coupled engine-propeller dynamic system were then considered, taking into account the flexibility of the propeller blades, for the actual propeller to be used on the engine.

The dynamic systems appropriate to the engine coupled to the dynamometer and the engine coupled to the Rotol propeller are shown in Figs.1 and 2. The calculated displacement curves are also shown for the second overtone natural frequency in the case of the engine coupled to the dynamometer and for the fundamental frequency in the case of the engine coupled to the Rotol propeller.

Messrs. Blackburn Aircraft Co. had already made calculations on the engine and these were examined and checked by the R.A.E.

<u>Firing order 153624</u>

Maximum torsional vibration stress in the propeller shaft = ± 5.3 tons/sq.in.

Maximum torsional vibration stress in the crankshaft = ± 6.35 tons/sq.in.

These stresses are produced by a $4\frac{1}{2}$ engine order vibration at 2,170 R.P.M.

<u>Firing order 124653</u>

Maximum torsional vibration stress in the propeller shaft = ± 1.03 tons/sq.in.

Maximum torsional vibration stress in the crankshaft = ± 1.2 tons/sq.in.

These stresses are produced by the 4th engine order vibration at 2,440 R.P.M.

It was pointed out that the above calculations gave no indication of the behaviour of the engine on the test bed, neither did they give an accurate picture of the torsional vibrations of the engine-propeller combination, for which it is necessary to take into account the flexibility of the propeller.

Accordingly, Messrs. Blackburns carried out calculations on the torsional vibrations of the engine on the test bed and submitted them to the R.A.E.

R.A.E. found that the stiffness of the Hardy Spicer coupling was incorrect and should have been 0.0554×10^6 lb/ins/radian instead of 0.2216×10^6 lb.ins/radian. This correction altered the fundamental and first overtone frequencies considerably but had very little effect upon the second overtone and higher frequencies. As the important criticals are due to the second overtone frequency, the results obtained by Messrs. Blackburns are correct.

With firing order 153624 the maximum calculated torsional vibration stress in the crankshaft is ± 7.1 tons/sq.in. and arises from the $4\frac{1}{2}$ engine order vibration at 2,250 R.P.M. Allowing for a stress concentration at the fillet between the crankpin and crankweb, it was thought that this stress might have caused the crankshaft failure in the engine. When the firing order is changed to 124653 there are no dangerous criticals in the speed range of the engine.

Messrs. Blackburns also made further calculations on the engine propeller system, this time taking into account the flexibility of the propeller. The propeller frequency admittance curves were supplied by Messrs. Rotols. The calculations of the critical speeds were quite correct but in finding the vibration stresses and torques a dynamic magnifier of ten was used. According to Wilson[1], R.A.E. estimated that the engine dynamic magnifiers should be 15.1, 26.6 and 37.3 for the three modes of vibration respectively.

The stress in the propeller shaft due to the $4\frac{1}{2}$ engine order vibration at 2,290 R.P.M. was therefore increased from ± 1.66 tons/sq.in. to ± 6.2 tons/sq.in. The maximum stress in the crankshaft is slightly greater than that in the propeller shaft.

7 Comparison of theoretical and experimental Results

A comparison of the theoretical and experimental results shows that a fair agreement has been obtained for the resonant frequencies of the dynamic systems considered, but a large discrepancy exists between the calculated and observed maximum stresses in the crankshaft.

For instance in the case of the standard engine running on a dynamometer test bed the maximum calculated stress is ± 7.1 tons/sq.in. whereas the maximum measured stress is ± 2.2 tons/sq.in.

Similarly for the standard engine fitted with the Rotol propeller the maximum calculated stress in the crankshaft is ± 6.35 tons/sq.in. and the maximum measured stress is ± 2.1 tons/sq.in.

These discrepancies arise from an inadequate knowledge of the relevant damping coefficients. In the case of the engine coupled to a dynamometer, the damping in the latter must be considered in addition to the crankshaft damping. The amplitude at the dynamometer varies considerably for small changes in frequency and a correspondingly large variation in the dynamic magnifier takes place. It is not to be expected therefore that good agreement between theoretical and experimental results can be obtained for this condition.

Considering the case of the engine running with a propeller the damping consists mainly of propeller and crankshaft damping. Attempts have been made to determine values for P_p, the propeller damping coefficient, for various propellers but no satisfactory values have yet been deduced. The value of h_c the damping coefficient for the crankshaft depends upon engine design and size. From torsiograph observations on various engines the following empirical relationship has been suggested by Carter[2] for its evaluation:-

$$h_c = E_c \left(\frac{I_c}{386}\right)^{\frac{4}{5}} \text{ lb. in. per radian per sec.}$$

where I_c is the polar moment of inertia per crank in pounds inches squared, and E_c is a coefficient for which values have been deduced for several engine types.

Carter gives values of E_c varying from 20 to 360 based on observations made on seven different aero engines, and also quotes evidence which suggests that E_c can vary considerably with the value of mean torque.

In this case experience on similar engines suggested 20 as being a value for E_c appropriate to this engine. For a mode of vibration with a node at the propeller hub the damping contributed by the propeller is small in comparison with that contributed by the crankshaft. For the third mode of vibration therefore, considering only crankshaft damping, the dynamic magnifier is 33.2, and this compares with 12.6 which is the dynamic magnifier calculated from the observed amplitudes for this mode.

It is apparent that with the present state of knowledge on damping influences in engines and propellers, it is impossible to make calculations for the amplitude of vibration at a natural frequency of a coupled engine propeller system to a sufficient standard of accuracy.

The present method adopted by some engine manufacturers is to make use of empirical overall factors based on past experience with similar designs. This is not altogether satisfactory, particularly where an engine of a radically different design is considered. There is therefore a great need for further experimental work to be done on engine and propeller damping making the fullest use of modern techniques of measurement.

In the meantime the type testing of prototype engines should include a series of torsiograph tests with approved instruments, and these tests should be conducted with the engine coupled to an actual flight propeller of the same type as that on which it is ultimately intended to operate. Admittance curves should also be provided for engine and propeller to facilitate the initial matching of these items and the final interpretation of the test results. The ultimate criteria of acceptance should be based on allowable maximum vibration stress levels in the component parts of the system as derived from the foregoing tests.

8 Conclusions

The greatest measured crankshaft stress due to torsional vibration in the Musketeer engine is ± 2.2 tons/sq.in and is produced by a 4½ engine order critical at 2,000 R.P.M. This is not considered to be excessive and the crankshaft failures cannot therefore be attributed to this source; for this reason it is not recommended that a damper should be fitted. Attention is drawn to the inadequacy of our present knowledge of damping influence in engines and propellers and the need for further work on this aspect of engine vibration is emphasised.

References

No.	Author	Title etc.
1	Ker Wilson	Practical Solution of Torsional Vibration Problems.
2	B.C. Carter	Torsional Vibration in Aircraft Power Plants: Methods of Calculation. RAE Report E.3586 Part II.

TABLE I

Predominant Resonance Frequencies Measured on the Cirrus Musketeer Engine

Standard engine running on dynamometer test bed

Engine running conditions	Engine order of vibration	Measured Resonance speed R.P.M.	Frequency at Resonance C.P.S.	Amplitude at free end of crankshaft (degrees)
Power curve	$3\frac{1}{2}$	2,500	146	± 0.4
Power curve	$4\frac{1}{2}$	2,000	150	± 0.7
C.B. curve	$4\frac{1}{2}$	2,010	151	± 0.67
Throttle curve	$5\frac{1}{2}$	1,590	146	± 0.3
" "	6	1,520	152	± 0.6
" "	$7\frac{1}{2}$	1,280	160	± 0.5

Vibration damper fitted to free end of crankshaft engine running on dynamometer test bed

Engine running conditions	Engine order of vibration	Measured Resonance speed R.P.M.	Frequency at Resonance C.P.S.	Amplitude at free end of crankshaft (degrees)
Power curve	$1\frac{1}{2}$	2,140	53	± 1.22
C.B. curve	$1\frac{1}{2}$	2,200	55	± 0.84
Throttle curve	3	1,180	59	± 0.4
" "	$3\frac{1}{2}$	980	57	± 0.2
C.B. curve	$4\frac{1}{2}$	1,960	147	± 0.15
Power curve	$4\frac{1}{2}$	1,930	145	± 0.24

Calculated natural frequencies for engine dynamometer system:- 51 and 163 C.P.S.

Standard engine fitted with propeller

Engine running conditions	Engine order of vibration	Measured Resonance speed R.P.M.	Frequency at Resonance C.P.S.	Amplitude at free end of crankshaft (degrees)
Power curve	$1\frac{1}{2}$	1,850	46	± 0.5
Throttle curve	$1\frac{1}{2}$	2,050	51	± 0.34
Power curve	$3\frac{1}{2}$	2,500	146	± 0.75
" "	4	2,160	144	± 0.52
Throttle curve	4	2,130	142	± 0.76
Power curve	$4\frac{1}{2}$	2,040	153	± 0.75
Throttle curve	$4\frac{1}{2}$	2,020	151	± 0.83
" "	$5\frac{1}{2}$	1,530	140	± 0.2
" "	6	1,520	152	± 0.43
" "	$7\frac{1}{2}$	1,180	148	± 0.15

Engine fitted with propeller
Vibration damper fitted to free end of crankshaft

Engine running conditions	Engine order of vibration	Measured Resonance speed R.P.M.	Frequency at Resonance C.P.S.	Amplitude at free end of crankshaft (degrees)
Throttle curve	$1\frac{1}{2}$	2,500	63	± 1.9
" "	$2\frac{1}{2}$	1,600	67	± 0.6
" "	3	1,360	68	± 0.85
" "	$3\frac{1}{2}$	1,160	68	± 0.55

Calculated natural frequencies for engine propeller system:- 62.5 and 168 C.P.S.

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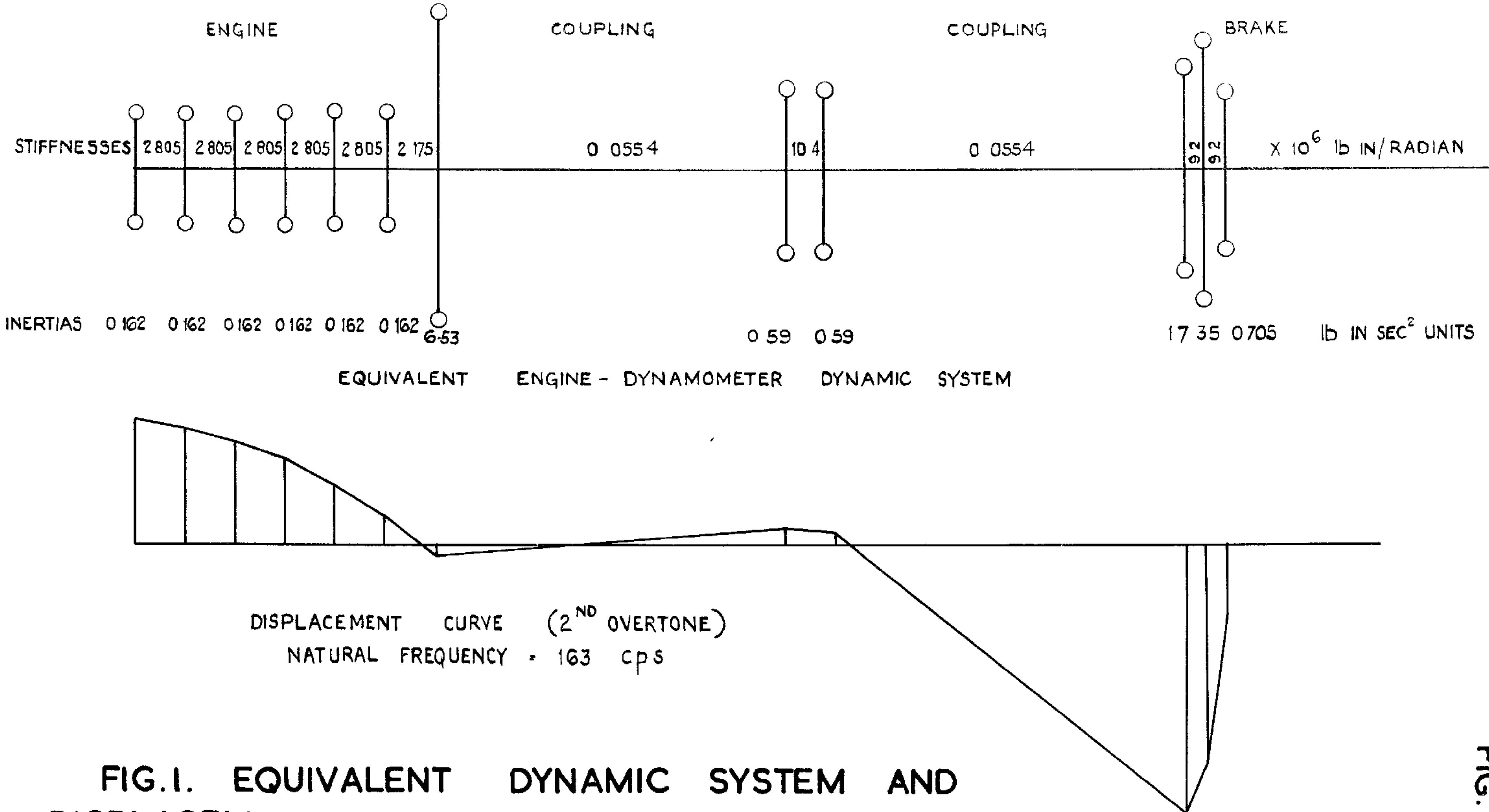


FIG. 1. EQUIVALENT DYNAMIC SYSTEM AND DISPLACEMENT CURVE FOR MUSKETEER ENGINE ON DYNAMOMETER TEST BED.

FIG. 1.

FIG. 2.

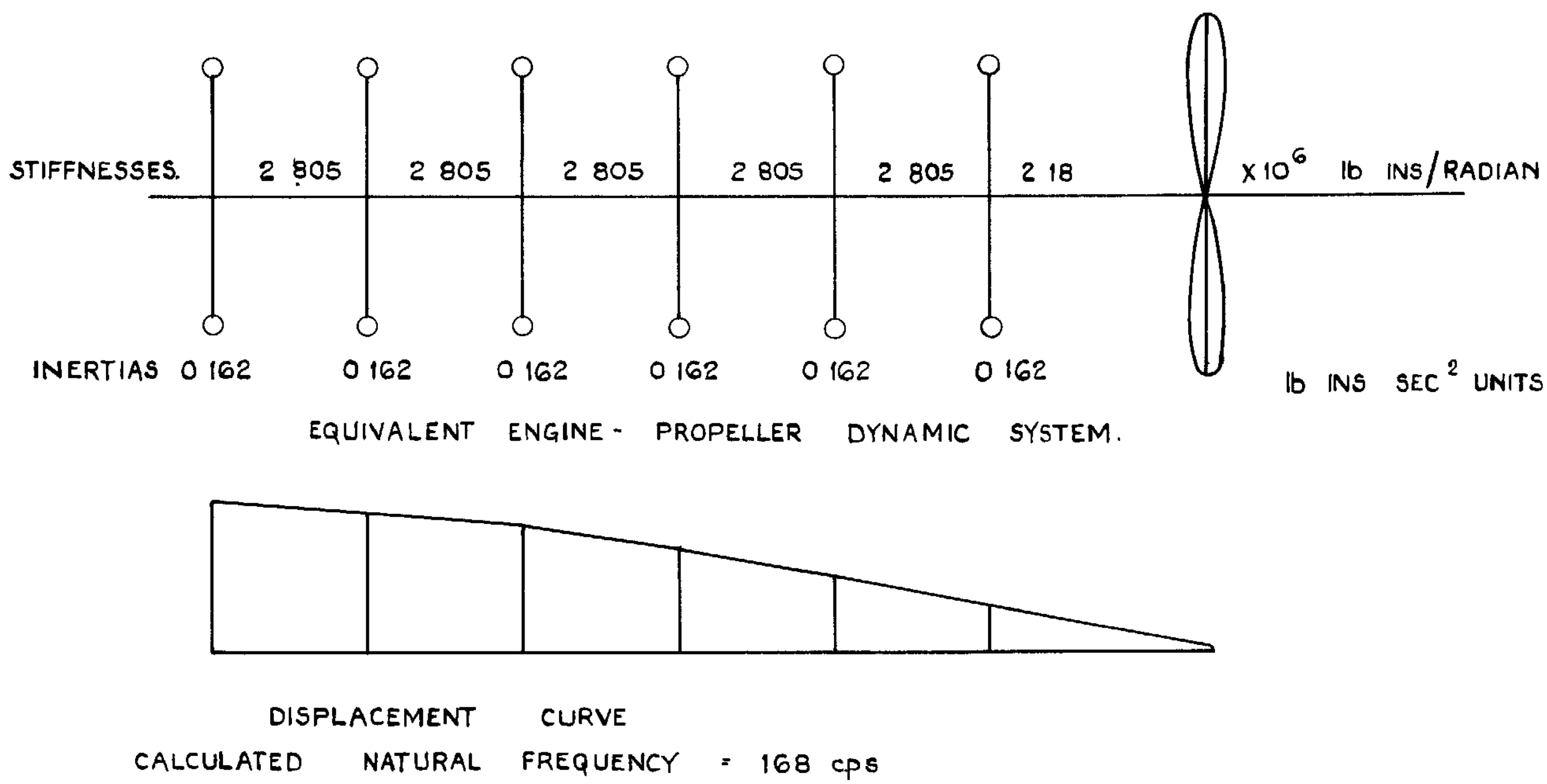
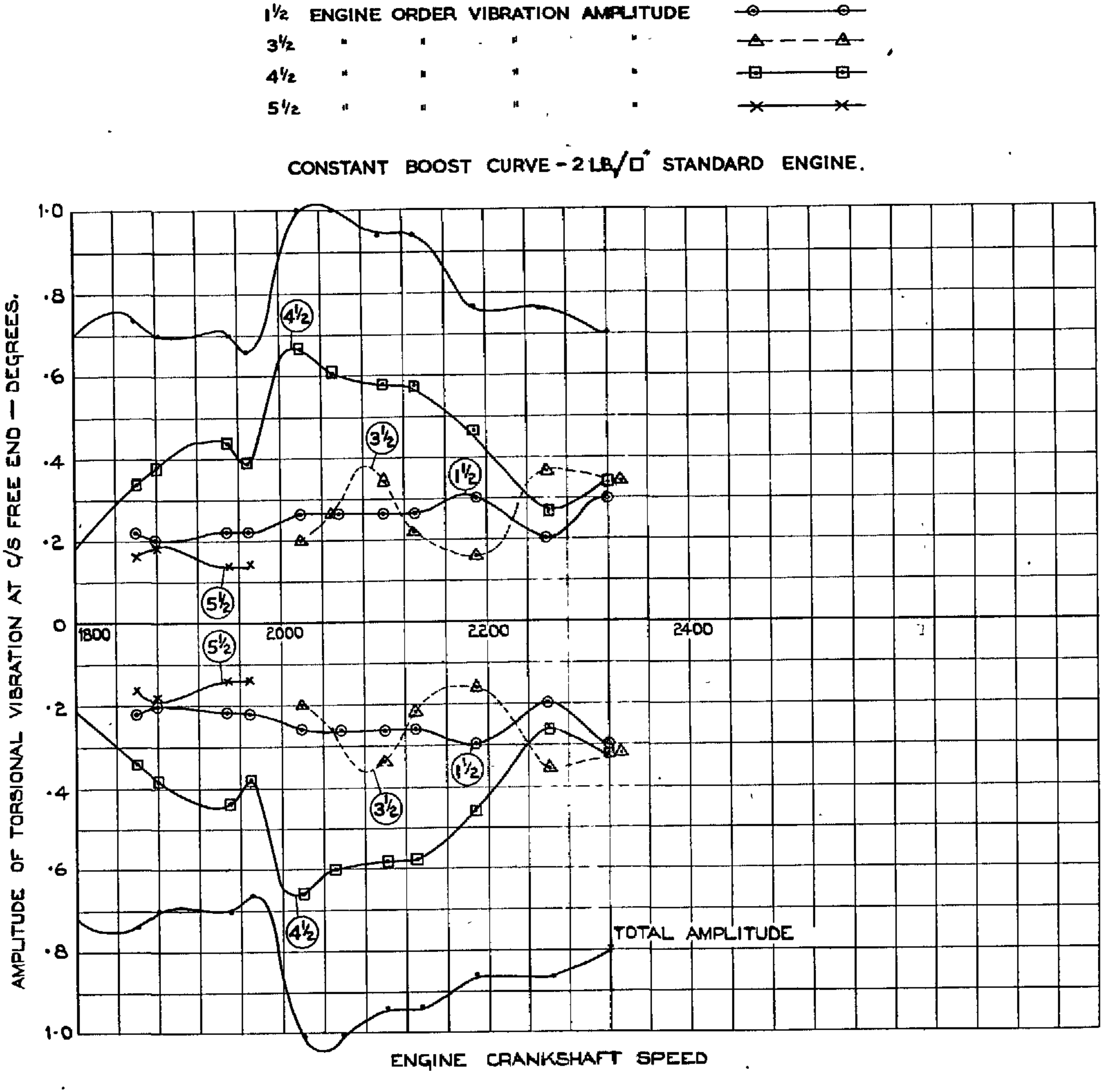


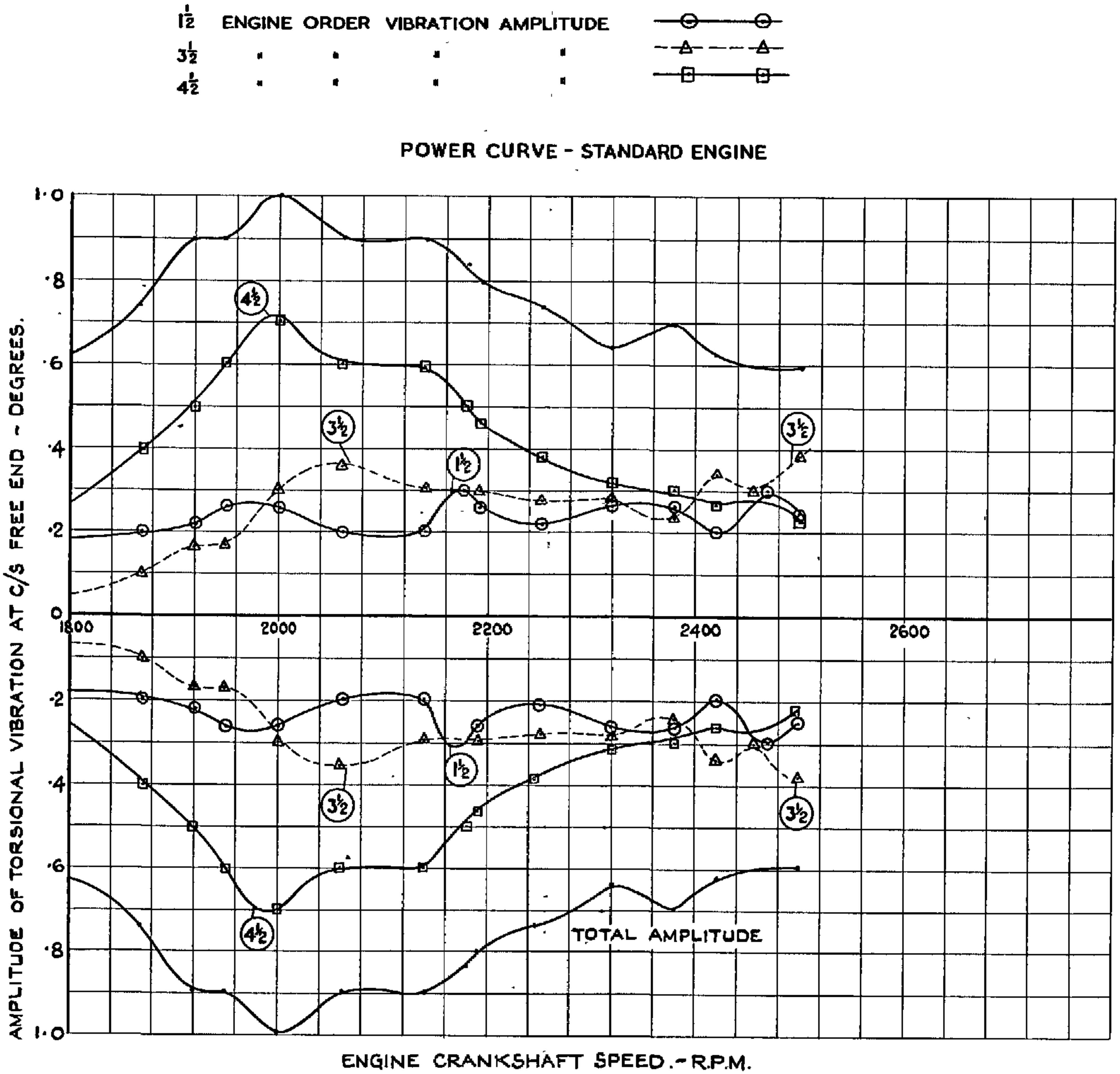
FIG. 2. EQUIVALENT DYNAMIC SYSTEM AND DISPLACEMENT CURVE FOR MUSKETEER ENGINE WITH ROTOL PROPELLER.

FIG. 3.



PREDOMINANT COMPONENTS OF TORSIONAL VIBRATION OF THE MUSKETEER ENGINE

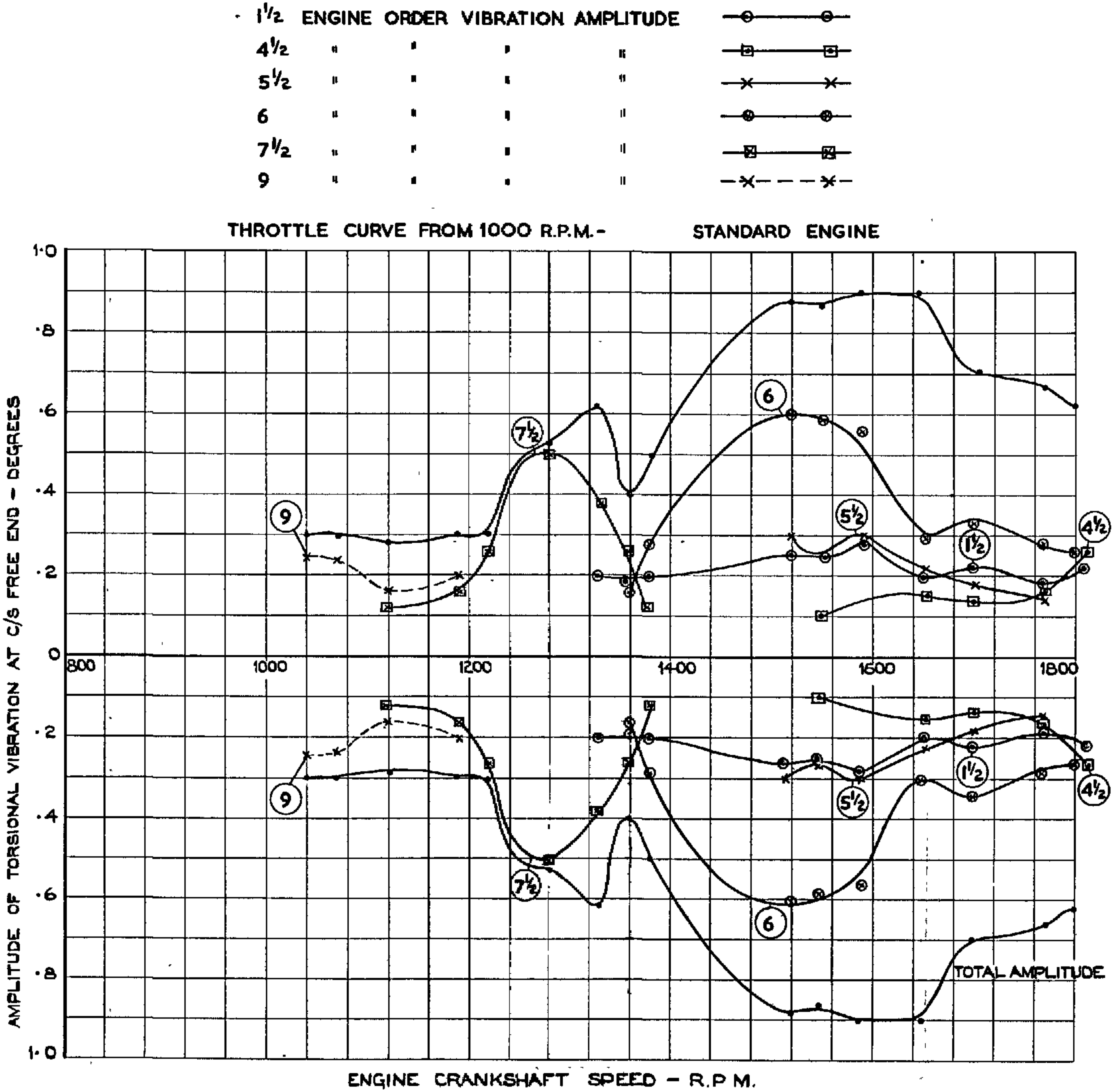
FIG. 3. MUSKETEER ENGINE RUNNING ON DYNAMOMETER TEST BED.



PREDOMINANT COMPONENTS OF TORSIONAL VIBRATION OF THE MUSKETEER ENGINE.

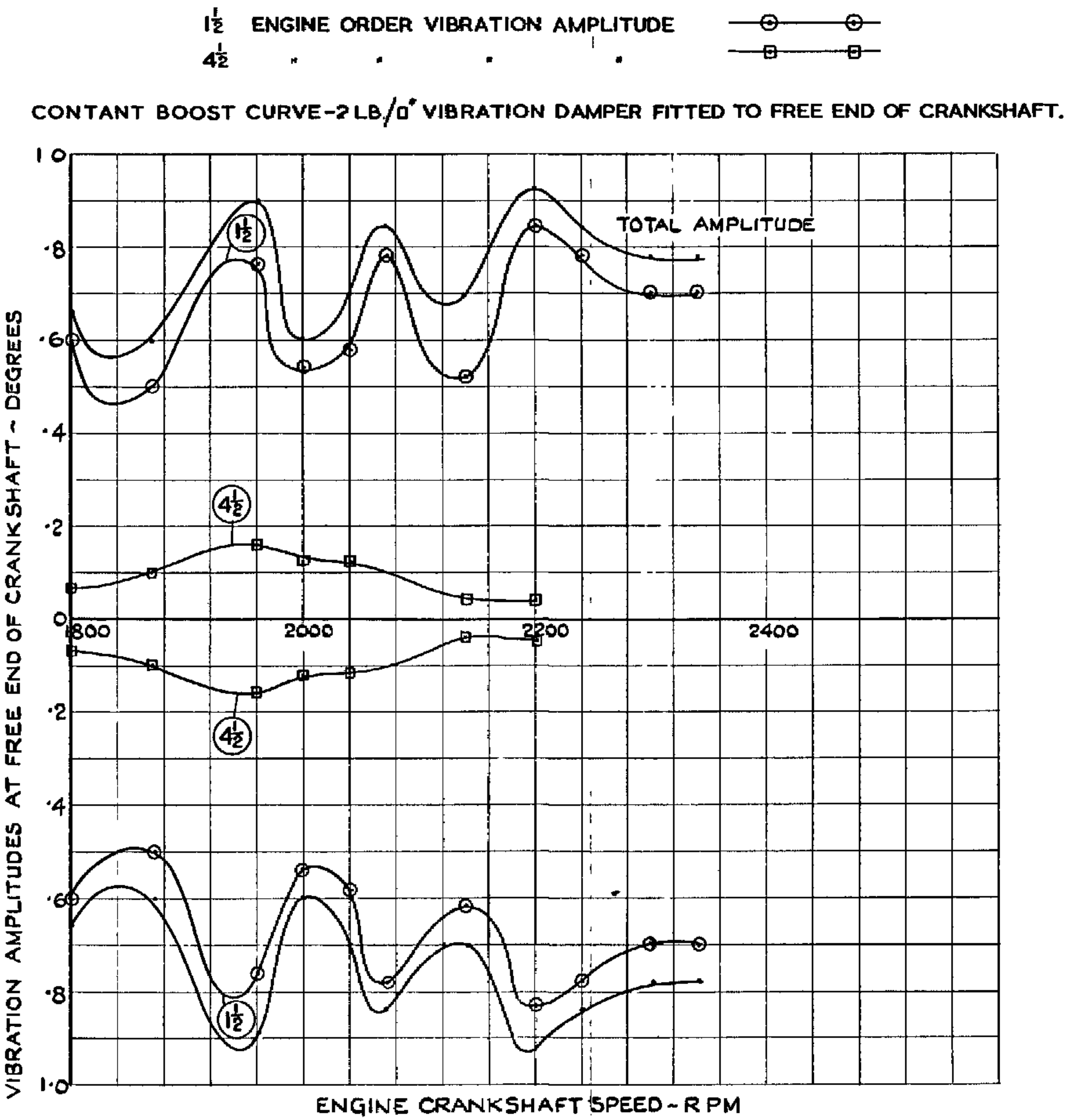
FIG. 4. MUSKETEER ENGINE RUNNING ON DYNAMOMETER TEST BED.

FIG. 5.



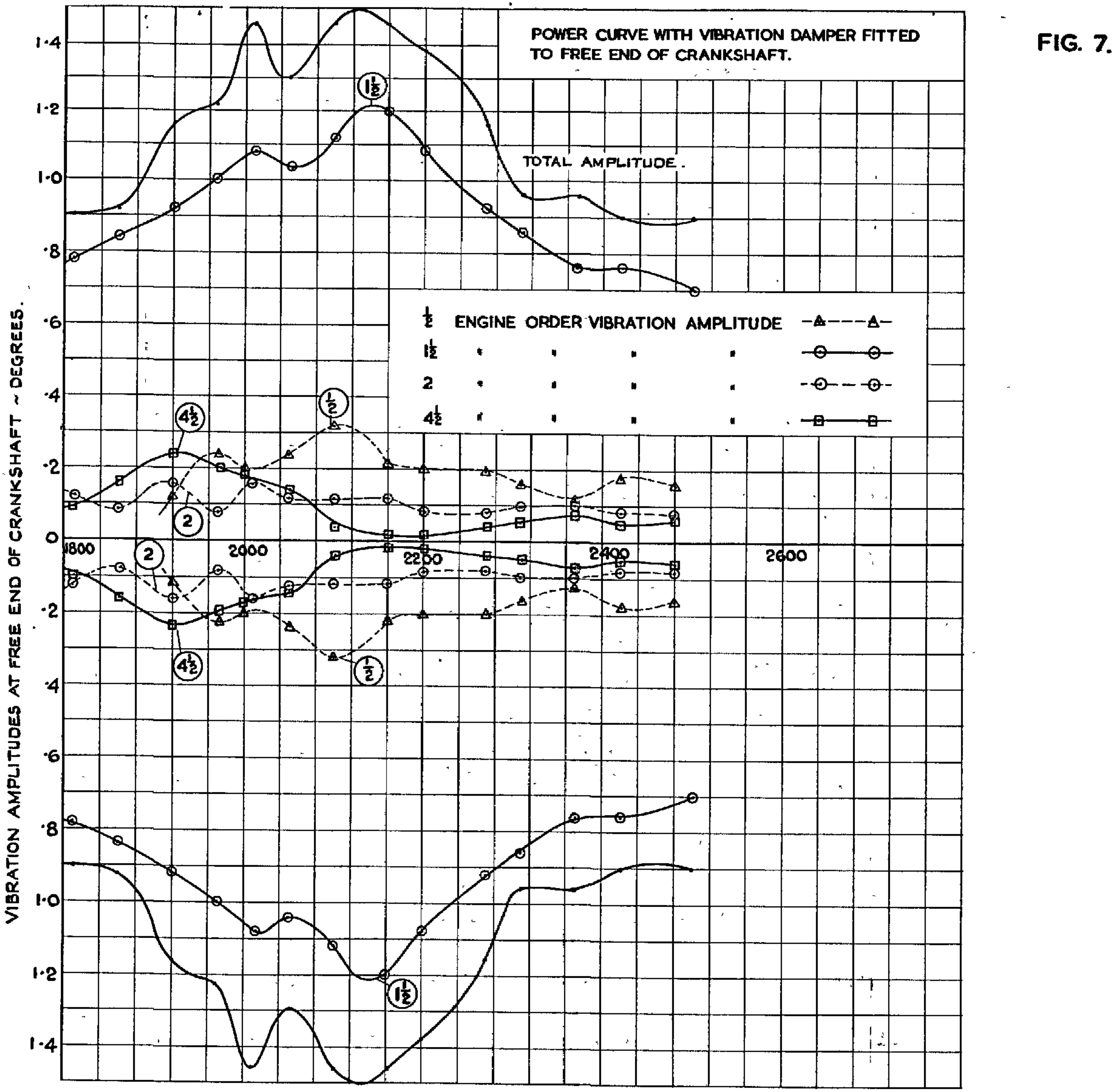
PREDOMINANT COMPONENTS OF TORSIONAL VIBRATION OF THE MUSKETEER ENGINE

FIG. 5. MUSKETEER ENGINE RUNNING ON DYNAMOMETER TEST BED.



PREDOMINANT COMPONENTS OF TORSIONAL VIBRATION OF THE MUSKETEER ENGINE

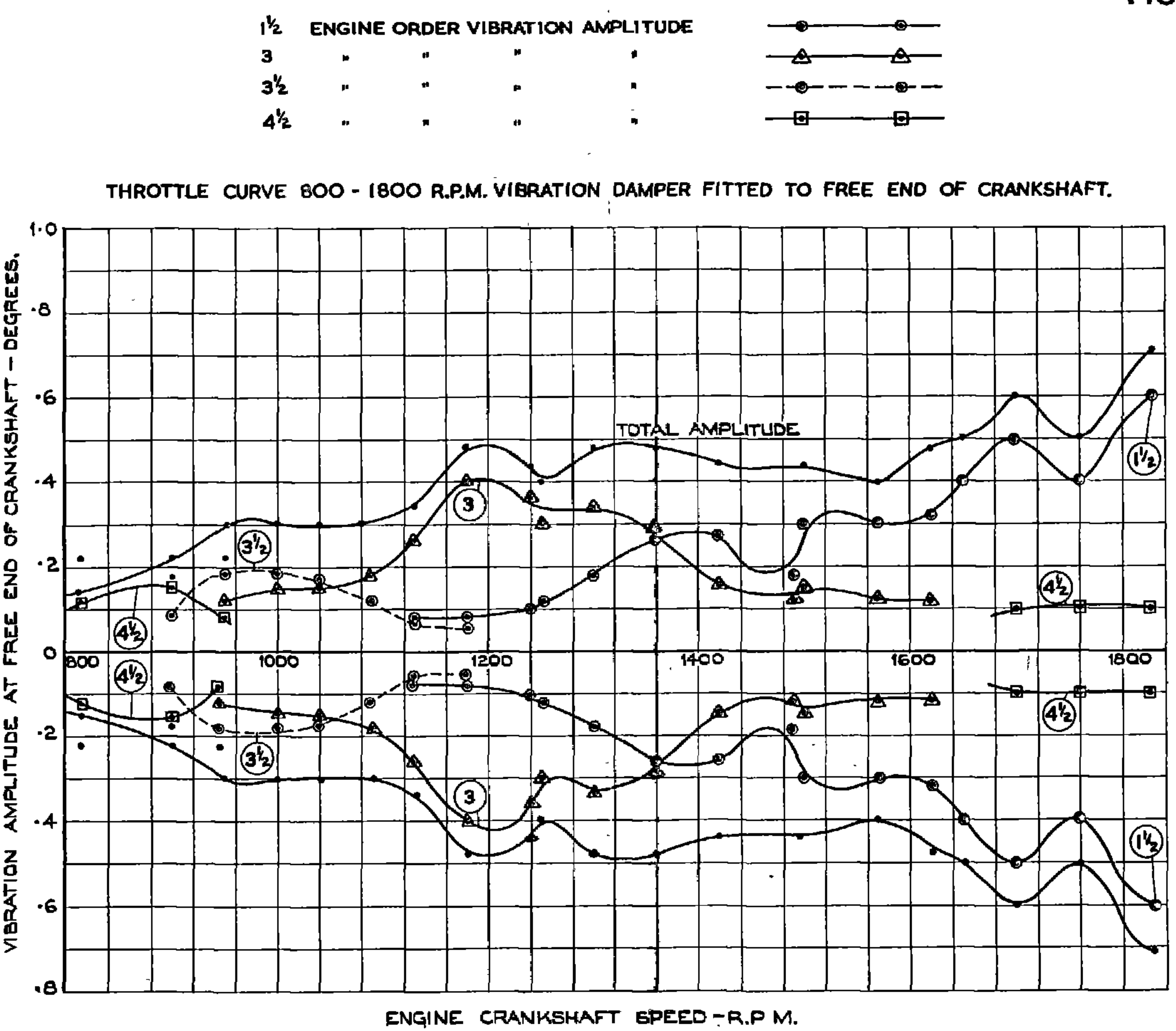
FIG. 6 MUSKETEER ENGINE RUNNING ON DYNAMOMETER TEST BED.



PREDOMINANT COMPONENTS OF TORSIONAL VIBRATION OF THE MUSKETEER ENGINE.

FIG. 7. MUSKETEER ENGINE RUNNING ON DYNAMOMETER TEST BED.

FIG. 8.



PREDOMINANT COMPONENTS OF TORSIONAL VIBRATION OF THE MUSKETEER ENGINE

FIG. 8. MUSKETEER ENGINE RUNNING ON DYNAMOMETER TEST BED.

FIG. 9.

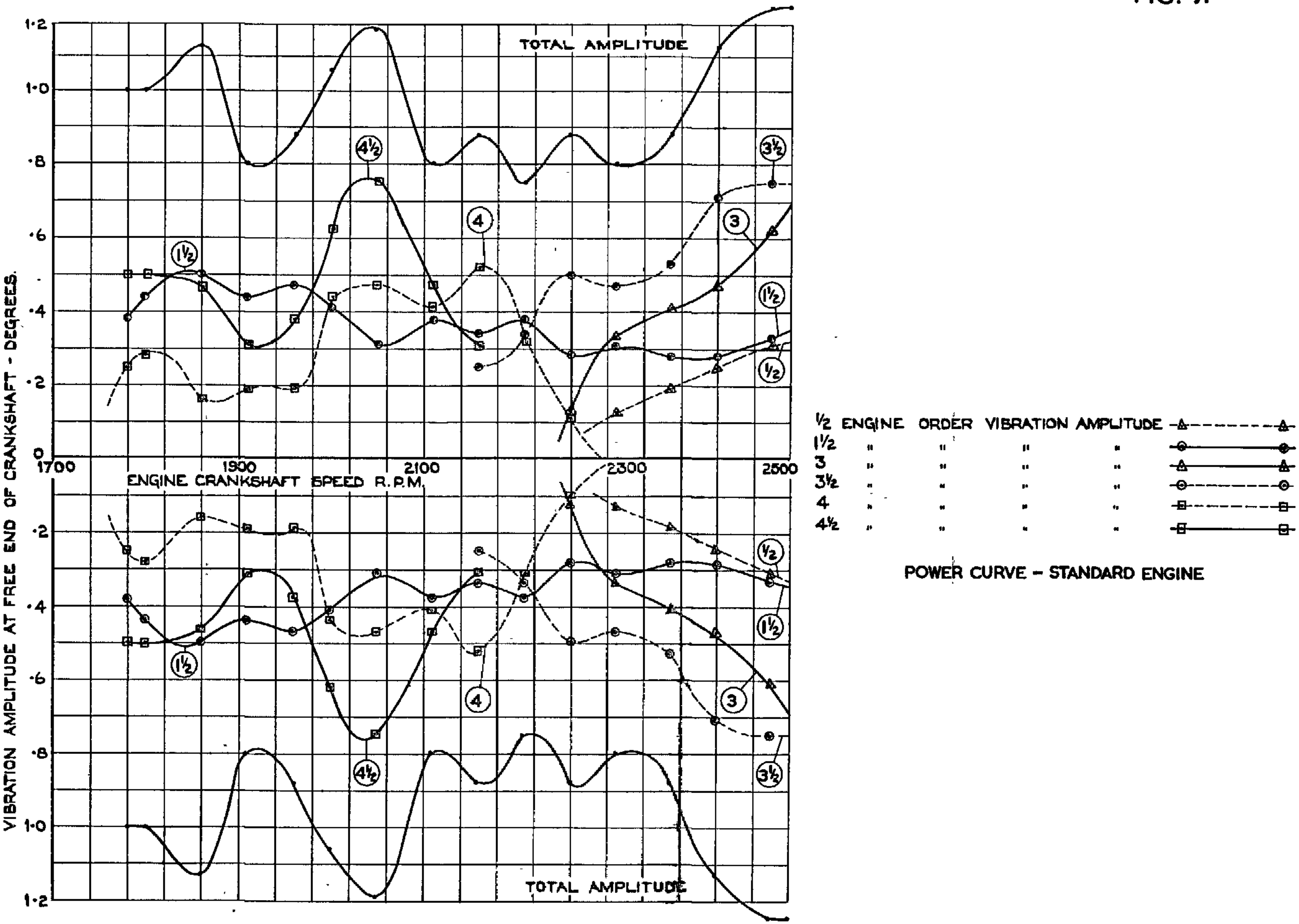


FIG.9. PREDOMINANT COMPONENTS OF TORSIONAL VIBRATIONS OF MUSKETEER ENGINE WITH PROPELLER.

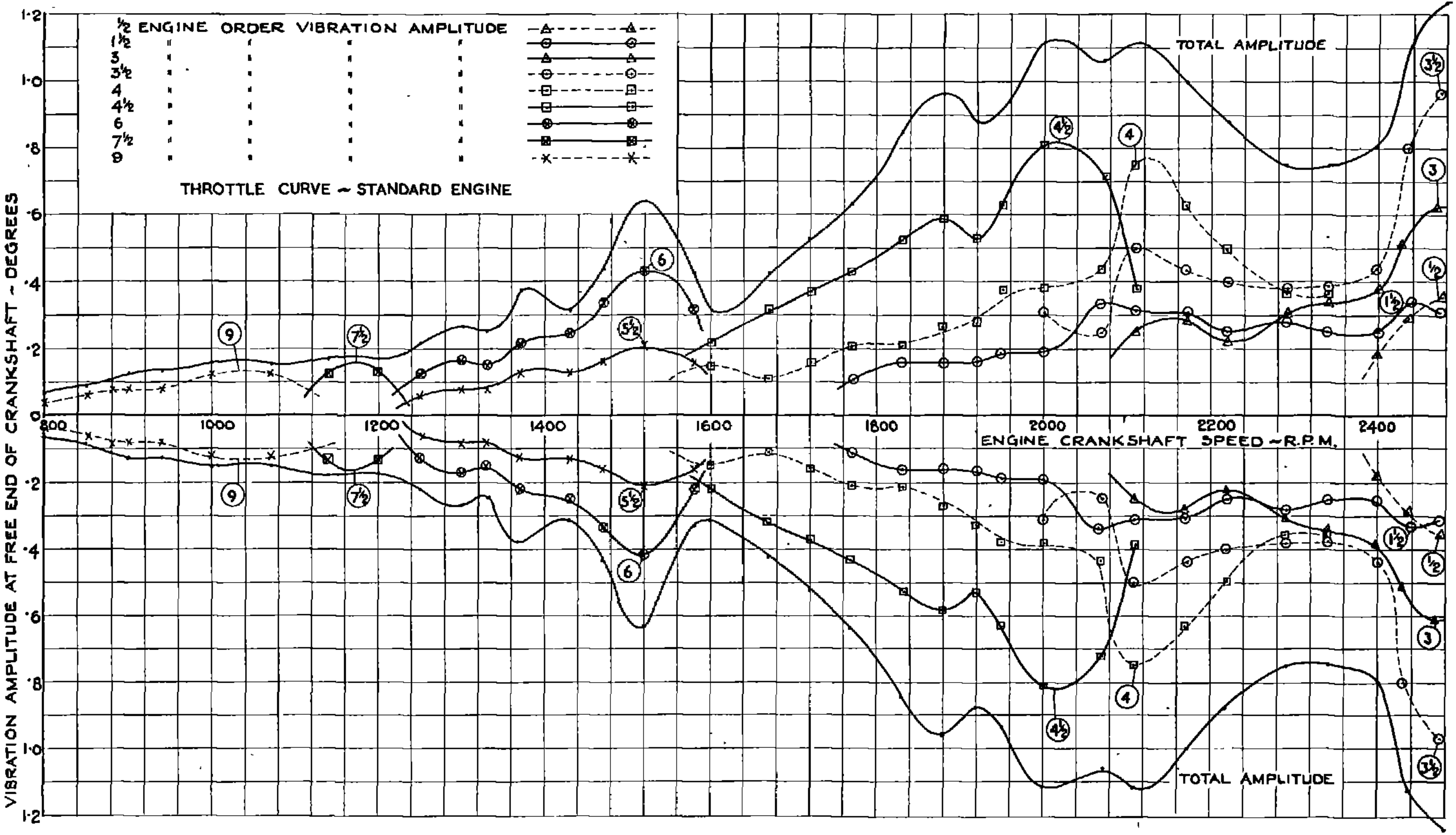


FIG.10. PREDOMINANT COMPONENTS OF TORSIONAL VIBRATION OF MUSKETEER ENGINE WITH PROPELLER

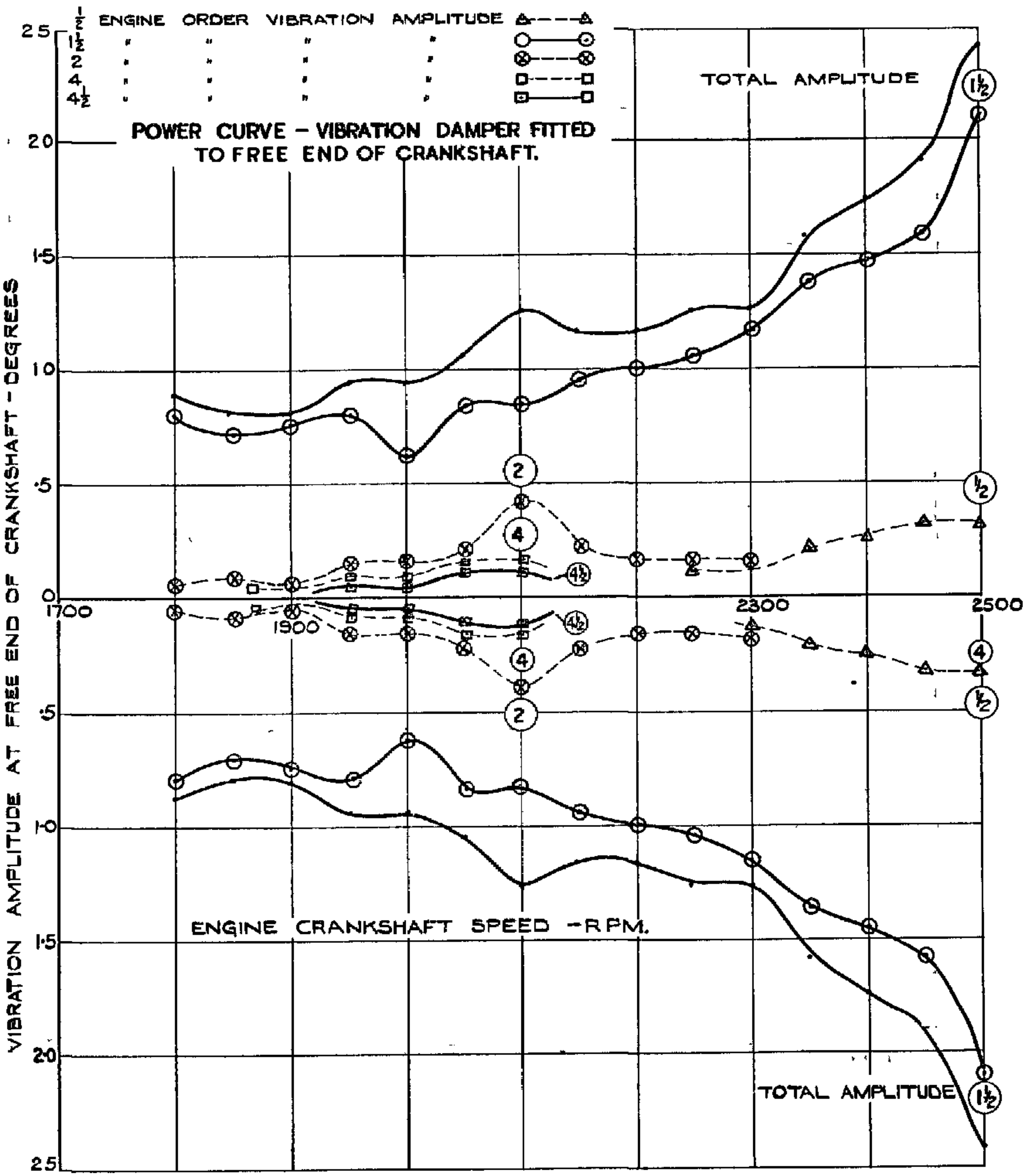


FIG. II. PREDOMINANT COMPONENTS OF TORSIONAL VIBRATION OF MUSKETEER ENGINE WITH PROPELLER.

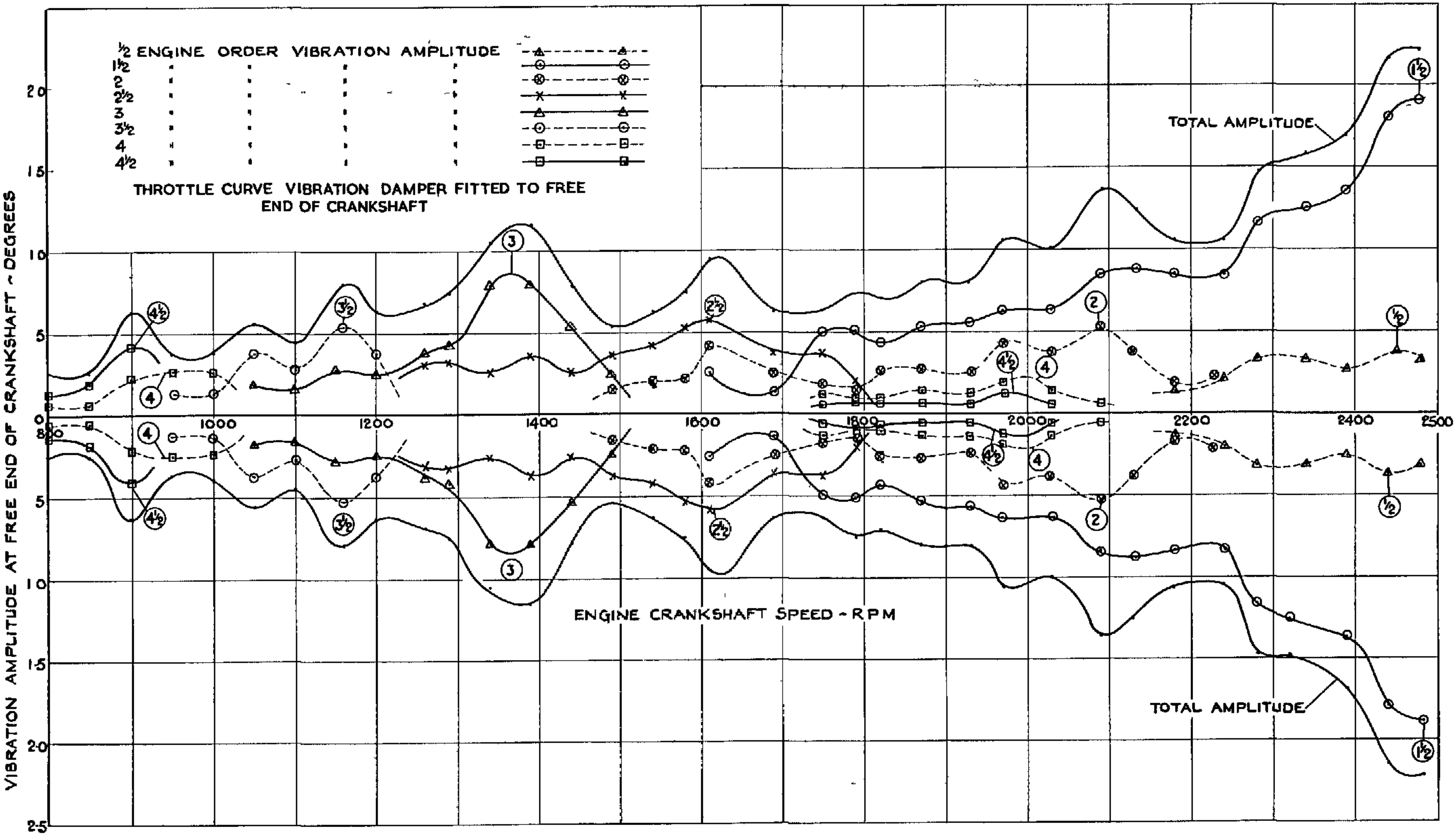


FIG 12. PREDOMINANT COMPONENTS OF TORSIONAL VIBRATION OF MUSKETEER ENGINE WITH PROPELLER

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