The Liberty Engine and Torsional Vibration

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July 17, 2008

1.0 Preface

When examining the engineering decisions of the aircraft engine designers of the past it is necessary to bear in mind the many gaps in the understanding of engine-related phenomena with which they had to cope. These gaps were due to a lack of information that analytical and experimental techniques would eventually yield and included such topics as detonation, the thermodynamic properties of fuel-air mixtures and their products of combustion, fatigue of metals, the inability to get accurate indicator cards from engine cylinders, and the prediction and amelioration of the effects of mechanical vibration, particularly torsional vibration. Despite these gaps in their knowledge, the engine designers could not be too conservative or aircraft would never have left the ground. Thus intuition and artistry played an important role in the early development of aircraft engines, and subsequent demands in this field advanced all of the mechanical engineering disciplines more than any other single mechanical device.

Torsional vibration problems occurred early and often in the history of aircraft engines. One of the earliest, the Manley-Balzer, apparently exhibited the effects of running at a critical speed when coupled to the shaft and bevel gear propeller drive system of Langley's aerodrome (1). Manley seems not to have recognized the problem as due to torsional vibration since he ascribed the heating and yielding observed in the drive shafts to "reverse torque" and appears not to have considered reducing the stiffness of the drive system to change the natural frequency, but rather added mass in the form of flywheels, thereby adding about 10% to the weight of the engine. The infamous A.B.C. Dragonfly radial engine of 1917 was inadvertently designed to deliver its rated power at a critical speed so that crankshaft life was a matter of a few hours. This could have had serious consequences for the allied war effort if World War I had continued into 1919 (2). In 1929 the Graf Zeppelin left for the U.S.A. from Europe but was forced to turn back after four of its five engines suffered crankshaft failures. These engines were equipped with flexible couplings and vibration dampers, both of which were improperly applied (3).

The obstacles to recognizing, understanding, and controlling the problem of torsional vibration in aircraft engines were numerous and sometimes obscure. In many of the early designs engine life was so short that the fatigue life of the crankshaft was unlikely to be an issue unless, as was the case with the Dragonfly, it was extremely short. When confronted with a broken crankshaft the designer might simply conclude that the shaft was not sized properly for the amount of torque it was being asked to carry based only on gas pressure (which he was probably guessing at) and inertia forces and neglecting vibratory loads. If the shaft was then strengthened and (if he was lucky) the problem disappeared, he would likely assume his diagnosis was correct and would probably ignore the possibility that the stiffer crankshaft had moved a critical speed out of the operating range. With the real problem eventually recognized the analytical tools at his disposal were inadequate to allow the designer to construct an accurate mass-elastic model of the crankshaft-propeller system from which to predict natural frequencies and, if he did attempt to construct such a model, no instrumentation was available to allow him to verify his numbers. By the mid-to-late 1920s enough stiffness tests had been done on crankshafts to allow the construction of mass-elastic models, and high-speed torsiographs were becoming available. There was also a lack of information regarding the magnitude of the exciting torques for the numerous orders of vibration as well as the amount of damping present in the engine. It was not until the mid-1930s that theory and instrumentation allowed fairly accurate prediction of torsional vibration amplitudes, but by then the introduction of metal, variable-pitch propellers had complicated the problem enormously due to the interaction of complex modes of propeller vibration with engine torsionals. The history of the understanding and control of torsional vibration from the 1890s onward is given a good summary in the introduction to W. Ker Wilson's five-volume work on the subject (4).

While surveying the subject of torsional vibration in aircraft piston engines I was struck by the use of a 45° bank angle in the twelve-cylinder Liberty engine rather than the 60° of most other successful V-12s and began to wonder what effects, if any, this had on the torsional vibration characteristics of that engine. The remainder of this paper summarizes my findings.

2.0 Introduction

The Liberty engine was designed in a Washington, D.C. hotel room in late May and early June of 1917 at the behest of the Aircraft Production Board, an agency of the U.S. government. The designers were E.J. Hall of the Hall-Scott Motor Car Co. and J.G. Vincent, vice president for engineering at the Packard Motor Car Co. Both of these firms had developed or were developing aircraft engines at the time Hall and Vincent were recruited. Because this was a government project from design through production it received more comment and criticism than it would have as a private development effort. The political and economic history of the engine is dealt with in some detail by Dickey (5). I have not seen a complete engineering history of this very long-lived engine which would deal with the influence of its immediate antecedents at Hall-Scott and Packard on its design, the improvements made to its bearings, lubricating system, and cooling system, or the many versions of the L-12 which appeared in an attempt to utilize the large number of surplus engines available (geared, turbocharged, air-cooled, inverted, etc.). Dickey's technical discussion of the engine is pretty much limited to Vincent's response to various criticisms of the engine at the time it was designed, and he offers no analysis based on modern hindsight.

In this paper I am considering only the effect of the choice of the 45° bank angle on torsional vibrations, and Dickey makes it quite clear there was controversy over that choice at the time. Less than a year after the design of the Liberty, in May 1918, one Leon Cammen delivered a lecture to the Aeronautical Society of America titled "Criticism of the Liberty Engine." Dickey quotes Cammen as follows: "From a general knowledge of balancing, it would appear vibration would occur at 1450-1550 and again above 2000." This statement is somewhat misleading, since "balance" in an engine usually refers to the effects of reciprocating and rotating inertias and is not a term one uses when speaking of torsional vibrations, but reference to two distinct speeds at which vibration is predicted to occur could make one think he was referring to torsional vibration. Vincent's response (again, quoted by Dickey) in a letter of Dec. 1918 states that "the 45° arrangement … reduced synchronous vibration of the crankshaft, due to breaking up of the evenly spaced intervals," leaves little doubt that he, at least, was talking about torsional vibration. The "breaking up of evenly spaced intervals" refers to the uneven firing order the bank angle and crankshaft configuration mandated. According to Vincent, his primary reason for selecting the 45° bank angle was to reduce the engine's frontal area and hence its drag when installed in an aircraft. This argument is somewhat incomplete since the frontal area of an engine of a given displacement is a function of a number of variables besides the bank angle; including the ratios of stroke to bore, piston length to bore and connecting rod length to stroke. All of these ratios are on the high side for the Liberty design as compared to other aircraft engines of that era, particularly the German engines. The U.S. government's mandate to Hall and Vincent, that their design be restricted to tried and proven design features, apparently did not require them to adhere to any particular values of these ratios so, given that fact and the very short design time, no real attempt could have been made to minimize engine frontal area. I suppose that, confronted by my argument, Vincent and Hall would simply respond that those values of the design ratios were what were "tried and proven" in their experience.

Table 1
Liberty Engine Characteristics

Arrangement	12 cylinder V 45° bank angle
Bore	5.000"
Stroke	7.000"
Displacement	1,649.3 in^3
Displacement	137.4 in^3/cylinder
Rating	420 bhp @ 1,700 rpm
Brake Mean Effective Pressure	119 psi
Rotating weight	6.3 lb/crankpin
Reciprocating weight	12.4 lb/crankpin

The comments by Cammen and Vincent in 1918 as quoted by Dickey were what aroused my interest in looking into the question of the 45° bank angle's influence on the torsional characteristics of the Liberty L-12 and they will be referred to again as the analysis progresses. The results of actual torsional vibration testing on the L-12 in the late 1920s and early 1930s will be compared to my analysis as well.

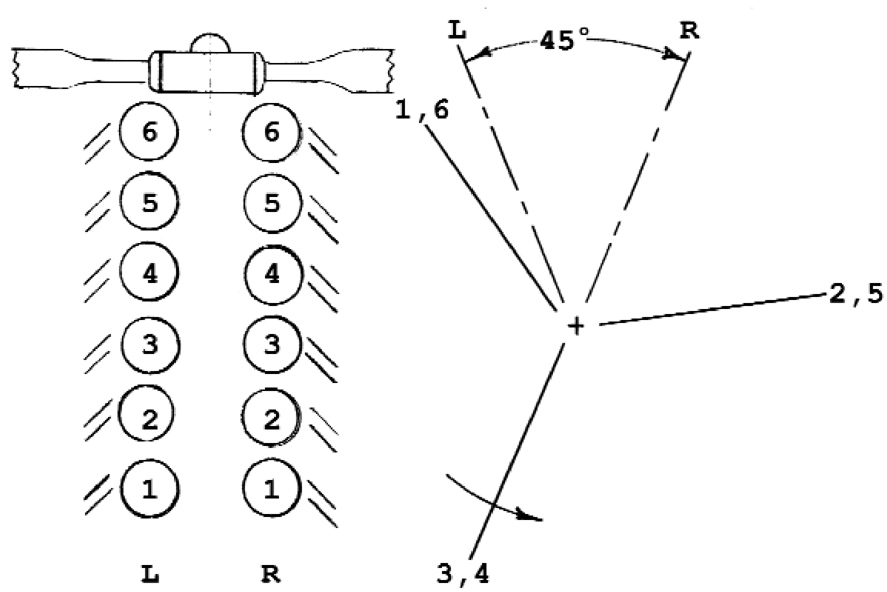
3.0 Torsional Analysis of the Liberty L-12

3.1 Introduction

The relevant characteristics of the L-12 appear in Table 1. The rotating and reciprocating inertias are included because they influence the harmonic components of engine torque (reciprocating only) as well as the mass-elastic model for calculating the natural frequency of the crankshaft system. The firing order and crankshaft arrangement are shown in Figure 1. The Liberty crankshaft is a conventional six-throw symmetric arrangement used in most in-line six and V-12 engines. This crank provides completely balanced inertia forces and moments for each bank of six cylinders and hence for any V-12, regardless of bank angle. Counterweights, if provided, are solely for the purpose of reducing main bearing loads and have no impact on balance if properly arranged. The L-12 has no counterweights. Interchanging the four intermediate crank throws changes the firing order but not the torsional characteristics of the crankshaft.

3.2 Phase Vector Sums

A crucial insight into the analysis of torsional vibration in engines is provided by the adoption of a Fourier series to represent the instantaneous torque versus crank angle over an engine cylinder's full cycle. A series of sine waves of various amplitudes, frequencies and phase angles relative to the top dead center firing position of the cylinder are summed to represent the actual torque due to pressure and



Crank Angle	0°	75°	120°	195°	240°	315°	360°	435°	480°	555°	600°	675°	720°
Firing Order	1L	2R	5L	4R	3L	1R	6L	5R	2L	3R	4L	6R	1L
Firing Interval	75°	45°	75°	45°	75°	45°	75°	45°	75°	45°	75°	45°	

Fig. 1. Liberty 12 Cylinder and Crankshaft Arrangement

the inertia of the reciprocating parts. The frequency of each sine wave is an integer multiple of the number of cycles and, since there is one cycle in two revolutions of a four-stroke engine, the frequencies are ½, 1, 1½, 2, 2½, … etc. multiples of the engine speed. These are referred to as the "order" of a particular exciting frequency. The amplitude of each of these orders is a function of the mean effective pressure (mep), compression ratio, fuel/air ratio and spark advance with the mep being the most significant. The lower the order number, the larger the amplitude down to the 1st order, the ½ order being somewhat less. Usually the torque curve is defined with sufficient accuracy by neglecting orders beyond the 12th. The reason this method is so valuable is that at a resonant speed, only the order in phase with that frequency does work on the crankshaft. With knowledge of the forcing torque for that particular order and a measure of the amplitude of torsional displacement at some point in the crankshaft, the damping inherent in the engine can be calculated. With the damping factor known for a particular engine or design the amplitude of vibration and therefore the stress in the crankshaft can be calculated for any operating condition. With no damping, the amplitude of vibration at a resonant speed is infinite and the crankshaft would obviously break, thus the importance of knowing how much damping is present.

The problem now is to look at how the various orders interact with each other for all of the cylinders in the engine. Some will add, the so-called "major" orders, and some will, until their vectors are combined with the mass-elastic diagrams, appear to cancel each other. These are referred to as the "minor" orders. How they behave is obviously a function of the crankshaft layout and firing order. We will start with one bank of the Liberty, say the left bank (see Figure 1), where the firing order is 1-5-3-6-2-4, equally spaced at 120°. Adding all of the orders vectorially, from ½ to 12, results in the phase vector diagrams shown in Figure 2, giving one major order diagram and three minor order diagrams. We can see from Figure 2 that orders 3, 6, 9, and 12 are major orders, which means that those harmonics of all six cylinders add (the vectors representing all six cylinders point in the same direction). The numbers spaced around the periphery of the circles indicate the direction of the phase vectors for that cylinder corresponding to the order number shown below the circle. At this stage all the vectors for each order are assumed to have unit values; we

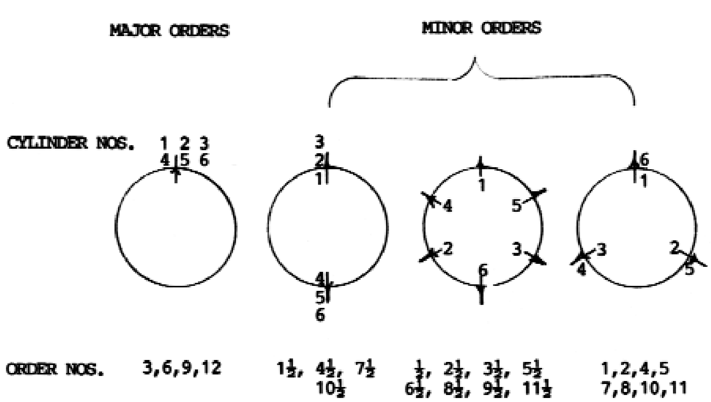


Fig. 2. Phase Vector Diagrams for One Liberty 12 Bank (both banks are the same)

will assign relative values to each torque harmonic at a later stage.

The next step in the analysis is to combine the vectors from the other (right) bank, which has the same firing order, to get the resultant phase vector diagrams for the entire engine. These are presented in Table 2 for the two bank angles 45° and 60°. The unit vectors for one bank, Figure 2, have combined to give values between zero and 2 for orders 3 to 6. The most significant item in this tabulation with regard to our initial question is that the 45° bank angle has re-introduced the 3rd major critical order at a value of 77% of what it would have been in a six-cylinder engine, while the 60° bank angle would eliminate the 3rd order entirely. This means that the L-12, designed to operate at about 1,800 rpm, would have had a major critical in its operating range if the natural frequency of the crankshaft assembly vibrating against the propeller had a natural frequency of 5,400 (3 x 1,800 rpm) vibrations per minute (vpm) or less.

The other point of interest in Table 2 is that the 6th order major critical is significantly reduced for the 45° bank angle. Recalling Vincent's response to criticism of his choice of bank angle ("…reduced synchronous vibration of the crankshaft due to breaking up of the evenly spaced intervals…"), one wonders if this was based on his intuition, or did he actually do some vector sums? Without the insight of the Fourier analysis, engine designers of his era probably thought no further than the number of firing impulses per revolution (six in this case) as sources of torsional excitation, so I doubt if he recognized anything at three times engine speed as being significant, though it seems that if the 6th is "broken up" might not the 3rd "come back" should have occurred to him.

3.3 Natural Frequencies

In order to determine the relative importance of the various orders (major and minor), it is necessary to know the frequencies at which the crankshaft/propeller system can vibrate and how the various elements in that system deflect with respect to each other at those frequencies. This is accomplished by pretending the crankshaft and its associated connecting rods and pistons can be treated as a series of flywheels representing the inertia at each crank throw

Table 2
Resultant Phase Vectors for Two Banks with Bank Angles of 45° and 60°

	RESULTANT VECTOR*	
ORDER #	**45°**	**60°**
3	0.77	0
3½	1.96	1.93
4	0	1.0
4½	1.96	1.41
5	0.77	1.73
5½	1.66	0.52
6	1.41	2.0

*The magnitude is based on an arbitrary value of 1 for each cylinder of one bank. See Figure 2 for vector orientation.

connected by shafts of a stiffness which, taken together, represent the overall stiffness of the crankshaft, or the amount the shaft would twist if fixed at one end and a torque were applied to the free end. This is known as a mass-elastic diagram, and one of these for the L-12 is shown in Figure 3. This diagram was derived from the dimensions of the L-12 crankshaft shown in Figure 4. The inertia of the crankshaft elements were calculated using standard engineering techniques, to which were added all of the rotating inertia of the connecting rod big ends and one-half the total reciprocating inertia per crankpin. I used the B.I.C.E.R.A. (6) Provisional Formula to calculate the stiffness of the crankshaft elements. Since all the crank throws are similar for the L-12, one need calculate only the inertia and stiffness of a single throw. The effective stiffness from the last crank throw to the propeller was also calculated with information from reference (6). The calculated value gave a 1-node natural frequency about 5% higher than the observed value, so the stiffness of this element was reduced slightly to bring the numbers into agreement.

At the time the L-12 was designed, the technique just outlined was not in use, and the most that might have been done would have been to represent the system as a two-mass (flywheel) system, one representing the engine and the other the propeller. This could have worked for the major orders (3, 6, 9, etc.) if the effective stiffness of the crankshaft were known. By the mid-1920s enough crankshafts had been tested in torsion to make the two-mass technique valuable for predicting the fundamental natural frequency.

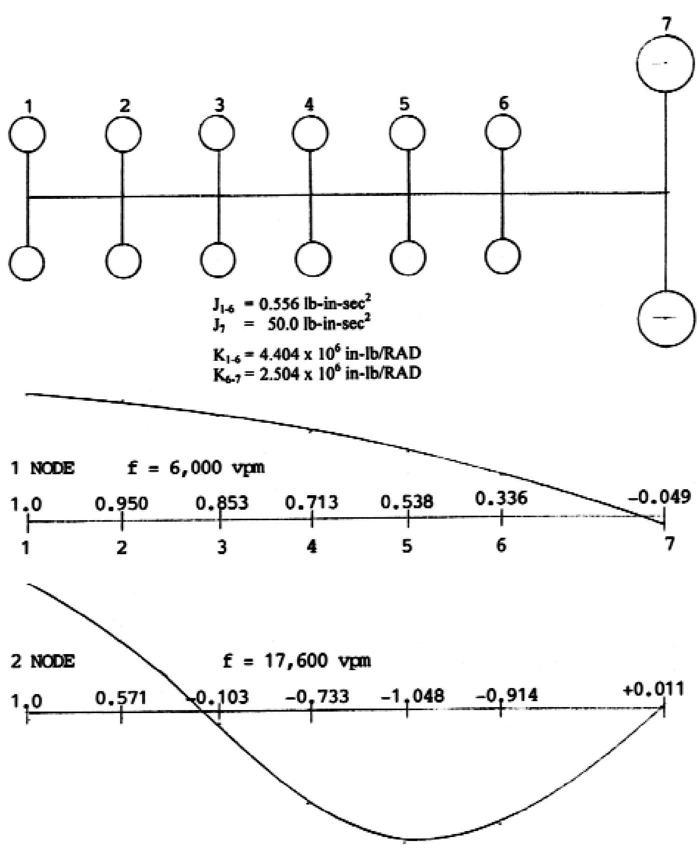


Fig. 3. Liberty 12 Mass-Elastic Diagram and Relative Deflections at Resonance

With seven masses in the mass-elastic diagram, there are obviously many natural frequencies at which the crankshaft can vibrate simultaneously (like a violin string). Typically, only the lowest two are important in reciprocating engines, especially the low speed engines of the World War I era. I used the Holzer method to find the two lowest natural frequencies for the mass-elastic system, and these are shown in Figure 3 as well as the *relative* twist between elements of the crank and propeller. The lowest frequency, which can be thought of as the engine vibrating against the propeller, is referred to as the 1-node mode, the node being the place in the mass-elastic model where there is no angular twist, and the stress due to torsional vibration is a maximum. Note that the node is very close to the propeller due to the relatively high inertia of that element. Assuming the propeller inertia to be infinite would not change the 1-node frequency or relative deflections very much. The next higher frequency is known as the 2-node mode, and here one of the nodes is in the middle of the engine while the second is, again, close to the propeller. Looking at Figure 2, it is clear that the phase angle diagram for orders 1½, 4½, 7½ and 10½ could lead to high exciting torques for the 2-node mode since the vector sums for cranks 1, 2 and 3 are exactly opposed to cranks 4, 5 and 6. For a 2-node frequency of 17,600 vpm, only the 10½ order is in the operating range of the L-12.

At this point we have the vector diagrams for the various orders (Figure 2), their relative magnitudes for 45° and 60° bank angles (Table 2) and the *relative* amplitudes of vibration at the six crankpins of the engine (figure 3). Now we have to do another vector summation. The values of the relative exciting torques for each order (Table 2) must be multiplied by the relative deflection of each crankpin (Figure 3) because the product of torque and angular deflection is the work input of that order at resonance. The resulting vectors are then added vectorially per the diagrams of Figure 2. For example, the major orders (3, 6, 9, 12) are obtained as follows, from Figure 3:

$$\overrightarrow{\Sigma\Delta} = 1.0 + 0.95 + 0.853 + 0.713 + 0.538 + 0.336$$

$$\overrightarrow{\Sigma\Delta} = 4.39$$

and the resultant vectors are this value multiplied by the values in Table 2, for the 45° bank angle:

3^{rd} order — (0.77) (4.39) = 3.38 = $\overrightarrow{\Sigma\Delta}_3$

6^{th} order — (1.41) (4.39) = 6.19 = $\overrightarrow{\Sigma\Delta}_6$

The resultant vectors for all of the orders for the 1-node frequency of vibration are given in Table 3.

We still are dealing with relative values of torque, so the next step is to give them real values. I have used coefficients for the gas pressure torque from reference (7). These were obtained from a 4-stroke spark ignition CFR engine for various values of the variables already mentioned. Since our purpose here is to compare the effect of two bank angles, these values should be sufficiently representative. We will only be considering orders down to 3, so that is the only order that needs a term added for the effect of reciprocating inertia on the torque (the contribution above order 3 is negligible). Without going into the details, the mean torque of one cylinder of the L-12 at an IMEP of 115 psi is

1,258 inch-pounds. The 3rd order vibratory torque due to only inertia is 1,900 inch-pounds at 2,000 rpm. This is balanced somewhat by the gas pressure torque coefficient to give a 3rd order exciting torque of 994 inch-pounds. The important point is that the value of the inertia component is enough to cause trouble even if the engine were not producing any torque, as might happen in a dive with a fixed-pitch propeller. In the case of the L-12, this could happen at 2,000 rpm, since the 1-node natural frequency is 6,000 vpm (Figure 3).

All of the foregoing is summarized in Table 3 for the 1-node frequency. The columns in this table represent the order and corresponding critical speed (columns 1 and 7), the dimensionless vibratory torque per cylinder for the order (column 2), the phase vector sums for all of the cylinders for 45° and 60° bank angles (columns 3 and 4), and the resulting excitation torque for the two bank angles (columns 5 and 6). These results were used to estimate the half amplitude of vibration at the anti-propeller end of the crankshaft, assuming a mean torque per cylinder of 1,250 inch-pounds and a magnification factor at resonance of 15. The results of this analysis are shown in Figure 5. The question of the effect of bank angle is easily interpreted. The 45° bank angle results in higher vibratory torque for all but the 6th order, which is at a low enough speed that the mean torque is certain to be low for any relevant propeller. The most dangerous effect of the 45° bank angle is to reintroduce the 3rd order at just beyond the operating range of the engine. I have attempted to estimate the so-called "flank" values of the vibratory torque to show how the 3rd order flank extends down into the operating range. The danger of windmilling at 2,000 rpm has already been mentioned. The only 2-node order of any significance is the 10½ at 1,676 rpm (not shown in Figure 5), which is also slightly higher for the 45° bank angle. Two-node orders would become increasingly important in the Allison V-1710 and Merlin engines where crankshafts were stiffer and speeds and mep's were much higher.

Table 3

Harmonic Torque Components, Phase Vector Sums and Resultant Vibratory Torque on the Liberty 12 Crankshaft, 45° & 60° Bank Angles

One node — f = 6,000 vpm

(1) n	(2) $\frac{T_{vn}}{T_m}$	(3) 45° $\vec{\Sigma\Delta}_n$	(4) 60° $\vec{\Sigma\Delta}_n$	(5) 45° $\frac{T_{vn}}{T_m}\vec{\Sigma\Delta}_n$	(6) 60° $\frac{T_{vn}}{T_m}\vec{\Sigma\Delta}_n$	(7) Ncr (rpm)
3	0.79	3.38	0	2.67	0	2000
3 ½	0.53	0.89	0.88	0.47	0.47	1714
4	0.40	0	0.20	0	0.08	1500
4 ½	0.31	2.39	1.72	0.74	0.53	1333
5	0.25	0.15	0.35	0.04	0.09	1200
5 ½	0.20	0.75	0.24	0.15	0.05	1091
6	0.17	6.19	8.78	1.05	1.49	1000

Nomenclature:
Tvn = harmonic vibratory torque for order "n" for one cylinder
Tm = mean torque for one cylinder
$\vec{\Sigma\Delta}_n$ = phase vector sum for all cylinders adjusted for relative deflections
Ncr = critical speed
n = order number

Even with the information of Figure 5, absolute vibratory stresses cannot be calculated because I have used an *assumed* value of damping implicit in the magnification factor. With no damping all of the critical speeds shown would cause the crankshaft to fail. With a measured value of vibration amplitude at a resonance condition at any point in the crankshaft (but as far from a node as possible, for accuracy), the actual damping can be calculated and the vibratory stress at any condition can be determined. As we have seen, Vincent appears to have had only one insight with regard to Figure 5, that the 6th order would be reduced for the 45° bank angle. He apparently had no idea of the 1-node natural frequency or that his bank angle re-introduced the 3rd order (present in all six-cylinder engines) dangerously close to his proposed operating speed. I think we have to conclude that Hall and Vincent were simply lucky that the 1-node frequency was just high enough to give them a safe operating window. Without the 3rd order introduced by the 45° bank angle the window would have been much wider and the 1-node frequency could have been reduced by introducing some flexibility in the portion of the crankshaft between the crank and propeller if the 3½ and 4½ orders were a problem.

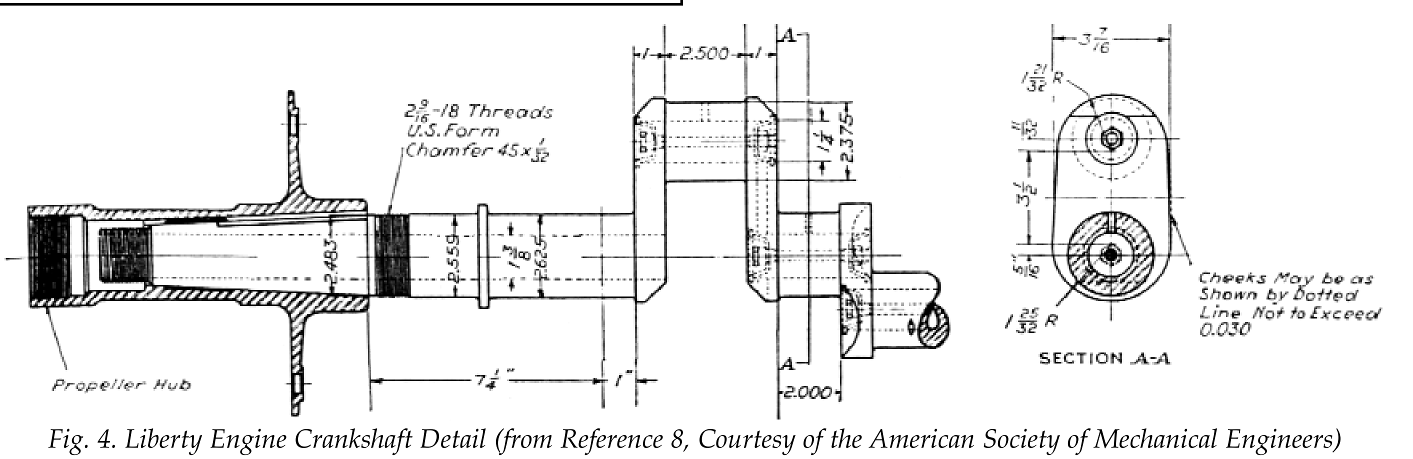


Fig. 4. Liberty Engine Crankshaft Detail (from Reference 8, Courtesy of the American Society of Mechanical Engineers)

I have discovered two technical papers from around 1930 that deal with testing the Liberty 12 for torsional vibrations. The remainder of this paper will discuss these results with reference to the foregoing analysis.

4.0 Torsional Test Results for the L-12

A paper (8) presented in May 1930 by Ford Prescott indicates that the Air Corps' Power Plant Branch began the study of torsional vibrations in aircraft engines in 1927 at McCook Field. The thrust of his paper is to validate the results of tests with a torsiograph developed at McCook, which appears to have been a variation on the Geiger device used to analyze the Graf Zeppelin engine failures. Prescott tested a number of engines, including the Curtiss D-12 and V-1570, the Pratt & Whitney R-1340, as well as the Liberty L-12. The natural frequency of the L-12 crank-propeller system was found to be 6,000 vibrations per minute (vpm) with critical speeds at 1,000 (6th order), 1,333 (4½ order), and 1,715 (3 ½ order) rpm. These results and a glance at Figure 5 indicates the reason for Prescott's comment to the effect, "Because of the smoothness of operation of the Liberty 12 at 1,450 to 1,550 rpm, the pilot prefers to use this speed range at the expense of output." Prescott also notes that crankshaft stress calculations based on gas pressure and inertia torque do not "reveal any marked weakness" but "there are speeds within the operating range where crankshaft failures occur." He goes on to state that the crankshaft "almost always breaks at one of the [crank] cheeks between 5 and 6 [crankpins] instead of the last cheek." If vibratory loads are ignored, this is not where one would expect the crankshaft to fail.

Prescott makes no attempt at the type of analysis I have presented in section 3, but calculates a 1-node natural frequency based on a two-mass model developed by Timoshenko (9) which gives a natural frequency which agrees fairly well with the observed value. As already discussed, the two-mass model would not predict the minor orders (4½ & 3½) observed in the testing, only the major orders (3 & 6). There is some criticism of this approach by Den Hartog in the discussion section of the paper, but neither Prescott nor any of the discussers mention the fact that the 3rd order is present or attribute any of the vibration observed at 1,715 rpm to that order. One of the discussers asks if any 2-node frequencies were detected and the author responds in the negative.

Prescott also mentions that supercharged versions of the L-12 could be equipped with a propeller which pulled the sea-level speed of the engine down to the 1,350-1,400 rpm range which caused it to develop maximum power while running at or near the 4 ½ critical. He does not indicate

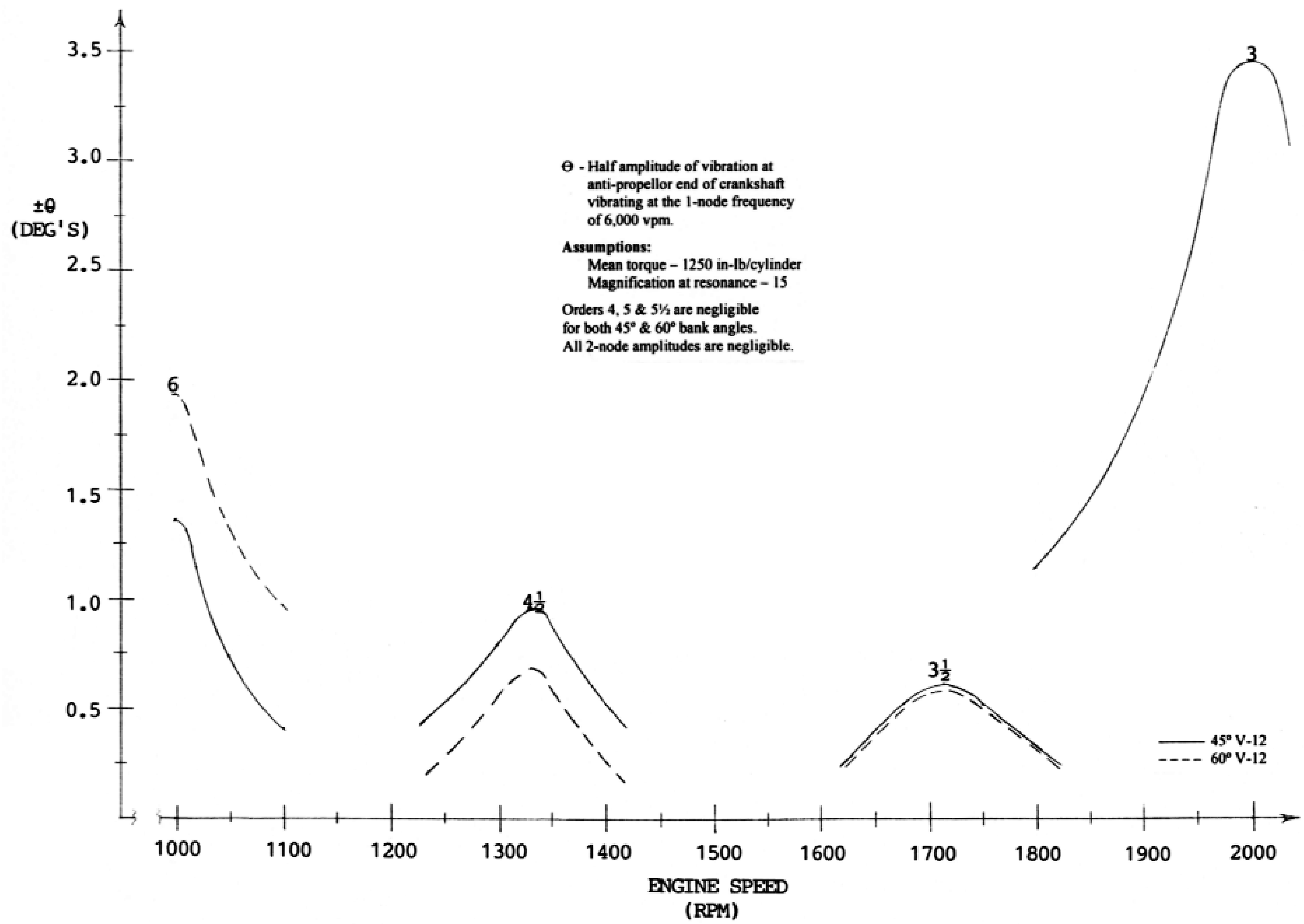


Fig. 5. Liberty 12 Estimated Half Amplitude of Vibration at Resonance Versus Engine Speed for Orders 3 to 6, 45° and 60° Bank Angles

which of the critical speeds (1,333, 1,715, or overspeeds to 2,000 rpm) were the principal cause of crankshaft failures.

A second paper dealing with torsional vibration in the Liberty engine appeared in 1933 and was the result of work carried out by the Royal Aircraft Establishment (R.A.E.) in Great Britain. Authored by Carter and Muir (10), this paper contains a complete analysis of the crank-propeller system but was not discovered by me until after I had completed the calculation of inertias and stiffness given in section 3. Fortunately my results were in reasonable agreement with theirs.

Using a torsiograph developed at the R.A.E. (11) which operated on an entirely different principle from the device used by Prescott, and which measured shaft twist between the final crank throw and the propeller, Carter and Muir found the 1-node natural frequency to be 6,000 vpm, the same result as obtained by Prescott. There is a complete discussion of the effects of the 45° versus 60° bank angle which agrees with my results as presented in Table 2 and Figure 5. Crankshaft failures were in the same location as described by Prescott and are referred to as "an epidemic" by the British authors.

The Carter and Muir paper indicates that an attempt was made by one of them to calculate the natural torsional frequency of the L-12 as early as 1918, and that the 6th order was found to be in the normal speed range. The authors apparently did not take this result too seriously, since "the information regarding crankshaft stiffness was known to be inadequate." They also ignored the 3rd major critical and all of the minor orders in the 1918 analysis.

Figures 6 and 7 show the results of tests at the R.A.E. on the Liberty with four different propellers (airscrews to them). The 4½ and 6th order criticals are obvious with the propeller which gives full power at 1,375 rpm (Fig. 6). These orders are less obvious in Figure 7 where the propeller gives full power at 1,950-2,000 rpm, because the mean torque is considerably less at those critical speeds. The 3½ order and the influence of the flank of the 3rd are very obvious here, and the authors state that failures would occur in the 1,700-2,000 speed range. Figures 6 and 7 agree with the analysis presented in Section 3 and summarized in Figure 5. The apparent 4th order vibration in Figures 6 and 7 is attributed to a mixture distribution problem by the authors. Theoretically, there should be no 4th order (see Table 3). Carter and Muir were able to calculate an absolute torque because their instrument measured the instantaneous twist in the shaft between the last crankpin from the free end and the propeller. Prescott's device measured only the amplitude of vibration at the free end. However, examining the photographic records of Carter and Muir, it is difficult to assess the accuracy of their torque values. There is some obvious distortion in their records which was probably due to bending in the propeller shaft and which, presumably, could be calibrated out. The device they used for the work described here did not stay in use, perhaps because it required modifications of the propeller shaft and

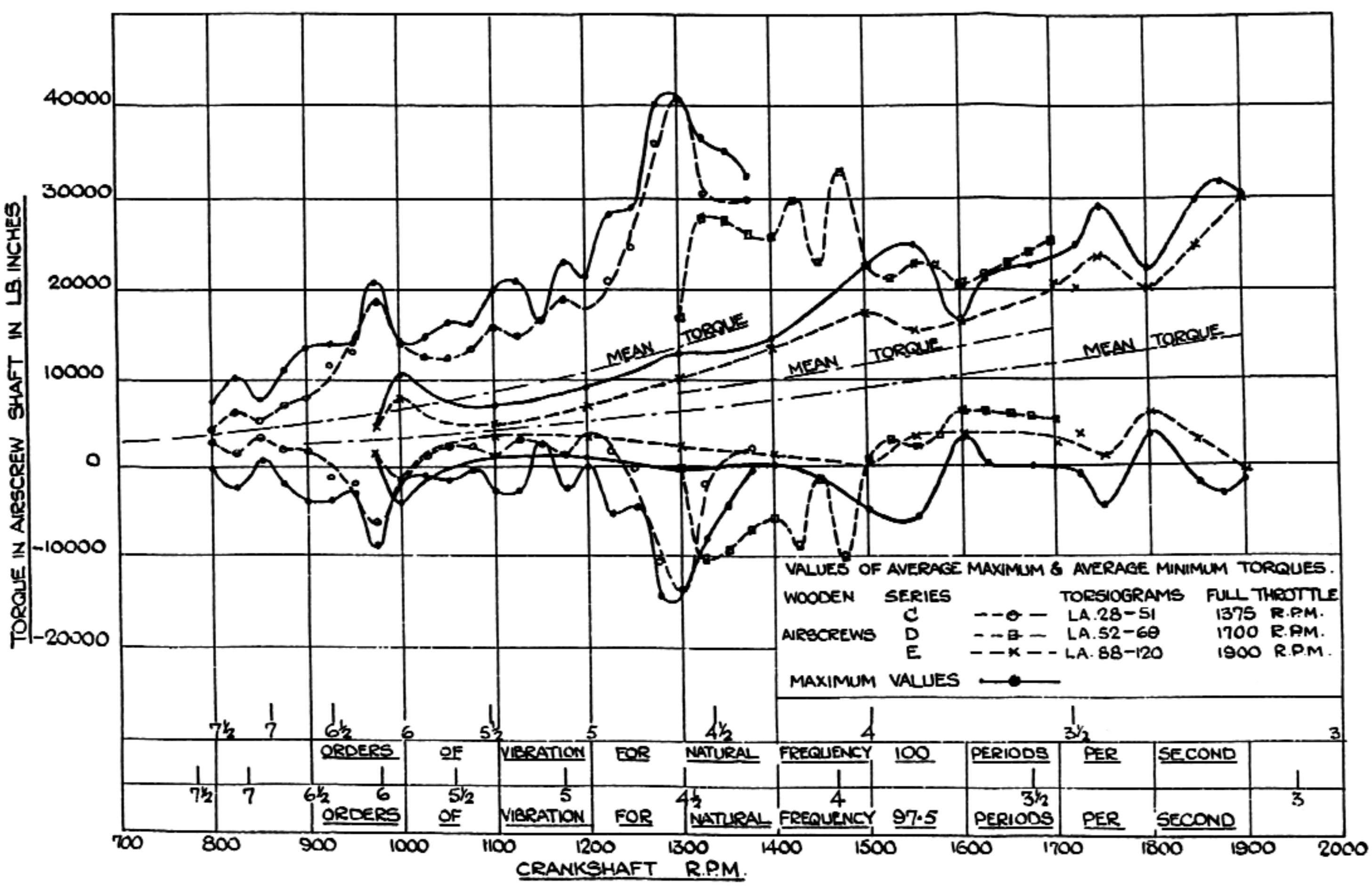


Fig. 6. Liberty 12 Torsional Resonance Curves (R&M 1304)

was almost certainly not adaptable to continuously variable pitch propellers.

In summary, Carter and Muir's work verifies Prescott's findings and extends the analysis to consider a complete mass-elastic model of the system and consideration of the effect of minor orders. They also attempted to determine the damping inherent in the L-12, but a discussion of this is beyond our scope here, since the bank angle should not affect damping. To do this, however, they needed a Fourier analysis of the torque versus crank angle, and this work was performed on a single cylinder Liberty and is reported in (12). As mentioned already, I used a different source for these coefficients because I had not discovered this paper when the analysis was carried out. This is not important because I am comparing the effect of bank angle, so the choice of coefficients is not an important factor. We can see from these two papers the progression in sophistication of torsional analysis, at least in the English-speaking world. From looking at the literature, the Germans may have been somewhat more advanced at an earlier date, but I have not examined this question in any detail.

5.0 Discussion

It is clear that the Liberty, like all other engines of its time, was conceived with little or no consideration given to torsional vibrations. If the crankshaft had been stiffer so as to put the 6th major order up into the operating range of the engine, then Vincent's one known insight with regard to the torsional characteristics of the engine would have resulted in a 30% reduction in the excitation torque for the 45° bank angle as compared to 60°. It is also clear that the L-12 broke crankshafts due to torsional vibrations and this was aggravated by the 45° bank angle. First, it increased the 4½ order torque at 1,333 rpm by almost 40%, and second, it introduced the 3rd major order at 2,000 rpm whose flank, combined with the 3½ order peak at 1,714 rpm could also break crankshafts. I have not examined the record for other V-12 engines of that era, but it is probably safe to say that most of them had operating speeds that the pilots were told to avoid. The Rolls-Royce Eagle, designed in 1914, was eventually equipped with a friction damper and clutch in the reduction gear housing (around 1922, according to Jane's) which would indicate a torsional problem. Locating a damper close to the node (see Figure 3) where the amplitude of vibration is a minimum rather than at the free end

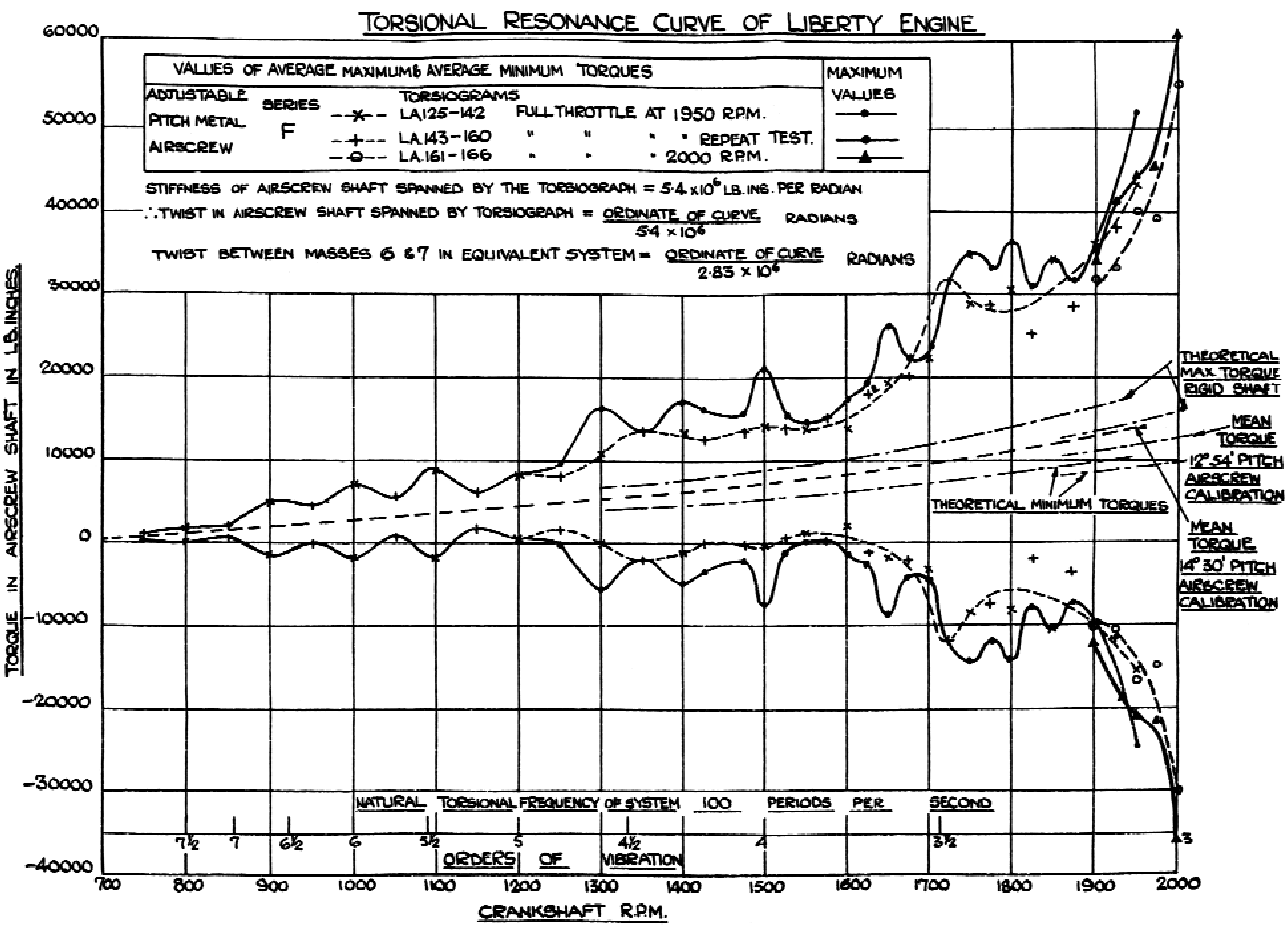


Fig. 7. Liberty 12 Torsional Resonance Curve (R&M 1304)

seems counter-intuitive. A reduction gear normally reduces the stiffness of the crank/propeller system, thereby reducing the 1-node natural frequency. Perhaps the Eagle's 1-node frequency was low enough to cause orders lower than 3 to give trouble.

The Liberty was eventually offered by Allison from the late 1920s to the early 1930s with the bore reduced to 4.625". According to Jane's, the direct drive version was rated at 1,880 rpm and a geared version at 1,900 rpm. The question naturally arises as to how Allison managed the 3rd order problem since they apparently used the Liberty 45° crankcase. Jane's for 1928 states that the geared version had a different crankshaft and crankcase as well as a spring coupling at the propeller hub. Presumably the coupling was soft enough to drop the 1-node frequency to a level where no major orders would be excited, but this does not explain how the direct drive version could be rated at 1,880 rpm without a soft coupling. The reduced bore would have resulted in lower reciprocating inertias, which would have increased the 1-node frequency and, combined with the reduced gas pressure and inertia torques, this might have been sufficient to avoid crankshaft failures.

Packard went on to offer large aircraft engines throughout the 1920s, and they were all 60° V-12s. Jane's lists J.G. Vincent as vice-president of engineering at that firm until at least 1930. It would be interesting to know how he came to change his mind about the 45° bank angle. By the time the Allison V-1710 and the Rolls-Royce Merlin were designed in the early 1930s, no one in the high output aircraft engine business could ignore the subject of torsional vibrations at the design stage. As reference (10) clearly indicates, enough was known to incorporate a complete torsional analysis as part of the design process. This would by no means guarantee an engine free of torsional problems, particularly in in-line engines with their long, flexible crankshafts, but the worst scenarios could at least be avoided. Engines that were relatively free of torsional problems in their early years could develop problems as the rated power was increased and variable pitch metal propellers were introduced, coupling propeller blade vibration with engine torsionals. A great deal of ingenuity went into dealing with these problems, the most significant being the tuned pendulum absorber adopted by all large radial engines and a few in-line engines as well. Perhaps the Liberty with its 45° bank angle, overwhelming numbers and low cost, contributed more to the understanding of torsional problems than it would have had it had a 60° bank angle.

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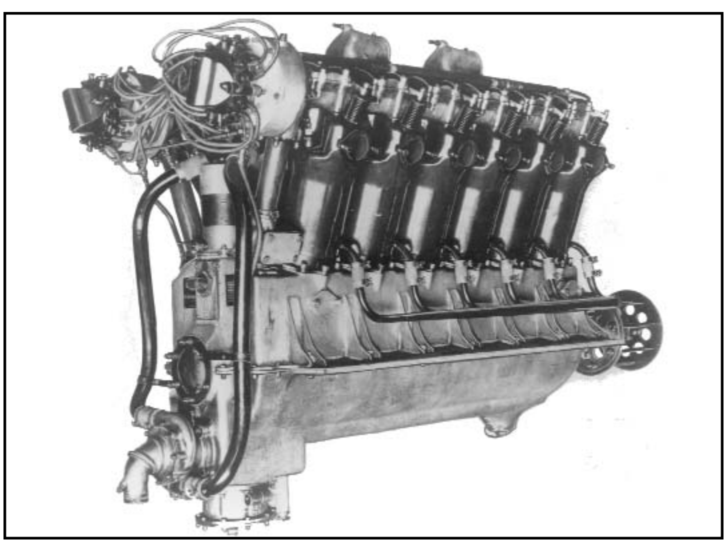

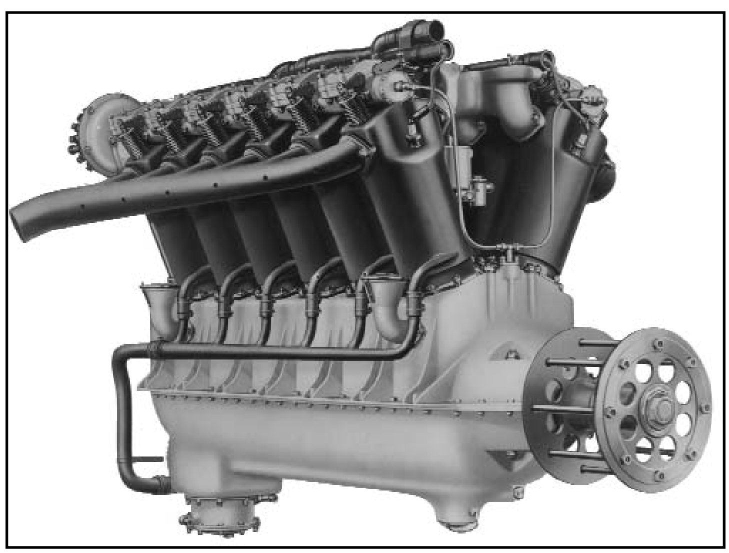

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