Vibration of Crankshaft-Propeller Systems

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MUCH remains to be desired in the vibration characteristics of present crankshaft-propeller systems, in the opinion of the author.

Discrepancies between torque-stand and flight measurements of torsional vibration on the same engine may explain propeller fractures due to the vibration of flexure. Recent fatigue fractures of crankshafts, differing from those due to torsional vibration, must be attributed to longitudinal vibration.

Degrees of freedom are discussed with a graphical summary of vibration frequencies. Vibration forms, sources, stresses, and resonances are subjected to mathematical analysis.

Three roads open to effective measures against vibration are given as: direct elimination of sources; subsequent destruction or damping of existing vibrations; and changing the pitch of the vibrating system, or displacing the resonance points to fields outside of the operating range. Of these methods the last is believed to be the most promising.

It is proposed to separate the crankshaft and propeller through the interposition of a flexible spring connection to obtain a crankshaft-propeller system that would be vibration-free to a large degree within the operating-speed range. Such construction would permit decreased crankshaft dimensions and weight.

FOR the noiseless and vibration-free operation of driving systems, freedom from vibration of the operating parts of the system is most important. This consideration is of utmost importance in the case of crankshaft-propeller systems, the vibration characteristics of which, to a great extent at present, leave much to be desired.

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A few years ago when crankshaft breakage was of frequent occurrence in in-line engines research, concerned most with the source of breakage, was directed exclusively toward the degree of freedom of torsional vibration. In these projects the propeller was regarded as a rigid body whose mass moment of inertia was so much greater than the mass of the driving system under consideration that even considerable change of the propeller moment of inertia did not affect measurably the vibration form and frequency. Accepting this hypothesis leads to the conclusion that the propellers used, brake or flight propellers of whatever type, are without effect on the torsional-vibration characteristics of the system.

Recently a few significant observations have been made that indicate that increasing the degrees of freedom in the system produces results of decided importance. The fundamental observations are as follows:

(1) Torsional-vibration measurements on the same engine made in the torque stand with a brake propeller and in flight with different flight propellers result in findings differing from one another more or less decidedly and to an extent far exceeding the deviation due to experimental error in measurement. Fig. 1 shows an excellent example of this deviation. The resonance point, $n_e/6$, found in measurements on the test stand at an engine speed of 1825 r.p.m. appears in flight measurements as two points of resonance, an upper and a lower one. The diagram brings to mind the well-known double pendulum as a resonance damper. In fact, later research showed that the natural torsional fre-

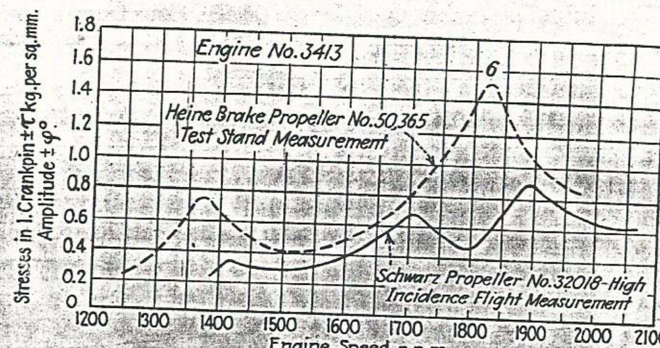

Fig. 1 – Torsional Vibration Curves of the Same Four-Cylinder In-Line Engine with a Flight Propeller and a Brake Propeller

Compare also this figure with Fig. 14 for Schwarz propeller No. 32018.

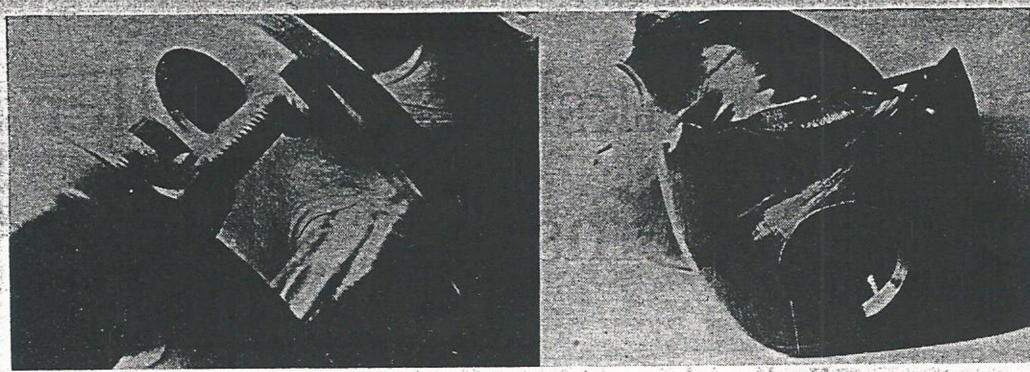

Fig. 3 – Fracture of an Aircraft-Engine Crankshaft Due to Longitudinal Vibration – (*Left*) Surface Crack in the Hollow – (*Right*) Surface of the Fracture

quency of the crankshaft-propeller system occurred at the same point as the first harmonic of the vibration of flexure at this speed (see Fig. 14). In this case, the propeller absorbed part of the vibration energy of the crankshaft and acted as a resonance-vibration damper for the torsionally vibrating crankshaft.

(2) As a possible result of this condition, we have the occurrence of fractures in wood and metal propellers used with certain engine types due to the vibration of flexure. In metal (Elektron) propellers the fractures occur solely in the base cross-section of the blade-root near the hub where there is increase of stress because of the clamping action of the hub and decrease in strength because of internal-frictional fatigue (see Fig. 2); in wood propellers cracks appear in the metal rim, and varnish peels at a definite point on the upper half of the blade (node of the first harmonic of the vibration of flexure).

(3) Recently, fatigue fractures, differing in appearance and location from those due to torsional vibration, have occurred in crankshafts, and these fractures must be attributed to longitudinal vibration (accordion-type vibration) of the crankshaft (see Fig. 3).

Degrees of Freedom

The foregoing observations make it necessary to consider all degrees of freedom that are possible and of interest in connection with resonance and the results attributable to these degrees of freedom. Fig. 4 characterizes the essential degrees of freedom; Figs. 5 to 9 give the forms of the corresponding free vibrations, the occurrence of which in the operating-speed range of the driving system is possible and could be shown.

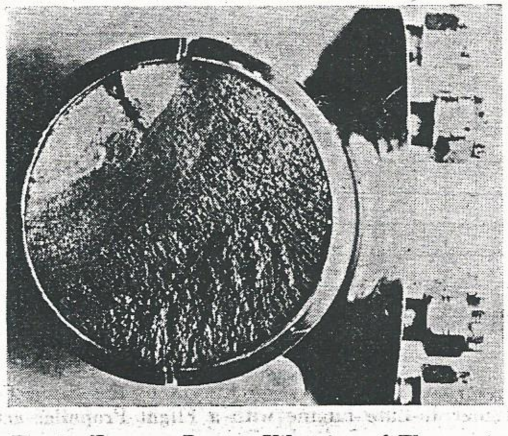

Fig. 2 – Fracture Due to Vibration of Flexure in an Elektron Propeller in the Base Cross-Section of the Blade Root Near the Hub

All degrees of freedom react upon one another, as can be seen easily by observing the change of form of the system through the introduction of a torsional moment (tangential force) on a crank throw. The introduction of a torsional moment (torsional vibration) brings about a change in the extent of the crankshaft in the direction of its longitudinal axis (longitudinal vibration) and the bending of the crankshaft (vibration of flexure). A torsional-vibration moment on the propeller hub causes the propeller blades to vibrate simultaneously in torsional vibration and the vibration of flexure. This relation can be understood easily if the crankshaft, by and large, is regarded as a helical spring and the propeller as an extension of it.

In connection with the resonance possibilities of the system of practical interest, it is important to know the vibration frequency and forms of the individual degrees of freedom of the complete driving group. In view of the interdependence of these degrees of freedom, it is not wholly sound to speak of the vibration frequency of the individual degrees of freedom; more accurately, there are only vibration frequencies and forms characteristic of the interrelated system as a whole; and, when resonance conditions exist, the deflection of all points of the system at the same time is naturally very considerable. The effective relationships are then most decidedly manifest and research on this phase is now in progress. However, for our preliminary consideration of resonance points, it seems permissible to speak of the vibration frequency and forms of the individual degrees of freedom, with the understanding that the findings set forth will be subject to certain exceptions and corrections, in most cases presumably only insignificant, to be disclosed by the research now in progress.

Fig. 10 summarizes the vibration frequencies; Figs. 5 to 9 show the associated free-vibration forms.

Vibration Forms

Fig. 5 shows the form of free torsional vibration for the customary driving system, almost the only fundamental vibration of interest, with one node close to the propeller. The resulting crankshaft fractures have been understood for some time and, fortunately, have been reduced greatly in number recently as a result of fundamental research and the development of suitable preventive measures.

Fig. 6 shows the lowest possible form of vibration of flexure with a reversing strain at the bearing, and with the proviso that the crankcase is rigid against bending and does not take part in the vibration. The corresponding vibration frequency (see Fig. 10) is very high, so that resonance in the operating range is very unlikely. In in-line aircraft engines with bearings at both sides of each crank throw, shaft breakages attributable to vibration of flexure are negligible as compared with the situation in automobile engines with fewer bearings and perhaps even as compared with radial aircraft engines.

Fig. 7 characterizes the form of free fundamental longitudinal vibration with a node close to the very considerable propeller mass.

An antecedent for the creation of this form of vibration is axial bearing play which must be present to a considerable extent. Longitudinal springing is for the most part provided by the crank cheeks which are susceptible to bending. Fig. 3 pictures a fatigue fracture in the hollow in the transition between crankpin and cheek, which fracture obviously is attributable to longitudinal vibration of the crankshaft.

The experimental determination of the frequency of longitudinal vibration through the creation of resonance by means

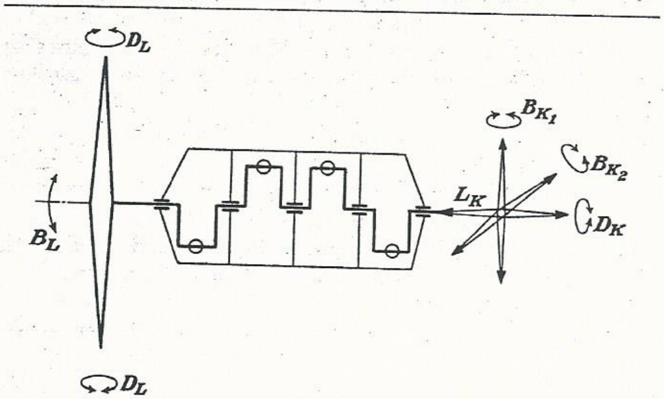

Fig. 4 – Degrees of Freedom of Vibration in the Crankshaft-Propeller System

L_K = Longitudinal vibration of the crankshaft (9)
D_K = Torsional vibration of the crankshaft (10)
B_{K_1}, B_{K_2} = Vibration of flexure of the crankshaft (11)
B_L = Vibration of flexure of the propeller (12)
D_L = Torsional vibration of the propeller (13)

of a small out-of-balance exciter is pictured in Fig. 11. For this apparatus, the frequency equation is:

$$\alpha = \beta tg\beta \qquad (1)$$

where

$$\alpha = \frac{M_u}{M_w} \qquad (2)$$

is the ratio of the additional mass of the out-of-balance exciter to the shaft mass. If in this determination the longitudinal-vibration frequency is found to be n', it is then reduced to the longitudinal-vibration frequency of the shaft without the additional out-of-balance exciter by means of the relation:

$$n' = \frac{2\beta}{\pi} n \qquad (3)$$

To make possible, in the future, advance calculations of the longitudinal-vibration frequency of crankshafts, longitudinal dilation tests are being made on different crankshaft types with the object of supplementing one of the familiar reduction formulas for torsional springing with corresponding reduction formulas for the longitudinal springing of crank throws.

Although as a rule, of all the forms of vibration of the crankshaft, only the fundamental vibrations are of importance, for the considerably more flexible propeller, even the harmonics must be considered. The corresponding forms of propeller vibration are shown in Fig. 8 (vibration of flexure) and in Fig. 9 (torsional vibration).

The forms of the vibration of flexure (Fig. 8) fall into two classes. In the forms denoted by *A*, the half on one side of the crankshaft axis is a reversed mirrored image of the other half; at the hub the form of the vibration is reversed relative to the common tangent; and the crankshaft is drawn into the vibrating system. In the form of vibration indicated by *B* the half on one side of the crankshaft axis is a mirrored image of the other half; the vibration nodes originate in the hub; and the crankshaft no longer participates in the vibration. The vibration frequency in class *B* is higher than the corresponding frequency of class *A*.

With relation to the considerably higher torsional resistance of the propeller in relation to the resistance to flexure, the form of fundamental torsional vibration shown in Fig. 9 should be considered.

Advance Calculation of Vibration of Flexure

This calculation frequently can be carried through with sufficient accuracy by the Rayleigh-Ritz method supplemented by graphical methods; Fig. 12 shows the experimental determination. The propeller is suspended lightly sprung in a rubber cable and is actuated by a small out-of-balance exciter fastened to the hub. Naturally, in this apparatus only the vibrations of class *B*, Fig. 8, are set up. To simulate more nearly actual operating conditions, the apparatus shown in Fig. 13 was developed. This apparatus reproduces, in addition, the effect of the coupling arrangement of the crankshaft-propeller system. The apparatus represents a vibrating image of the actual driving system.

A two-bearing shaft, the torsional springing of which corresponds to that of the crankshaft under investigation, carries at one end the appropriate propeller with a hub and, at the

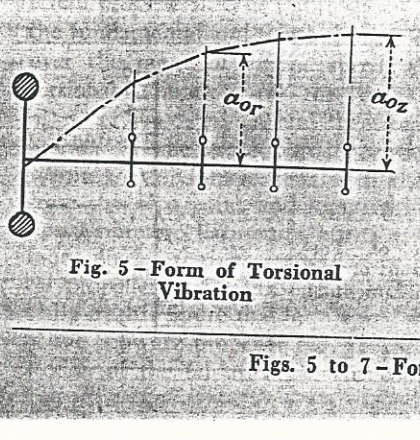

Fig. 5 – Form of Torsional Vibration

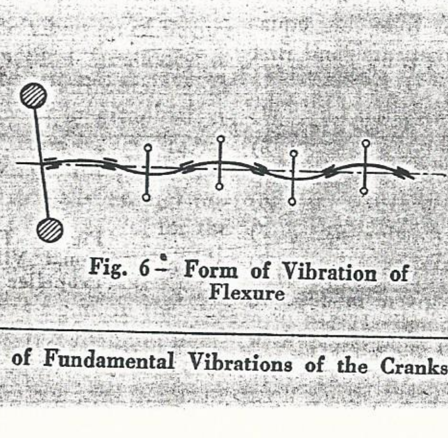

Fig. 6 – Form of Vibration of Flexure

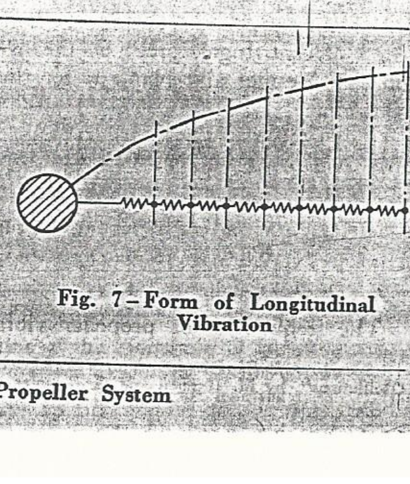

Fig. 7 – Form of Longitudinal Vibration

Figs. 5 to 7 – Forms of Fundamental Vibrations of the Crankshaft-Propeller System

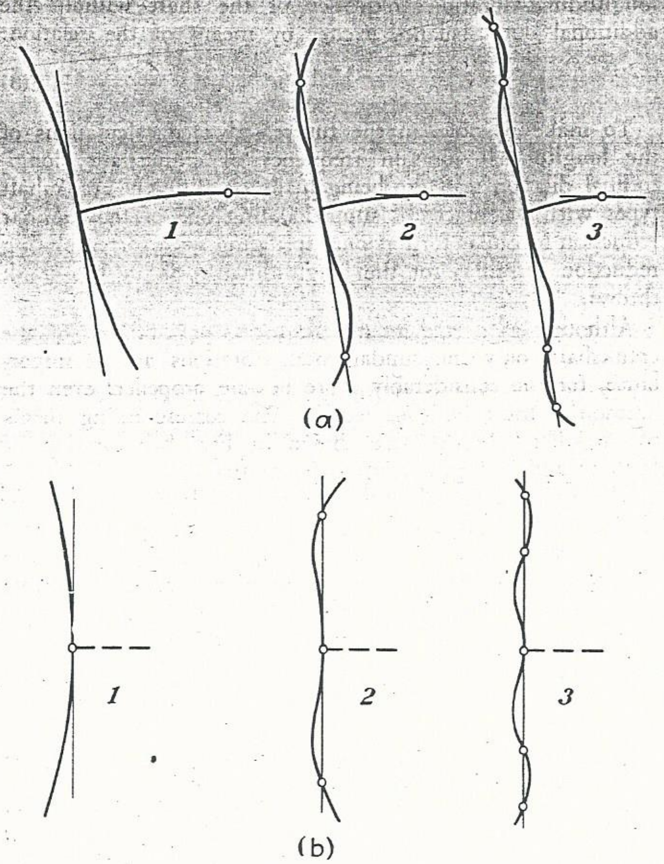

Fig. 8 – Vibration of Flexure of the Propeller

a (Above)—Fundamental and first and second harmonic with the participation of the crankshaft.

b (Below)—Fundamental and first and second harmonic without the participation of the crankshaft.

other end, a crossbar on which are two out-of-balance exciter elements displaced similarly at 180 deg. These elements introduce a pure vibrating moment corresponding to the torsional force harmonic set up by the engine. The moment of inertia of the crossbar corresponds practically to the sum of the mass of the engine-driving system. The speed of the out-of-balance moment is adjustable in the range of from about 200 to about 18,000 r.p.m., so that the entire frequency range of the different degrees of freedom of the system can be explored. With the help of a properly mounted mirror, the vibrations of the points of interest (crossbar, hub, propeller blades) are observed. The frequency determined by this apparatus for the stationary propeller is reduced to that of the rotating propeller (taking into consideration the stiffening effect of centrifugal force) by means of the following equations of Mesmer and Hansen[1] for the fundamental and first harmonic:

$$\nu_n^2 = \nu_o^2 + 1.45n^2 \quad \text{(Fundamental vibration)} \quad (4)$$

$$\nu_n^2 = \nu_o^2 + 4.4n^2 \quad \text{(First harmonic)} \quad (5)$$

where

ν_n = Vibration frequency at propeller speed n (6)

ν_o = Vibration frequency at propeller speed $n = 0$ (7)

n = Propeller speed (8)

The frequencies of propeller vibration given in Fig. 10 were determined in this apparatus; their agreement with the vibration behavior of the propeller revolving in the aircraft was confirmed by observations of the propeller in operation in the aircraft. A report will be given later on that part of the current research dealing with the coupling and the reciprocal effect of the degree of freedom of the system.

Vibration Sources, Stresses, and Resonance Points

Following this summary of the forms and frequencies of the free vibrations, arise the questions of sources of vibration, resonance points, and the stresses imposed by vibration on the system.

Sources

Although the creation of torsional and longitudinal vibration and the vibration of flexure in the crankshaft by the harmonics of gas and inertia forces is understood generally, such knowledge concerning the sources of propeller vibration is only partially available. Literature on the subject of propeller vibration either ignores the question of causes or goes into it only in a general way. As a rule, air-flow phenomena are given as sources – such as turbulence vortices, periodic pulsation of the propeller blades in passing wings or other resistance bodies, and overlapping in the propeller fields in multiengine aircraft. On the contrary, recent research, carried out by the German Institute for Aeronautical Research on the exciter apparatus, Fig. 13, and on aircraft, has demonstrated that, because of the coupling between crankshaft and propeller, the torsional harmonics of the engine should be regarded as of first importance in considering the source of propeller vibration.

In this connection it must be emphasized that occasionally there occurs in propellers vibration that is obviously attributable to air forces as the source. For these conditions, the term "flutter" has been coined, without any explanation of the phenomenon being given. As discovered in these observations flutter is accompanied by a detonating, machine-gun-like report, whereas vibration causes hardly any noticeable noise. Flutter occurs on the propeller test-stand when the propeller is driven by an electric motor and when there is no engine present, whereas vibration is produced only when the propeller is engine-driven. These observations seem to indicate clearly that air forces are the cause of flutter and that the torsional harmonics of the engine are the cause of vibration. Unlike the situation with regard to vibration in resonance, no clear understanding exists as to the nature of flutter, whether it be torsional vibration or a self-actuated phenomenon. An explanation of its relationships seems of utmost importance. At all events it should be stated that flutter has not yet, up to the present time, lead to propeller breakage, whereas the fractures illustrated in Fig. 2 can be explained clearly by vibra-

Fig. 9 – Form of Torsional Vibration of the Propeller (Fundamental)

[1] See *Aircraft Engineering*, Vol. VII, March, 1935, pp. 65-69; "Airscrew Oscillations", by M. Hansen and G. Mesmer.

				HZ
Crankshaft		Torsional vibration (fundamental vibration, Fig. 5)	$n_{e_1} \sim$ 4500 to 15,000 1/min.	75 TO 250
		Longitudinal vibration (fundamental vibration, Fig. 7)	$n_{e_1}^{\prime} \sim$ 6000 to 15,000 1/min.	100-250
		Vibration of flexure (Fig. 6)	$n_{e_1} \sim$ 23,000 to 30,000 1/min.	383-500
Propeller	Vibration of Flexure, ν_o	Fundamental vibration (Fig. 8, *B*, 1)	$n_{e_1} \sim$ 2500 to 4500 1/min.	42-75
		1st harmonic (Fig. 8, *B*, 2)	$n_{e_2} \sim$ 7500 to 14,500 1/min.	125-241
		2nd harmonic (Fig. 8, *B*, 3)	$n_{e_3} \sim$ 19,000 1/min.	316
	Torsional vibration (fundamental vibration, Fig. 9)		$n_{e_1} \sim$ 5500 to 7000 1/min.	91-116

Fig. 10 – Individual Vibration Frequencies for the Complete Driving Group of the Crankshaft-Propeller System

tion of flexure in resonance with the torsional harmonics of the engine.

Only slight danger is to be anticipated from flutter, since flutter is observed rarely and only with certain propellers. Moreover, the loud noise associated with flutter-speed causes avoidance of this speed, whereas the noise of the vibrating propeller can hardly be heard above the other noises of the driving gear in operation and, therefore, there is a possibility that the propeller may remain in the critical range for a long time without the fact being evident.

The manner in which torsional harmonics excite vibrations has been understood generally for some time as a result of recent research on crankshaft torsional vibration. This research has made it possible to specify for each driving system of a given firing order, coupling system, and torsional-vibration form, the characteristic series of torsional harmonics to be anticipated, both as to amplitude and frequency. For instance, for the normal four-cylinder, in-line engine with rigidly coupled propeller, this series is:

$2n, 4n, 6n$ ....(n = engine speed)

for the six-cylinder type with normal firing order:

$3n, 4\frac{1}{2}n, 6n$ ....

for the twelve-cylinder V-type with 60-deg. crank angle and normal firing order:

$3\frac{1}{2}n, 4\frac{1}{2}n, 6n$ ....

Resonance Points

The coincidence of any of these externally excited frequencies with a natural frequency of the crankshaft-propeller system is known as a resonance point and hence a critical speed. Fig. 14 shows these relationships for a four-cylinder in-line engine, which was flown successively with eight different propellers. The figure shows for the various propellers, the fundamental and first harmonic of the vibration of flexure over the range of engine speed, the natural frequency of torsional vibration of the crankshaft (horizontal zone $n_e \sim$ 11,000 1/min.), and the frequency of longitudinal vibration of the crankshaft (horizontal zone $n_e \sim$ 14,750 1/min.). All intersections with the straight lines radiating from the zero point and representing externally excited torsional-vibration harmonics, $2n, 4n, 6n$ ...., for the four-cylinder engine, are resonance points and likewise critical propeller and crankshaft speeds. It is to be noted especially in this figure that, at a speed of 1825 r.p.m. in the case of the Schwarz propeller No. 32018, there is a common intersection of the 6th torsional harmonic, the first harmonic of vibration of flexure, and the frequency of natural torsional vibration of the crankshaft ($n_e \sim$ 11,000 1/min.). The effect of this natural coincidence on the torsional-vibration curve of the engine is shown in Fig. 1. Mention already has been

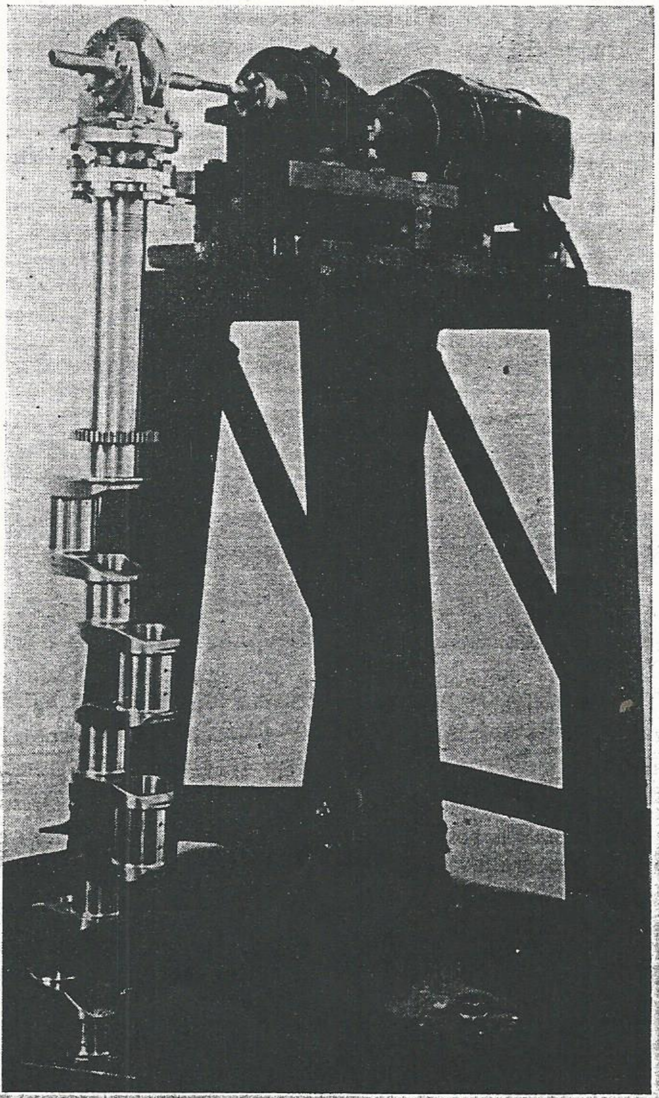

Fig. 11 – Determination of the Frequency of Longitudinal Vibration of the Crankshaft by Setting Up Resonance by Means of an Out-Of-Balance Exciter

made of the resonance damping effect of this propeller and the displacement of the torsional critical point caused by it.

Another diagram of this type for the same engine but with other propellers, is shown in Fig. 15. This diagram is worthy of note in that it confirms the exclusive effect of the torsional harmonic as a cause of vibration. The propellers denoted as No. 5220 and No. 52277 continued in flight operation for about 1½ years without interruption, whereas the propeller marked No. 52269 broke repeatedly after short operating periods (fracture of the blade tip in the node of the first harmonic). Propeller No. 52235 is a new development, the operating results with which are not yet available. The diagram shows that propeller No. 52269 vibrates in resonance with the sixth torsional harmonic close to the much-used maximum speed of the engine (2150 r.p.m.), whereas the corresponding resonance point for the three remaining propellers is not reached within the operating speed.

The diagrams in Figs. 14 and 15 give the impression that, within the operating range of the engine, there is hardly any non-critical speed. Hence arises the question of the vibration deflections in the critical speeds, and hence arises that of vibration stresses and the risk of breakage connected with them.

Vibration Stresses

The deflection in resonance of sole interest in this connection can be determined basically by a consideration of the energy force involved, since this deflection is determined through the equilibrium between the work of excitation and the work of damping. If the amount of this work and also

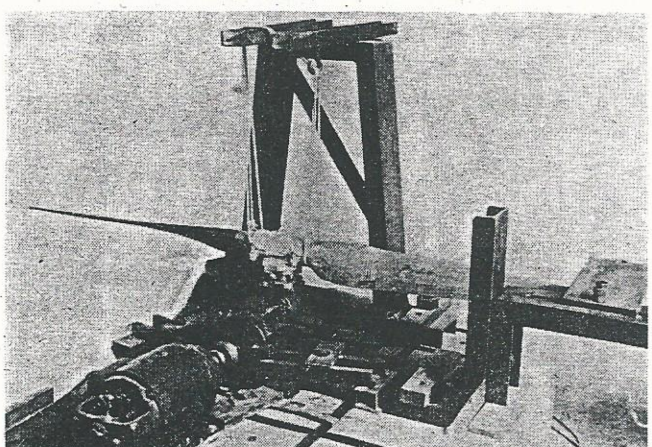

Fig. 12 – Determination of the Frequency of Vibration of Flexure of Propellers by Setting Up Resonance by Means of an Out-Of-Balance Exciter at the Hub

the fatigue strength of the individual parts of the system are known, then an estimation of the breakage risk at the resonance points is possible.

The energy involved in setting up the torsional harmonic can now be set forth exactly. For example, at the moment of observation, there is effective on the n throw of the crankshaft with z throws the following harmonic torsional component:

$$T_{n_k} = T_{o_{n_k}} \sin k(wt - \phi_n) \qquad (14)$$

where k is the degree of the harmonic in question and ϕ_n its phase angle.

At the same time the amplitude of torsional vibration of this throw is:

$$a_n = a_{o_n} \sin k(wt - \psi_n) \qquad (15)$$

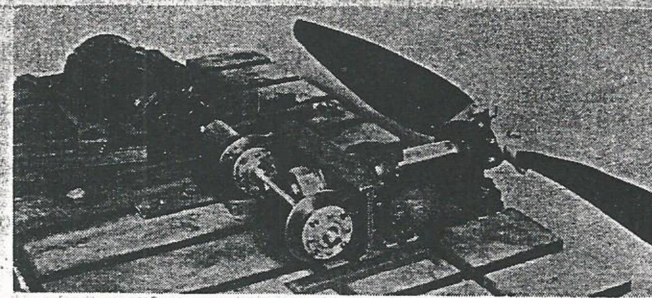

Fig. 13 – Exciter Apparatus for Combined Vibrations of the Crankshaft-Propeller System – Reproduction of Vibrating Conditions of the Driving System

where ψ_n is the phase angle of the deflection. During the time dt the crank throw in question vibrates about the angle:

$$da_n = a_{o_n} \cos k(wt - \psi_n)\, d(kwt) \qquad (16)$$

The work performed by the torsional harmonic during the time dt then amounts to:

$$dA_{T_{n_k}} = T_{n_k}\, da_n \qquad (17)$$

The amount of work performed during a full vibration is then summarized as:

$$A_{T_{n_k}} = 2\int_o^{\pi} dA_{T_{n_k}} \qquad (18)$$

Carrying through the integration results in:

$$A_{T_{n_k}} = \pi T_{o_{n_k}} a_{o_n} \sin k(\psi_n - \phi_n) \qquad (19)$$

If then there is taken the sum of all z crankshaft throws, taking into consideration their opposing displacement $(\Sigma)_{geom.}$ and with the proviso that all z throws attain their maximum vibration at the same time (ψ_n for all throws alike), then the total work of excitation of the k torsional harmonics for the entire engine is:

$$\sum_1^z A_{T_{n_k}} = \pi \sum_{geom.} T_{o_{n_k}} a_{o_n} \sin (k\psi_n - \gamma) \qquad (20)$$

where γ is the phase angle of the resultant from

$$\sum_{geom.} T_{o_{n_k}} a_{o_n} \qquad (21)$$

The amount of this work reaches its maximum in resonance; then the phase angle of the vibration $k\ \psi_n$ appears perpendicular to the phase angle γ of the resultant from

$$\sum_{geom.} T_{o_{n_k}} a_{o_n} \left(\text{d.h. } k\psi_n = \gamma + \frac{\pi}{2}\right) \qquad (22)$$

Therefore the exciting force of the torsional harmonics in resonance for all z throws is:

$$\sum_1^z A_{T_k} = \pi T_{o_k} \sum_{geom.} a_{o_n} \qquad (23)$$

or, with interchangeably similar torque diagrams for all cylinders of the engine

$$(T_{o_{n_k}} = \text{constant} = T_{o_k}): \qquad (24)$$

$$\sum_1^z A_{T_k} = \pi T_{o_k} \sum_{geom.} a_{o_n} \qquad (25)$$

If then the unknown amplitudes a_{o_n}, with the aid of the known form of free torsional vibration (see Fig. 5), are reduced to the single unknown amplitude a_{o_z} of the last crankthrow then there results finally:

$$a_{o_n} = \beta a_{o_z} \qquad (26)$$

$$\sum_{1}^{z} A_{T_k} = \pi T_{o_k} \sum_{\text{geom.}} \beta a_{o_z} \qquad (27)$$

The amplitude T_{o_k} of the k torsional harmonic is known from the harmonic analysis of the torque diagram of the engine. For example, in Fig. 16 are shown the amplitudes of the individual harmonics up to the 16th degree for the four-stroke cycle carbureter engine, in per cent of the mean indicated pressure, and these amplitudes can be determined for all four-stroke cycle carbureter engines from this curve. Corresponding diagrams for the two-stroke cycle and for the various Diesel types may be developed on the basis of analysis of the torque diagram.

If, just as for the exciting force, the sum of all the damping forces at the various points of the vibrating system be set forth then, from a comparison of both of these amounts of work, the resonance amplitude at each point of the system can be calculated beforehand. A condition for practically carrying through such a calculation would be the assumption that damping forces are proportionate to the speed. When, however, it is considered that this assumption is, in part, not valid for the various damping forces acting on the crankshaft-propeller system (for example, piston and bearing friction, impact losses in the driving gear, and dissipation of vibration energy in the engine mounting); and that, further, damping forces at different parts of the system obey different laws, then the hopelessness of arriving at any advance calculations of total damping force that will accurately represent actual conditions can be understood. In fact, all efforts in this direction have up to the present been futile. For the present it is far more advisable to measure the amplitudes and stresses of vibration directly on the driving system with the help of suitable test apparatus. This statement is true at least for torsional vibration of the crankshaft. For the participating vibration of the propeller, the prospects for damping calculations are perhaps more favorable since the damping effects of air and construction material, here the only effective forces, are more susceptible to calculation.

However, the mathematical analysis of the work of excitation developed previously performs a worthwhile service in connection with a qualitative judgment of the risk of breakage at the resonance points. For a final amplitude, $a_{o_z} = 1$, a specific excitation force for individual harmonics may be given and hence at least the relative danger at each of the resonance points may be judged in advance (see Figs. 14 and 15) and appraised the one against the other. For example, on the basis of such an investigation the importance of in-

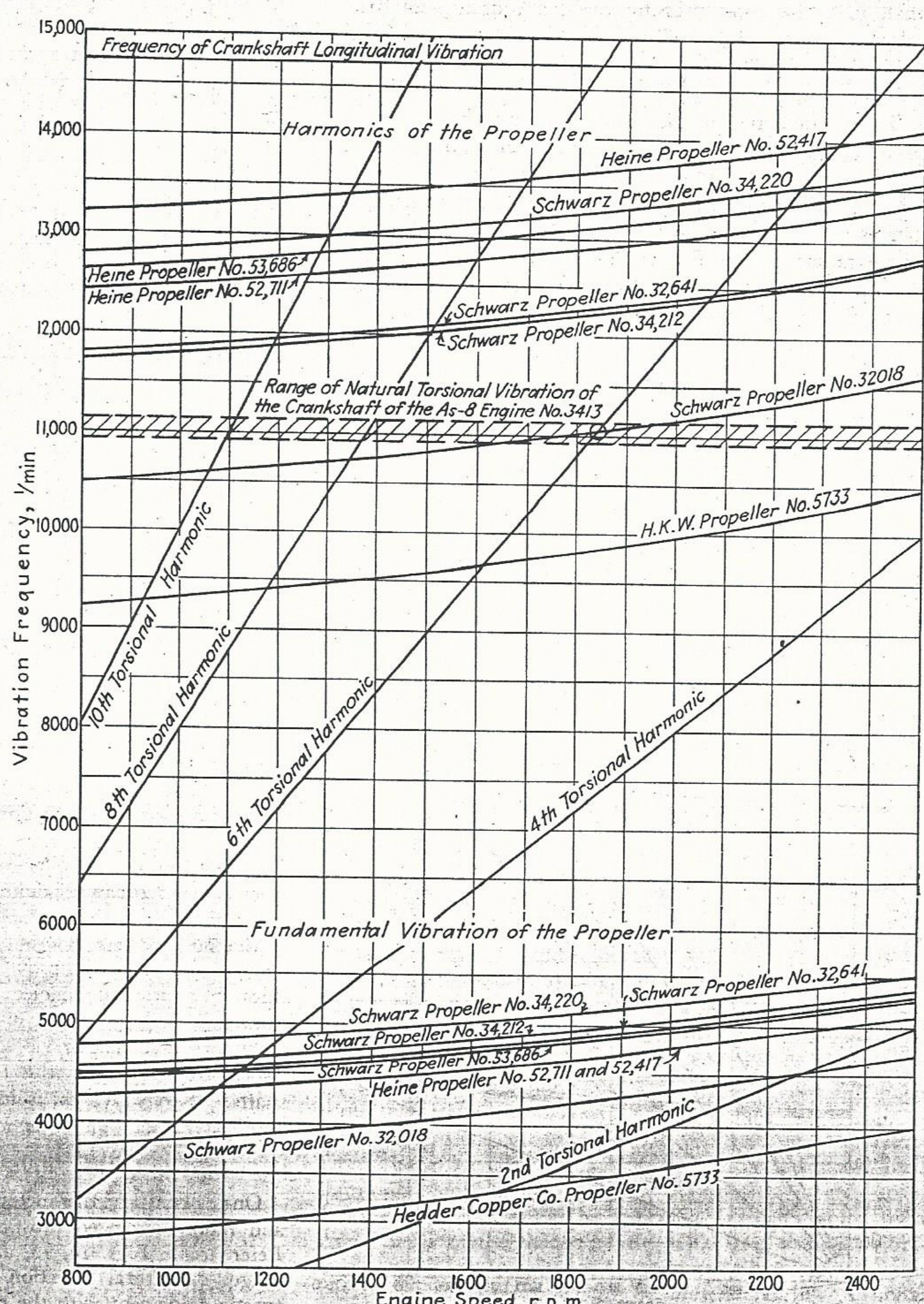

Fig. 14 – Points of Resonance between the Torsional Harmonics of a Four-Cylinder In-Line Engine, the Frequency of Vibration of Flexure of Various Propellers, and the Frequency of Torsional and Longitudinal Vibration of the Crankshaft

(Critical propeller and crankshaft speeds; compare also this figure with Fig. 1, with reference to Schwarz propeller No. 32018) dividual critical points through various indicated ranges was characterized in Fig. 15.

Suitable and tested measuring apparatus is available for the direct measurement of the amplitude of torsional vibration of crankshafts (German Institute for Aeronautical Research torsiograph). Vibration-measuring or stress-measuring apparatus for propeller vibrations is now being developed; the technical difficulties, involved in such measurements, that must be overcome are naturally decidedly much greater. So that, in a few most pressing cases, a rough estimation of vibration stresses in propellers might be made, the following method was developed in the German Institute for Aeronautical Research:

The tip deflection of the propeller operating and vibrating on an aircraft is estimated by means of a measuring rod. The propeller is then mounted in the exciter-apparatus shown in Fig. 13 and set into vibration of the same form and with the same tip deflection. The form of vibration is then recorded photographically (see Fig. 17) and measured accurately. Assuming that this form of vibration does not differ essentially from that of the vibrating propeller, the vibrating stress

$$\delta = \frac{ETy''}{W} = Eey'' \tag{28}$$

is determined through graphical differentiations of the form of vibration (see Fig. 18). In the foregoing case, an Elektron propeller, the considerable bending stress of

$$\delta \sim \pm 10 \text{ kg./mm.}^2 \tag{29}$$

was evident at the location of the node of the first harmonic of the vibration of flexure. This bending stress surpassed by far the endurance strength of the propeller blade and indicated the risk of fatigue fracture of the propeller.

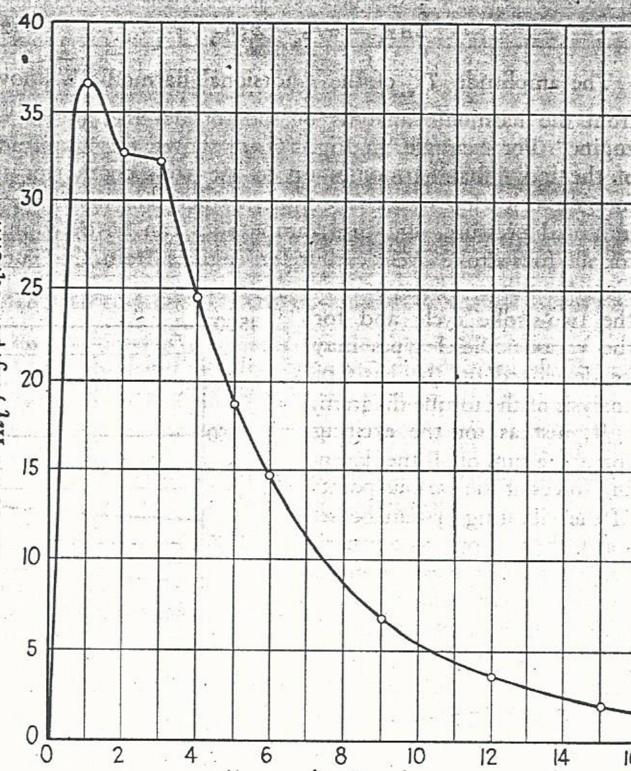

Fig. 16 – Amplitudes of Torsional Harmonics in Per Cent of the Indicated Mean Pressure, Valid for All Four-Stroke-Cycle Carbureter Engines

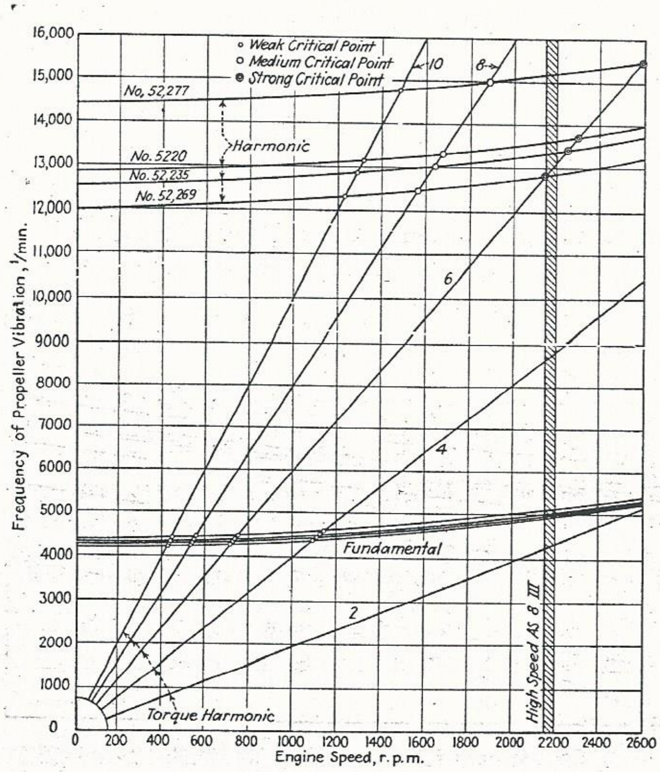

Fig. 15 – Resonance Points between the Torsional Harmonics of a Four-Cylinder In-Line Engine and the Frequency of Vibration of Flexure for Different Propellers (Critical Propeller Speed)

Efforts Directed Against Vibration

The trend toward power and speed increase makes imperative, to a greater extent than ever before, effective measures against vibration. For this purpose, basically three roads are open:

(1) Direct elimination of the sources of vibration.

(2) Subsequent destruction or damping of existing vibrations.

(3) Changing the pitch of the vibrating system, or displacing the resonance points to fields outside the operating range.

Method (1) will not be discussed at this time; its possibilities (for example, providing a vibration-free speed range through appropriate choice of firing order) are extremely limited. The destruction or damping of existing vibrations through supplementary vibration dampers is familiar through its use in connection with the torsional vibration of crankshafts. Its effectiveness is in reality limited to damping degrees of freedom, and it is accompanied by increased weight.

Changing the Pitch of the Vibrating System

On the contrary, this method seems to offer more promising and more variegated prospects without increase of weight. Reference to Figs. 10, 14, and 15 shows that the frequency ranges of natural vibration of current driving systems are exactly coincident with the effective section of the frequency band of excitation for the torsional harmonics (harmonics of lower order and of larger exciter amplitude, Fig. 16). Conditions for the creation of critical speeds could hardly be more favorable, and the question arises as to whether a displacement of the natural frequency spectrum into the range of higher or lower frequency seems promising.

In a few of the more recent experimental engines, an attempt has been made to shift over into the range of higher frequency (n_e = 12,000 to 18,000 1/min.) by building as rigid a crankshaft against torsional vibration as possible, in order to come into resonance only with the higher torsional harmonics of lower exciter amplitude (see Fig. 16). However, the crankshaft rigid against torsional vibration is obtainable only at the cost of distinct disadvantages (increased weight and size, higher peripheral speed, and increased bearing troubles). In this connection, the advance in vibration technique will either be overtaken or outrun by the approaching speed increase. The behavior of current propellers in connection with vibration will not be affected significantly by crankshaft rigidity. A decided displacement of the frequency of propeller vibration seems hopeless as regards current propeller types, since blade dimensions are determined basically by aerodynamic requirements. The use of hollow or articulated propellers would of course result in a decided raising of the frequency of vibration of flexure; at the same time the frequency of fundamental vibration would swing

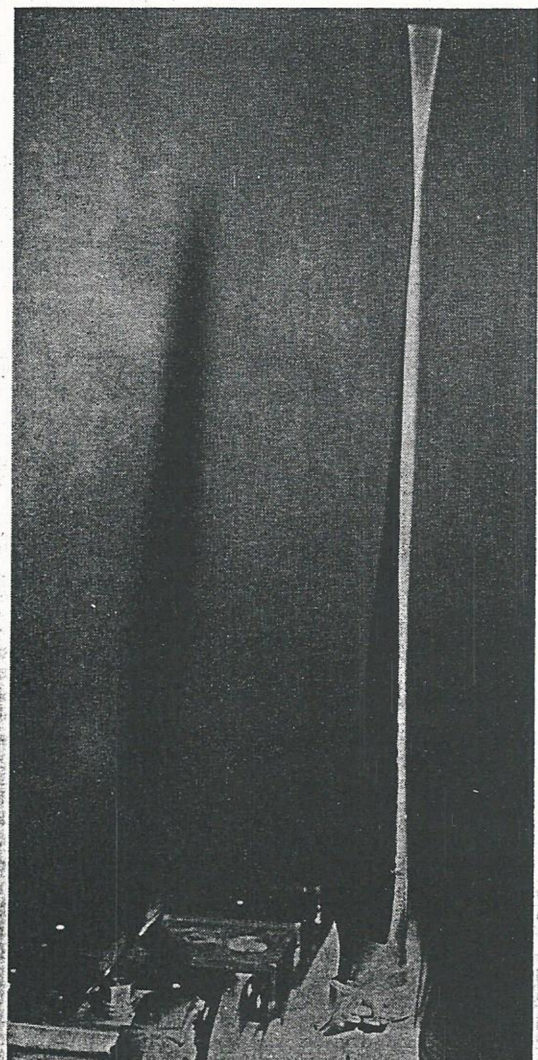

Fig. 17 – Vibrating Propeller Blade (First Harmonic) on the Exciter Apparatus Shown in Fig. 13

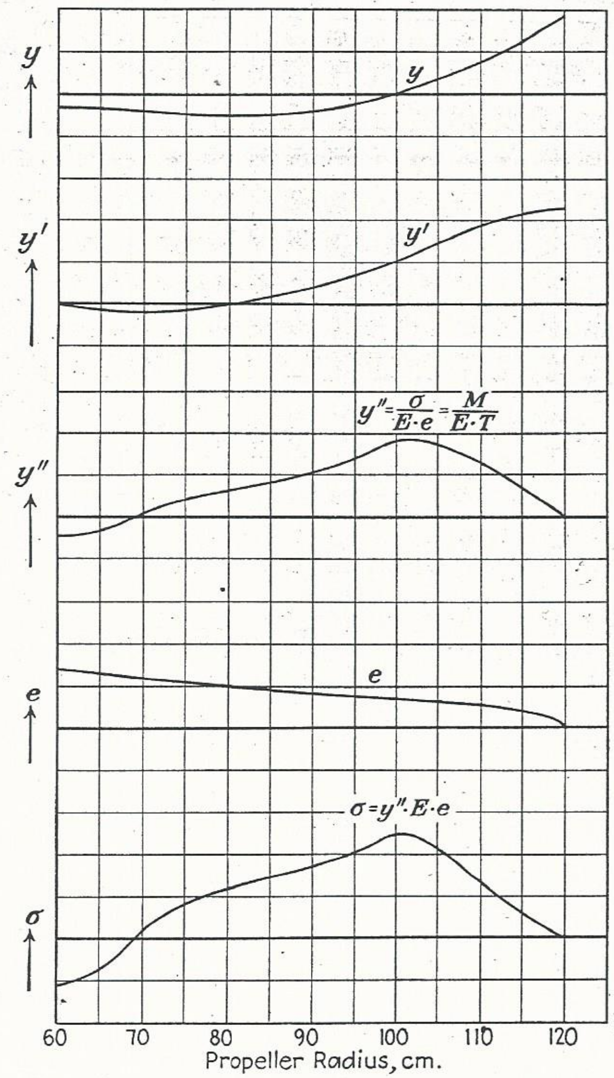

Fig. 18 – Vibration Forms and Stresses of the Propeller Blade Shown in Fig. 17 (Upper Half of Blade)

to the neighborhood of the present first harmonic of vibration of flexure.

Efforts pointed toward the opposite direction in general seem promising. With the understanding that the frequency of torsional vibration of the crankshaft is really controlled by the propeller mass rigidly joined to it, it is again proposed, obeying the laws of vibration, to separate crankshaft and propeller through the interposition of a flexibly sprung connection. Such a sprung connection ideally should transmit only the mean torque moment of the engine to the propeller. The effect, from the viewpoint of vibration technique, would be somewhat as follows:

The frequency of fundamental vibration of the system would be so low that its resonance with the lowest torsional harmonic would still occur below the operating speed range.

This condition would be generally attainable since, as a consequence of the changed form of the free torsional vibration (approximating similar vibration amplitudes for the collective throws), there would be present no low torsional harmonic of noticeable excitation amplitude except that of the order $n.z/2$ (n = engine speed; z = number of cylinders). However, in each case a broad vibration-free zone within the operating speed range should be provided by

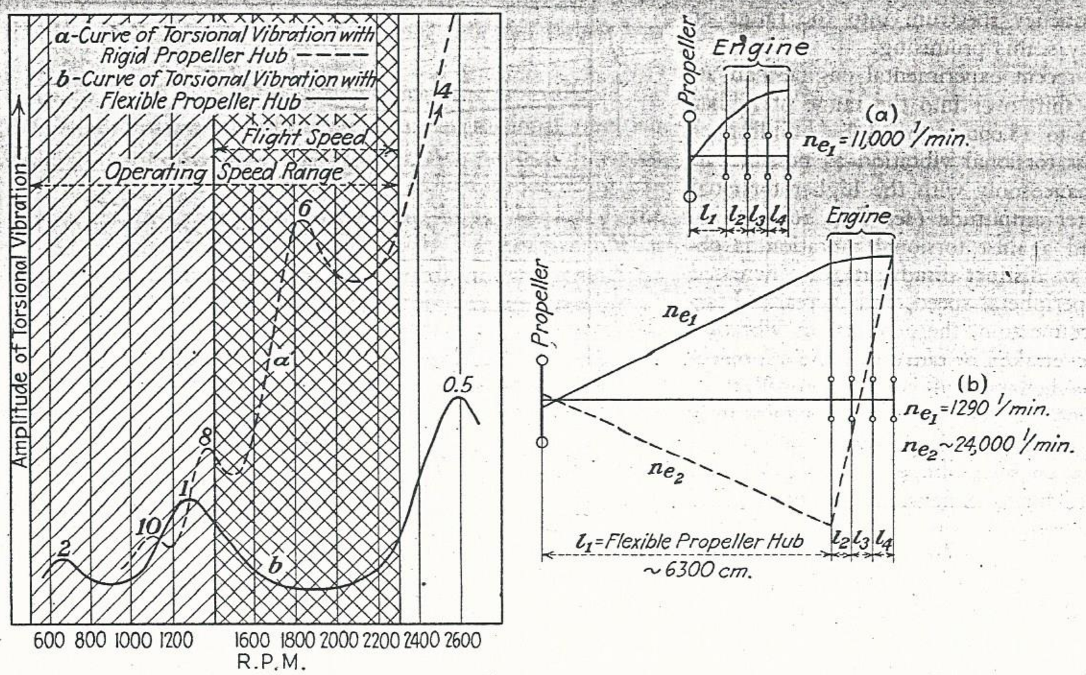

Fig. 19 – Forms and Curves of Torsional Vibration of a Four-Cylinder In-Line Engine with a Rigid and a Flexible Propeller Hub (Propeller-Hub Suspension Through Torque Arms and Tubes)

means of suitable dimensions of the sprung connecting member. The first harmonic, moreover, would almost coincide with the fundamental vibration of the free crankshaft and would therefore rise to double the value of the current frequencies of fundamental torsional vibration of rigidly coupled propellers. No noticeable moment of torsional vibration originating in the crankshaft could be transposed to the propeller over the sprung connecting member; propeller vibrations due to torsion would no longer be possible. The crankshaft-propeller system would therefore to a large degree be vibration-free within the operating speed range.

Future Developments in Driving Systems

In these developments the use of such sprung connecting members would have broad and favorable results. Crankshaft dimensions (weight and size) would be decreased and could be basically controlled by the mean torsional moment. In view of the increasing speed, the weight thus saved could, with the greatest advantage, be converted into counterweights, which should, by all means, not be rigid, but sprung to the throws about the periphery to avoid reducing the natural torsional frequency of the free crankshaft. Less costly metal could be used for crankshafts.

The development of a connecting member so designed as to be satisfactory from the viewpoint of vibration technique is primarily a question of construction and choice of suitable springing material. Next to the familiar metal springs for torsional and bending suspension, a combination suspension of metal and rubber joined through adhesion seems under the circumstances to offer most promise. Research in this direction is at present under way at the German Institute for Aeronautical Research.

In driving systems with reduction gearing the introduction of the springing in the gears is under consideration (Fiat A 30); in direct drive the springing will be most appropriately placed in the propeller hub (Carter hub). In this connection it should be noted that the sprung connecting member should be designed in advance in accordance with the vibration characteristics of the entire driving system and developed along with it. The supplementary introduction of such a device into a driving system already fully developed and subject to vibration always will be beset with difficulties from the viewpoint both of vibration technique and construction.

The vibration characteristics and the elimination of dangerous conditions of torsional vibration otherwise present in the driving system by means of a flexible propeller hub for a four-cylinder engine are shown in Fig. 19. This hub employs torque arms and tubes for torsional springing. Outside the limits of the questions under discussion, it might be mentioned that this hub with suitable torque arms also may be used as a measuring instrument for torsional moments.

Differing Radial-Engine Vibration Characteristics Analyzed

– W. G. Lundquist
Project Engineer, Wright Aeronautical Corp.

MY principal issue with Mr. Lürenbaum is that he discusses the vibration of crankshaft-propeller systems in general but apparently takes no particular cognizance of the fact that the type of powerplant to which the propeller is attached fundamentally controls the problem at present, particularly in regard to the efficacy of damping devices. Aircraft engines can be divided roughly into two camps—radial engines and in-line engines. The crankshaft-vibration characteristics of these two groups differ from each other and hence some of Mr. Lürenbaum's general statements, although true for one type of engine, are misleading for the other type.

For instance, Mr. Lürenbaum states that the principal sources of disturbance causing crankshaft and propeller vibration of all kinds are the torsional harmonics of the engine which are transmitted to the propeller. A little further along he says that the effectiveness of dampers "is in reality limited to damping degrees of freedom and is accompanied by increased weight".

These statements are generally true and constitute a real problem in certain types of in-line engines. With the radial-type engine, however, the advent of a good dynamic vibration absorber in the form of a dynamic counterweight (dynamic damper[a]) has so changed the picture that, although the foregoing statements are still true (except that concerning weight increase), they now describe a fortunate condition in a radial engine and are a cause for rejoicing rather than alarm, because we can now eliminate the one torsional harmonic which ordinarily would annoy the propeller. That radial engines should enjoy such an advantage over in-line engines is due to the fundamental difference in the vibration characteristics of the two types. The explanation is as follows:

A radial engine of one or two rows suffers principally from only one harmonic torque variation which has a frequency equal to the firing frequency—(the major torque variation of the engine)—a definite frequency in cycles per revolution of the crankshaft. It has, therefore, usually only one important critical speed for torsional crankshaft vibration; that is, the major critical, single-node mode of vibration. The next integral multiple of this harmonic has twice its frequency and hence its critical speed is usually down in the idling range where it is negligible. It is also of very small magnitude in engines of nine or more cylinders. The largest torque harmonic introduced by link-rod action is of small relative magnitude and low frequency which puts any critical speed from such a source far above the maximum operating speed of the engine. This state of affairs is taken care of perfectly by one polychronic dynamic vibration absorber which will, at all times, remain synchronous with the one troublesome major harmonic. The dynamic-damper counterweight does just that and, hence, eliminates the effect of the troublesome major critical speed. But it also does more than that. Being always synchronous with the major torque variation of the engine and receiving its excitation from the motion of the crankshaft under the influence of this torque, it lags the motion of the crankshaft by 90 deg. at all times and, hence, is constrained continually to absorb the energy of any crankshaft oscillation of that frequency at all speeds. This statement means that not only does it reduce the resonant magnification of torque variation at the critical speed, but also that it has the ability to absorb, at all speeds, the energy input to the crankshaft by the normal torque variation, thus reducing the crankshaft oscillations produced by this harmonic throughout the whole speed range. The amount of this reduction in the so-called normal gas torque variation depends upon the design of the individual engines, but there appears to be no reason why it should not be possible by this means to deliver the power of any radial engine to the propeller with a torque variation as low as ±10 per cent of the mean engine torque regardless of the number of cylinders on the engine. In other words, the dynamic damper will do virtually all of the speed regulation in the engine if it is permitted to do so. Incidentally, the dynamic damper does not add any weight to the engine as Mr. Lürenbaum charges, but merely puts some of the weight already there to work.

In comparison to this performance, a line engine not only suffers from a major harmonic as does the radial engine, but it also suffers from a whole series of minor harmonic torque variations which, because they are not all applied at one point along the crankshaft, combine to put energy into the vibration of the shaft although their algebraic sum for the whole shaft is zero. The line engine therefore has one major critical speed and in addition a series of minor critical speeds, all of the same frequency in cycles per second however, but occurring at different engine speeds. An isochronous dynamic vibration absorber (spring actuated) will eliminate the serious effects of all these criticals but may give trouble between criticals. It is also difficult to build a cheap and reliable spring-actuated damper which does not have undesirable friction characteristics that limit the effectiveness of the device. The friction-type damper (Lanchester type) will reduce the severity of all the critical speeds but adds weight. It is not as effective as a good vibration absorber.

[a] See S.A.E. Transactions, March, 1936, pp. 81-89; "Eliminating Crankshaft Torsional Vibration in Radial-Aircraft Engines", by E. S. Taylor.

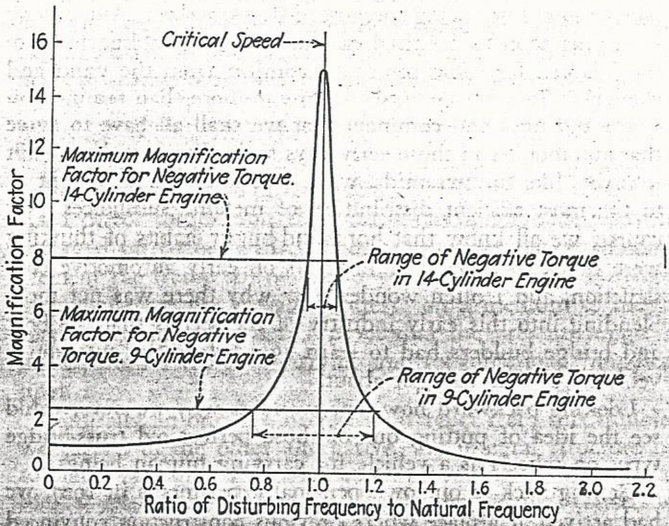

Fig. A – (Lürenbaum Discussion) – Radial-Engine Resonance Curves

Again, Mr. Lürenbaum goes to considerable trouble to calculate the energy input to a vibration as the first step toward calculating the amplitude of vibration at a critical speed. Here again the type of engine under consideration affects the problem. Energy-input calculations are useful from an academic standpoint in the study of any engine type to assist us in getting a sound comprehension of what occurs during critical speeds, but they give us no quantitative answer as regards vibration amplitudes because we do not know how much work the damping forces do, as Mr. Lürenbaum admits. Energy calculations are of more assistance in the study of in-line engines than in the study of radial engines because they are useful principally in determining the effect on torsional crankshaft vibration of firing orders and crank arrangements in in-line engines. Radial engines are simpler in this respect and are, therefore, susceptible to adequate analysis without the use of energy calculations. With either type of engine, however, if we want to estimate amplitudes of vibration, we have to rely on our experience based on torsiograph studies. Fortunately, as Mr. Lürenbaum points out, there are instruments available now to measure these amplitudes for us directly which instruments, particularly in the case of radial engines, give us the data for estimating amplitudes quickly and accurately enough for design purposes and with no energy-input calculations required.

For instance, we know from torsiograph studies that, at a critical speed for torsional crankshaft vibration, a conventional radial engine without a damper will develop a torque variation in the propeller shaft 10 to 20 times the normal torque variation of the engine. In other words, the magnification factor for this type of engine lies between 10 and 20. Knowing then the torque variation imposed by any particular harmonic (from a harmonic analysis) we merely multiply it by 10 or 20 and then add the result to the normal peak torque minus the harmonic under consideration to obtain the limiting values of the maximum torque imposed on the propeller shaft at a critical speed, the values obtained with the magnification factor 10 being the best we can hope for and those with the factor 20 being the worst we are ever likely to get. Incidentally, we have found that a magnification factor of about 15 comes fairly close to being the correct value for engines of this type. From these torque values, we can estimate our stress conditions and govern ourselves accordingly.

Knowing the magnification factor, we also can obtain a value for the overall damping factor of the engine from the formula,

$$\text{Magnification factor} = K = \frac{1}{\sqrt{(1-X^2)^2 + \Delta^2 X^2}}$$

where Δ = Damping factor

and

$$X = \frac{\text{Disturbing force frequency}}{\text{Natural vibration frequency}}$$

At synchronism, this value becomes $K = \dfrac{1}{\Delta}$

From this formula we can plot the resonance curve of the engine under consideration as shown on Fig. *A*. This resonance curve not only shows us what to expect at the critical speed but also shows how close to a critical speed we can operate without exceeding some predetermined stress condition in the engine. For example, we know that negative torque in the propeller-drive train is very injurious to reduction gears, propeller splines, and so on. In order to locate this point on the resonance curve, we merely divide the minimum engine torque (after removing the harmonic under consideration) by the amplitude of the harmonic under consideration and obtain a value for maximum permissible magnification factor. We then draw a line on the resonance curve denoting this factor and note that the zone between the two points where this line cuts the resonance curve is the prohibited operating range. Typical examples are shown on Fig. *A* for a nine- and a fourteen-cylinder radial. Notice that a fourteen-cylinder engine can operate much closer to a critical speed than a nine-cylinder engine can. The use of a dynamic vibration absorber with a radial engine, of course, completely changes this resonance picture as discussed earlier.

The same type of analysis is usable for line engines in the case of critical speeds where the torque harmonics are directly additive for all cylinders, notably the major critical. The minor critical speeds are more difficult to handle, and very often they are the most troublesome offenders.

I believe that it is apparent from all these considerations that the engine type affects the problem of crankshaft and propeller vibration so basically that it must be recognized in any discussion of the subject.