An Examination of the Torsional Vibration Characteristics of the Allison V-1710 and Rolls-Royce Merlin Aircraft Engines

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Preface

By 1940, most of the high output radial aircraft engines were utilizing tuned pendulum dampers to minimize torsional vibrations in the crankshaft-propeller system. The situation in the in-line high output configuration was less consistent. While a number of earlier engines had adopted various types of dampers (mainly of the friction type) by 1940 all but one engine, the Allison V-1710, were without dampers of any kind. Rolls-Royce, Daimler-Benz and Junkers, all with liquid cooled V-12s and operating at comparably high outputs, were damper free. The Rolls-Royce Griffon, under development but not yet in service, never employed a damper.

This paper is an attempt to explain why the Allison engine was unique in this respect. To do so, I am using the Merlin to compare the torsional characteristics of the two engines since they are close in displacement with similar bore/stroke ratios. There is nothing in their designs that could lead one to think their vibration damping characteristics might be different. If anything, the larger main and connecting rod bearings in the Allison would indicate more damping and, as I have shown elsewhere, the friction mep of the two engines was probably very similar.

The data needed to carry out this analysis are not entirely complete but more information is available than for any other similar engines. In their published work Allison does not present test data that indicates a need for dampers in any documents available to me nor have I found actual torsiograph data that indicates they are necessary to stay within the limits specified by Army-Navy standards.

Introduction

The subject of torsional vibration in piston engines is difficult to digest in one sitting. Most mechanical engineers are not exposed to the subject either in school or in their work environment but, of course, are familiar with the fundamentals of vibration. The numerous terms — nodes, modes, orders (major and minor), critical speeds, etc. — are confusing when first encountered. I will try to explicate as we go along but the reader may want to look at my paper on the Liberty-12 available on the AEHS web site where the analysis is more detailed than it will be here.

The crankshaft is like a violin string; it can have many *modes* of vibration, the simplest in the case of the violin being a half wave length with a *node* at either end. Superimposed on this fundamental mode are multiple higher frequency modes. Twisting rather than lateral deflection characterizes torsional vibration in a crankshaft and the first two modes are the most important. The first mode can be thought of as the engine assembly vibrating against the propeller with a single node located in the propeller shaft and the maximum angular deflection at the rear of the crankshaft. The second mode is at a much higher frequency and has two nodes, one in the center of the crankshaft and a second in the propeller shaft near the propeller. Maximum deflections are usually at the rear and front of the crankshaft or the gearbox, depending on the relative stiffness of the coupling between the crank and the gearbox. This mode can be thought of as the front and rear halves of the crank vibrating against each other.

Determining the shape of these modes and their associated natural frequencies is the first step in any analysis of torsional vibration in an engine's rotating assembly. This is accomplished by replacing the crank, connecting rods and pistons with a series of flywheels and shafts that represent the inertia and stiffness of the various elements in the system. Through engine tests with torsiographs, which measure the frequency and amplitude of vibration of a crankshaft, and static stiffness tests on crankshafts this technique was developed to give satisfactory results. These models are called mass-elastic diagrams and Figure 1 is an example. The natural frequency is calculated by giving the first inertia a one-radian deflection and estimating a frequency. If the estimate is incorrect there will be a remainder torque at the last inertia and the procedure is repeated until the remainder is zero. This is known as the Holzer method.

Once the two natural frequencies are established the next task is to investigate the forces that excite the vibrations in the crankshaft. It is obvious that the torque due to pressure and inertia varies dramatically over the 720-degree cycle for each cylinder, with about a 20% variation in output torque between the last pair of cylinders and the propeller in a V-12 engine and much more drastic variations as you go back toward the rear of the engine. The magnitude and amplitude of these fluctuations determine the stress level and establish the fatigue limit of the crankshaft. To these values one must add the stress induced by the vibration of the crankshaft. It is intuitively obvious that there are six equally spaced strong pulses in one revolution of the crankshaft so one would expect that an engine speed corresponding to one sixth of a crankshaft's natural frequency might be problematical, which in fact it is. The sixth is a *major order of excitation* in a V-12 engine. The magnitudes of all the other orders, major and minor, are determined by carrying out a Fourier analysis of the torque versus crank angle curve and reducing it to an equivalent series of sinusoidal curves of varying frequency and amplitude. These are then represented by vectors which are combined for all of the cylinders to give a resultant value or *phase vector sum*. The various *orders* vary in frequency from that corresponding to 1/2 engine speed in frequency increments of 1/2 to about 8, beyond which their magnitude becomes insignificant. When all of the vectors of a given order point in the same direction, that is a major order — like the sixth just mentioned. When they point in different directions, usually symmetrically around a center, they are called *minor* orders. Because the crankshaft does not deflect uniformly along its length the minor orders can be troublesome as well.

Each order of vibratory torque has a magnitude that depends on the mean effective pressure, compression ratio, and, to a lesser degree, other engine operating variables. This vibratory torque multiplied by the vector sum associated with that order is the excitation torque for torsional vibration. If, for instance, the engine is operating at 2,500 rpm and the natural frequency of the crankshaft system is 6,250 vibrations per minute the 2½ order would excite vibrations in it. If there were no damping in the engine the amplitude of the vibration would be infinite and the crank would fail. At this point the designer would need to know the damping characteristic of the engine in order to calculate the amplitude, and hence the stress, in the crankshaft.

By the mid 1930s the technique just outlined had been refined enough that the designer could modify values of stiffness in the system so as to avoid severe torsional vibration problems. He may not have been able to predict the amplitude of vibration without prior experience with a similar engine design but the magnitude of the various excitation torques and associated critical speeds could be ascertained and influence the design process.

Analysis

As mentioned above, the first step is to construct a mass-elastic diagram for the engine to be analyzed. In this case I have chosen two configurations of the Allison engine and one of the Merlin. These are shown in Figures 1, 2 and 3.

Figure 1 is a composite of two mass-elastic diagrams provided by Dan Whitney for the V-1710-E (shown in Figure 3) and the twin crankV-3420 with close-coupled gearbox. I created the composite of Figure 1 to be closer to the Merlin's configuration because I thought it would provide a better comparison since the E version of the Allison engine had a long, flexible extension shaft and remote gearbox. There were at least two different couplings used by Allison to connect the

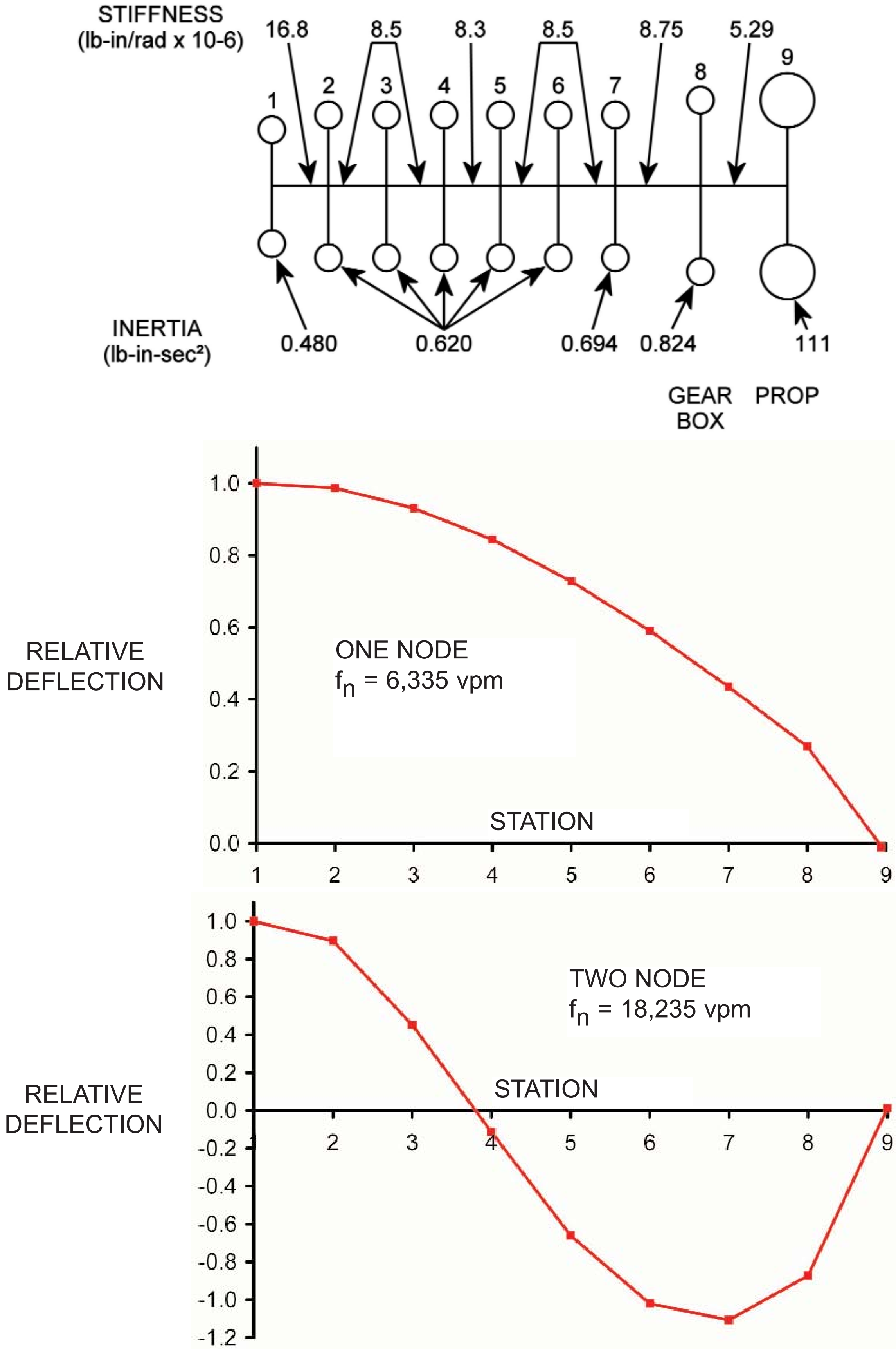


Fig 1. Mass-Elastic Diagram and Relativie Deflections for Two Modes of Vibration – Allison V-1710 Composite

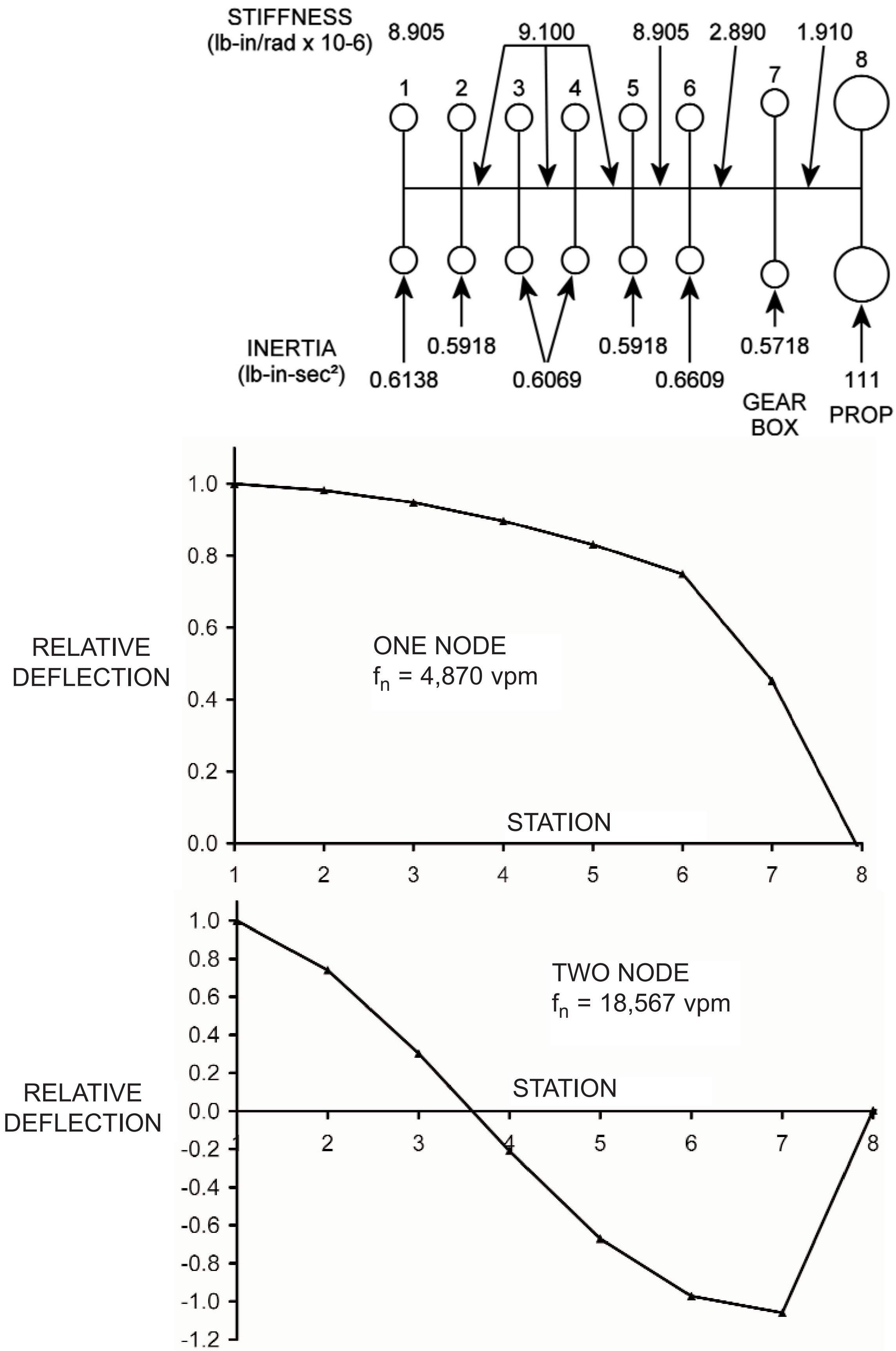


Fig. 2. Mass-Elastic Diagram and Relative Deflections for Two Modes of Vibration – Rolls-Royce Merlin

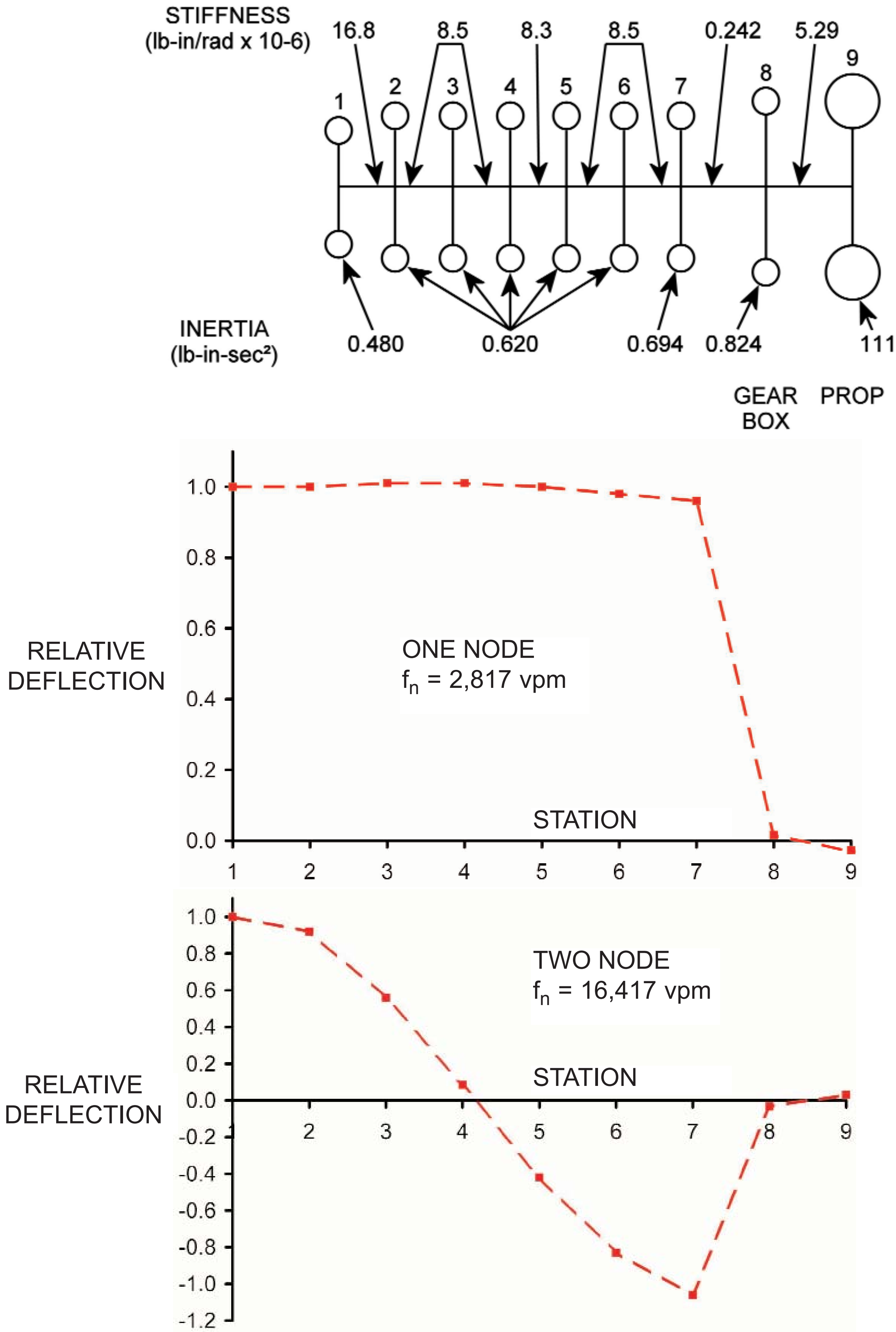


Fig. 3. Mass-Elastic Diagram and Relative Deflections for Two Modes of Vibration – Allison V-1710-E

crankshaft to the gearbox in the close-coupled configurations and information provided by Dan Whitney indicates that the numbers I used in Figure 1 conform to the later coupling design.

The Merlin mass-elastic diagram (Fig. 2) came from Reference 1 but did not include a propeller inertia (for reasons I will get into later) so I used the inertia of the Allison propeller. The Merlin diagram did not include the stiffness and inertia of the supercharger drive and I was unable to find those numbers. I do know from the drawings that the Merlin drive system was much more flexible than the Allison's and in two stage Merlins the inertia was effectively much higher so the natural frequency of the supercharger vibrating against the engine would be quite low.

I have chosen to ignore this lowest mode of vibration because it has little effect on vibratory stresses in the crankshaft. In calculating the natural frequency for the one and two node vibration modes shown in Figure 1, adding the supercharger from the mass elastic diagram for the Allison V-1710-E, not shown in Figure 3, changes the one node frequency by about 3% and the two node by an insignificant amount. The relative deflections remain unchanged.

Figures 1, 2 and 3 show the natural frequencies and relative shaft deflections at those frequencies for the three systems we are comparing. Note the relatively large differences in the one node frequencies of the three engines compared to the two node. This is due to the relative flexibility of the Merlin crank to gearbox coupling as compared to the composite Allison. This also explains the more gradual slope of the V-1710 composite deflection diagram. By contrast the V-1710-E one node frequency is considerably less due to the long extension shaft and its deflection curve is almost flat for the engine portion of the curve, which results in much smaller minor order resultant vectors for the one node vibration.

With the natural frequencies and the shape of the deflection curve established for the three cases the next step is to determine the phase vector sums for the various orders of excitation torques. The Merlin and Allison had different firing orders but the vectors combine in the same manner so there is no difference in the vibration characteristics of the two engines attributable to firing order. The only major orders in the operating range of these engines are the third and the sixth. The third order vector sum is zero for a 60° V-12 and the sixth order occurs only in the two node operating speed range so that the vectors associated with the back half of the crank are balanced to a great degree by the vectors from the front half. I examined the phase vector sums for all the orders in the operating range for one and two node modes of vibration. The ones with significant magnitude are included in Table 1.

Column 8 of Table 1 gives the resultant vector sums for orders that result in significant vibratory torque in the operating speed range of the three engines.

The next step is to determine the magnitude of the torques associated with the various orders. Reference 2 (an Allison paper) gives values of these (in terms of tangential pressure) for orders ½ to 6. I extrapolated their numbers to get the higher orders. The Allison numbers were consistent with the generalized data for spark ignition engines and I am assuming that the Merlin's are the same at the same imep. I converted their numbers from pressure to the ratio of vibratory torque to mean torque per cylinder. The mean torque is a function of the indicated mean effective pressure, which, in this case, follows a propeller curve up to about 2,000 rpm and 188 psi imep and, with the engine at full throttle gradually increases to 210 psi at 3,500 rpm. This is the load curve Allison used in its report to the Air Corps on the V-1710-E in 1939 and it contains a torsional analysis of the engine that would come to be equipped with the hydraulic and pendulum dampers for the remaining life of all the various V-1710 models. I thought it appropriate to use this load curve since the ratings of the Allison and Merlin were roughly the same in that time frame, ~1,000 horsepower.

Table 1, columns 4 and 5, show the engine speed and imep for the various orders under consideration. For each of these conditions there is a mean torque per cylinder for the imep shown (column 6) and a vibratory torque per cylinder (column 7) for the corresponding order. The product of the vibratory torque and the phase vector sum (column 9) gives the vibratory excitation torque in inch-pounds per radian of deflection at the rear of the crankshaft.

Figure 4 is a plot of the excitation torques for both modes of vibration for the three engines.

There is not much here to explain why the Allison was in need of dampers and the Merlin was not, especially considering that Allison was designing the damper system for the V-1710-E whose amplitudes are below the Merlin's across the board. The 4½ order, two node excitations are about equal in magnitude but the V-1710-E is peaking about 500 rpm lower and, therefore, closer to the operating range. It's doubtful that this would be more of a problem than the Merlin 1½, one node at 3,250 rpm.

Table 1
Data Used to Evaluate Vibratory Excitation Torque Amplitude of Vibration at Crankshaft Rear

	(1) Number of Nodes	(2) Natural Frequency f_n (vpm)	(3) Order n	(4) Engine Speed (RPM)	(5) Indicated Mean Effective Pressure IMEP (psi)	(6) Mean Torque per Cylinder T_m (in-lb)	(7) Vibratory Torque for Order n Tv_n (in-lb)	(8) Phase Vector Sum for Order n $\Sigma\Delta_v$	(9) Resultant Vibratory Torque $Tvn\Sigma\Delta v$	(10) Effective Engine Inertia J_E (in-lb-sec^2)	(11) Crankshaft Static Angle of Deflection θ_S (deg)t	(12) Magnification Factor at Resonance M	(13) Vibration Half Amplitude, Crank Rear θ (deg)
Allison V-1710 Composite	1	6,335	1½	4,223	210	2,382	4,716	1.42	6,697	2.87	0.30	10.0	3.00
	1	6,335	2½	2,534	203	2,303	2,741	0.73	2,001	2.87	0.091	10.0	0.91
	1	6,335	3½	1,810	162	1,838	1,452	0.73	1,060	2.87	0.048	10.0	0.48
	2	18,235	4½	4,052	210	2,382	1,191	5.67	6,753	3.51	0.030	3.6	0.11
	2	18,235	6	3,039	210	2,382	619	3.10	1,919	3.51	0.0086	3.6	0.031
	2	18,235	7½	2,431	202	2,291	367	5.67	2,081	3.51	0.009	3.6	0.034
	2	18,235	8½	2,145	193	2,189	263	2.46	647	3.51	0.0028	3.6	0.010
Merlin	1	4,870	1½	3,247	210	2,296	4,546	0.64	2,909	3.24	0.20	11.3	2.23
	1	4,870	2½	1,948	188	2,056	2,447	0.334	817	3.24	0.052	11.3	0.63
	1	4,870	3½	1,391	95	1,039	821	0.334	274	3.24	0.019	11.3	0.21
	2	18,567	4½	4,126	210	2,296	1,147	5.49	6,297	2.55	0.037	3.5	0.13
	2	18,567	6	3,095	210	2,296	597	0.38	227	2.55	0.0013	3.5	0.005
	2	18,567	7½	2,476	203	2,220	355	5.49	1,949	2.55	0.01	3.5	0.04
	2	18,567	8½	2,184	195	2,132	256	2.46	630	2.55	0.0037	3.5	0.013
Allison V-1710-E	1	2,817	1	2,817	208	2,360	5,192	0.075	389	4.30	0.06	13.6	0.81
	1	2,817	1½	1,878	175	1,985	3,930	0.113	444	4.30	0.068	13.6	0.92
	2	16,417	4½	3,648	210	2,382	1,191	5.07	6,038	2.62	0.045	4.0	0.18
	2	16,417	6	2,736	207	2,348	610	1.49	909	2.62	0.007	4.0	0.028
	2	16,417	7½	2,189	195	2,212	354	5.07	1,795	2.62	0.013	4.0	0.05
	2	16,417	8½	1,931	185	2,099	252	2.48	625	2.62	0.005	4.0	0.02

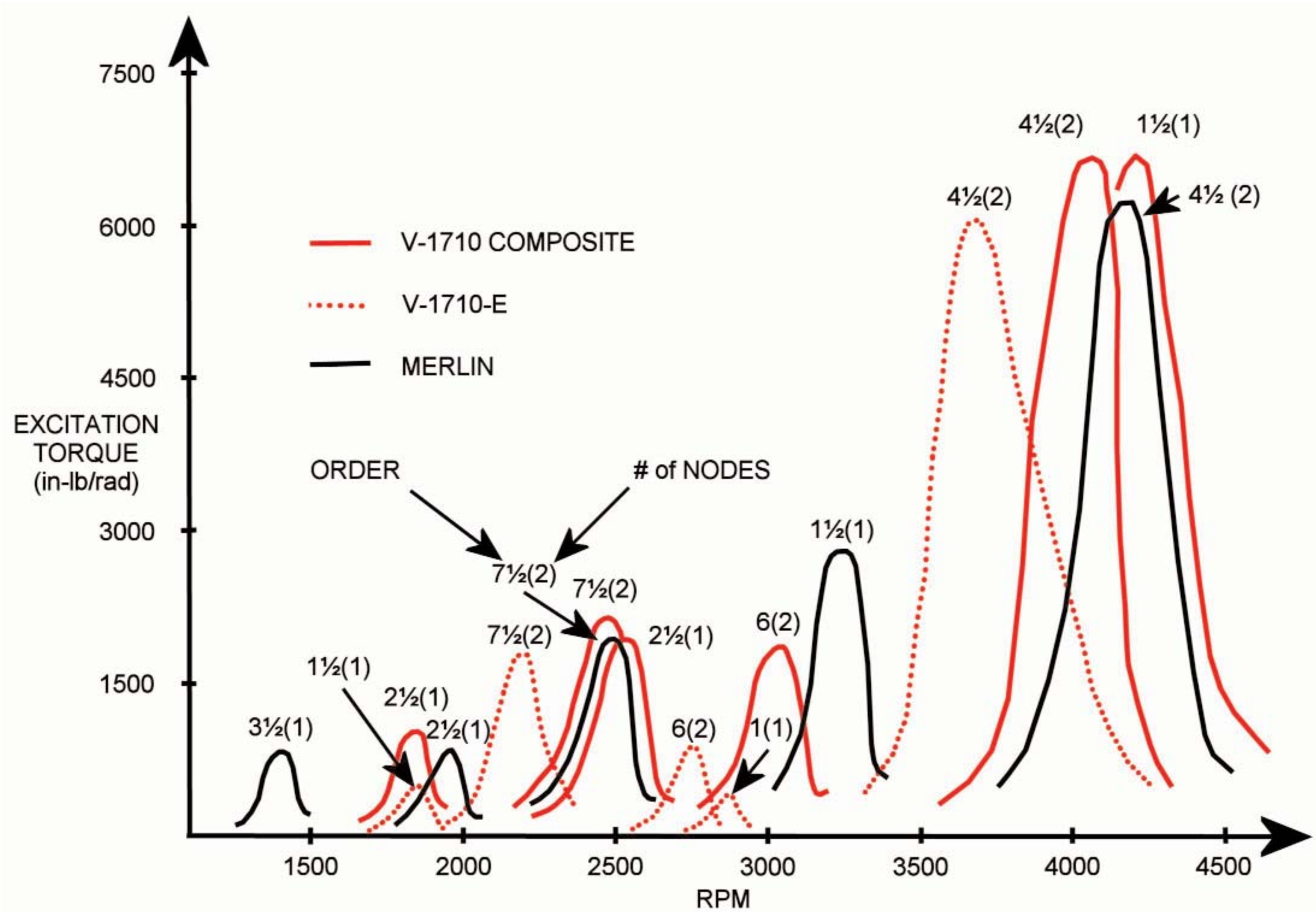


Fig. 4 Vibratory Excitation Torque Versus Engine Speed (see Table 1 for Mean Torque)
Note that the torque is for 1 degree of deflection at the rear of the crankshaft.

At this point I decided to try and predict an actual vibratory amplitude at the rear of the crankshaft to see how it might compare with Army-Navy Specification No.9504 ca.1942. This specification limits one node vibration amplitude to ±1.5° and two node to ±0.25°. Reference 2 (Allison again) gives a curve of magnification factor at resonance versus vibration frequency. If one equates the energy input for a particular order at resonance to the energy dissipated due to damping it is possible, with the magnification factor, to calculate the vibration amplitude. This is assuming there are no dampers in the system, only the natural damping in the engine itself. The Magnification factors for the relevant natural frequencies are shown in Table1, column 12. The equivalent inertia of the engine is given in column 10 and the static deflection is given in column 11, both of these values are used to calculate the half amplitude of swing at the rear of the engine, theta (column 13).

The results of this analysis are shown in Figures 5 and 6 for the one and two node modes of vibration.

The one node case does not appear to be a problem for either Allison configuration while the Merlin 1½ order could be considered problematical but Rolls-Royce never used a damper in that engine. The two node case shows everything to be below the value allowed by the A-N spec. With the higher engine ratings to come during the war years the V-1710-E 4½ order could be considered a problem, but certainly not in 1940.

The construction of a Holzer table allows one to calculate a torsional stress occurring when the crankshaft is in free vibration at its natural frequency since torques are calculated at each station in the mass-elastic diagram. Table 2 shows these stresses for our three cases.

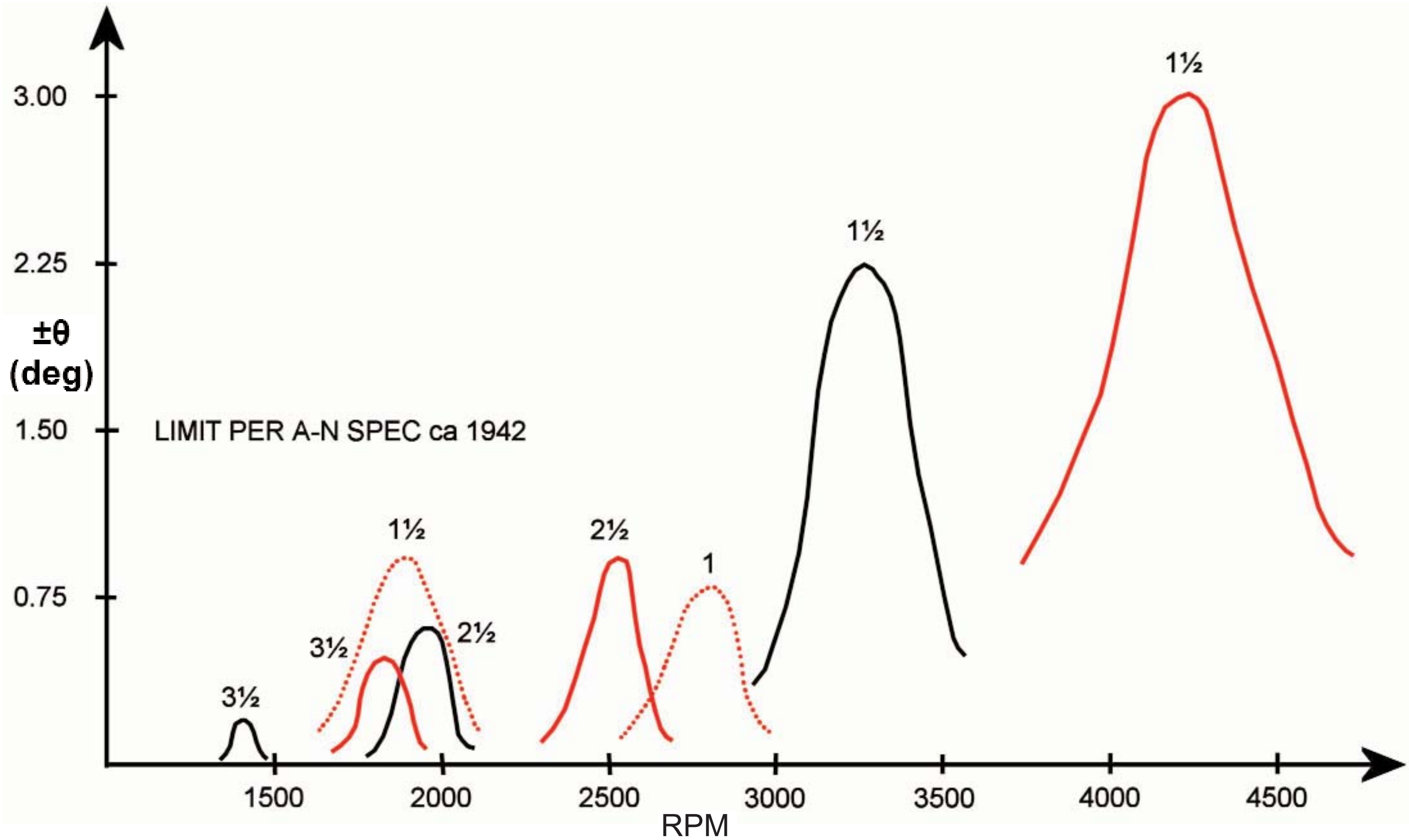


Fig. 5. Estimated Vibration Amplitude at Rear of Crankshaft versus Engine Speed – One Node Vibration Mode (see Table 1 for Mean Torque)

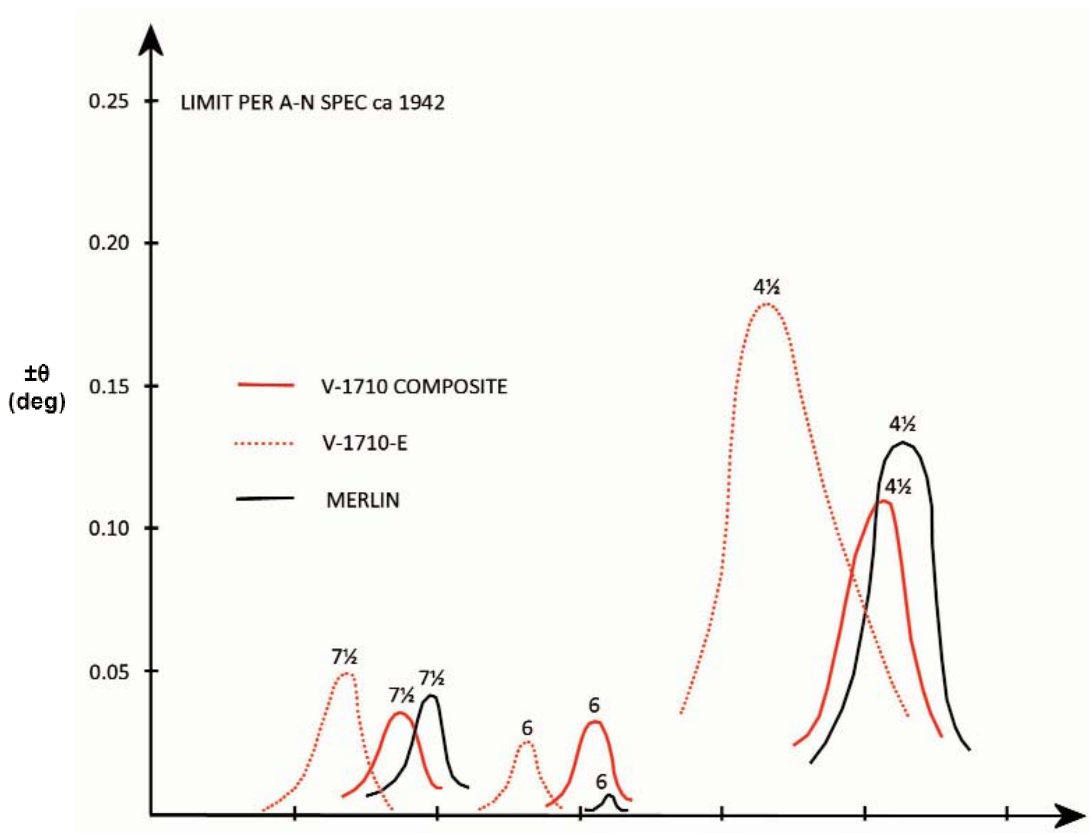


Fig. 6. Estimated Vibration Amplitude at Rear of Crankshaft versus Engine Speed – Two Node Mode of Vibration (see Table 1 for Mean Torque)

Table 2
Maximum Vibratory Stress in Crankshaft Journals for One Degree Amplitude at Rear Of Crankshaft. Free Vibration

	Stress (psi)		Location (from Engine Rear)	
	One Node	Two Node	One Node	Two Node
Allison V-1710 Composite	3,062	10,072	# 7 Main Journal	#3 Main Journal
Merlin	2,383	12,957	#7 Main Journal	#3 Main Journal
Allison V-1710-E	393	8,848	#7 Main Journal	#4 Main Journal

The Merlin's one and two node stresses are higher than the V-1710-E values by a significant margin. Some of this is due to the smaller diameter of the Merlin's journals. The two node value is lower also due to the lower natural frequency in the V-1710-E two node vibration mode and the low one node stress is due to the very little amount of twist in the crank as seen in Figure 3. Again, if the Merlin didn't need dampers the V-1710-E should not have needed them either.

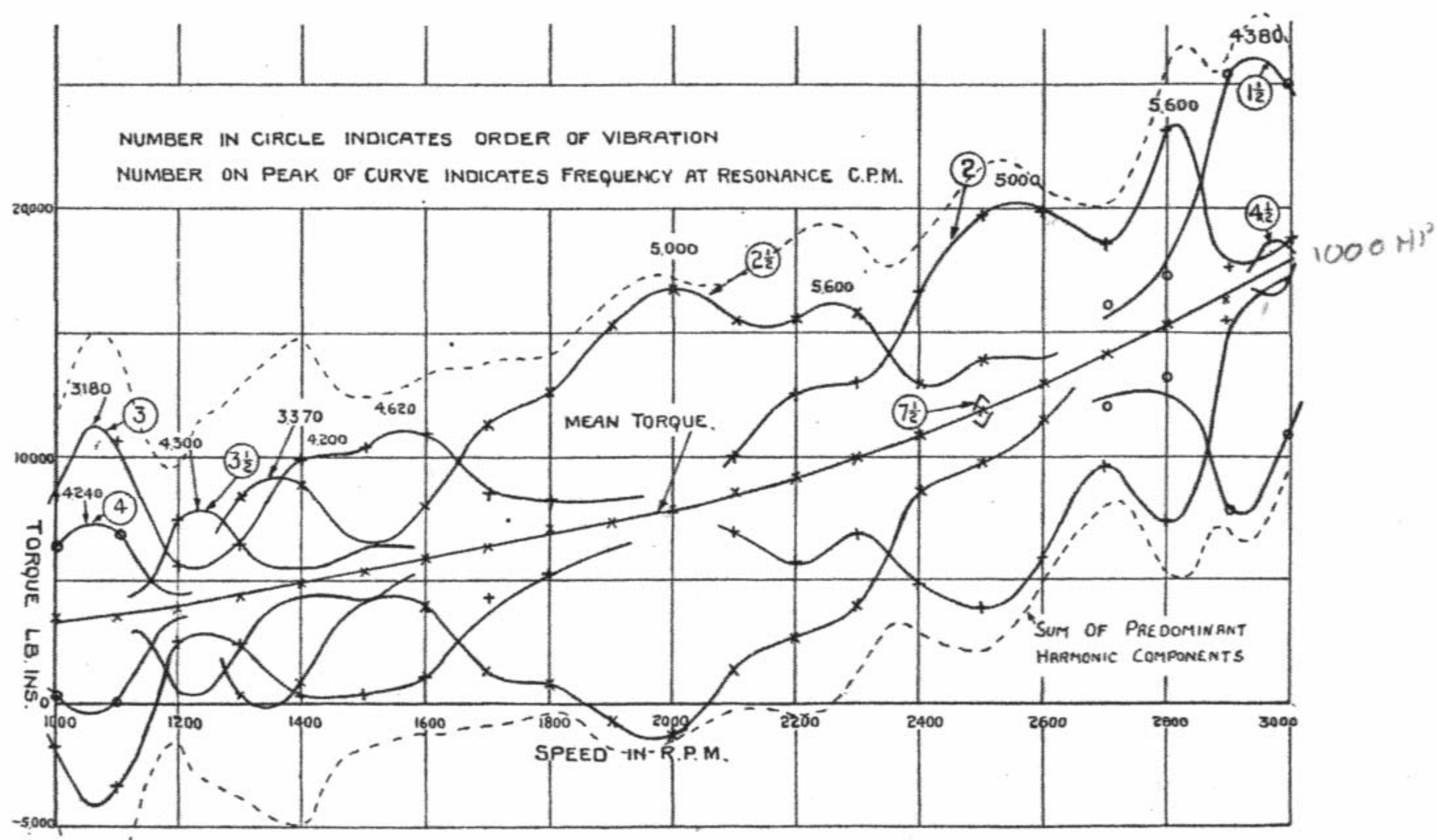
Table 2 also illustrates why the allowed maximum half amplitude is only a quarter of a degree for the two-node mode. The numbers in the table are for one degree half amplitude and restricting the amplitude to one quarter would get the vibratory stress down to the same level as for 1½ degree one node vibration.

Discussion

This section will consist of a discussion of my results as compared to the limited amount of data available and experience with the two engines in air racing and in hydroplane boat racing post WWII.

Tests conducted at Wright Field on a Packard-Merlin (Reference 3) gave a 2½ order peak of about 0.35° at station 6 in Figure 2. Using the relative deflections of Figure 2 this translates to about 0.5° at the rear of the crankshaft. My analysis gave about 0.6° as shown in Figure 5 so I am apparently at least in reasonable agreement. Their peak occurred at about 2,300 rpm while mine occurred at 1,950. This same Army report notes a 7½ order at 2,700 rpm but does not give a magnitude. My analysis gave a 7½ at about 2,500 rpm. An interesting comment in the Army paper is that they lacked a mass-elastic diagram for the Merlin and planned to carry out such an analysis. One would assume that if Packard had such an analysis they would gladly have supplied it and that Rolls-Royce would have supplied it to them as part of the licensing arrangements if torsional vibrations were a significant problem. Remember that this was in mid 1942. A comment I received from Dave Piggot of the Rolls-Royce Heritage Trust on this subject may shed some light on this: "In all the time I have spent in our archive, I have never come across any reports of some substance on the subject of torsional vibration on either the Merlin or Griffon".

Figure 7 shows the results of another torsional test on a Merlin. This was carried out at the Royal Aircraft Establishment in 1943 (Reference 1) and is the source of the mass-elastic diagram in Figure 2. I mentioned that no propeller inertia was included and that I had used the Allison propeller's inertia. Figure 7 is similar to Figures 5 and 6 except it is in terms of torque rather than angular deflection at the rear of the crankshaft. The torque in this case was measured between stations 6 and 7 in Figure 2. Note that Figure 7 shows the same orders of vibration occurring at a number of engine speeds and even a third order, which should not occur. This is due to the fact that then modern aluminum variable-pitch propellers did not behave as a solid inertia as in our mass-elastic diagram, but were subject to various



—Predominant harmonic components of torsional vibration. Blade angle 16·5°

MERLIN II

Fig. 7 (from Reference 1)

bending and flapping modes of vibration that could excite vibration in the crankshaft. In essence the propeller should be thought of as a branched system of distributed masses subject to bending and twisting. Apparently the three bladed prop caused the third order shown in Figure 7. The 2½ order has peaks at ~1,350, 2,000, and 2,250 rpm whereas my peak as shown in Figure 5 is at 1,950 rpm. Of the three their 2,000 rpm gave the highest amplitude and the resonant frequency recorded was 5,000 vpm, close to my value of 4,870.

The two node orders in Figure 7 (4½ and 7½) are almost non-existent in magnitude and the authors concluded that "judging from the corresponding stresses in the crankshaft, the two-noded mode of crankshaft torsional vibration appears to be relatively unimportant but it has been found that this mode of vibration may give rise to high stresses in the propeller blades". My analysis as shown in Figure 6 gives a 7½ at 2,500 rpm versus 2,500 rpm in Figure 7 and a 4½ order at 4,100 rpm versus 3,000 rpm in Figure 7. They may have been picking up the flank of that order or it could be that the propeller had shifted the natural frequency as it did for the one node cases. Runs similar to those shown in Figure 7 with changes in the pitch setting of the propeller gave very different results to those shown in Figure 7. The largest 2½ order occurred at 2,100 rpm with a blade pitch of 29° and a natural frequency of 5,200 vpm.

As an additional reality check the amplitude of vibrational torque at station 6 (Fig. 2) from Figure 7 is about 9,500 inch-pounds. Using the Holzer table for the Merlin and the vibrational amplitude from Figure 5, 0.6°, I calculate a vibratory torque of 9,000 inch-pounds. The analysis seems to fit what data I have for the Merlin engine. Unfortunately I lack any comparable data from the Allison engine. The Army report mentioned previously, Reference 4, was the source of the S.A.E. paper, Reference 2. Neither document contains the results of a torsiograph test that could explain their need for a damper to protect the crankshaft. There is a final figure in the S.A.E. paper that shows excitation torques and the results of a torsiograph test with and without a damper. It is not stated whether it is one damper or both that is being applied and the magnitudes and frequencies at which the excitation torques occur do not correspond to the results for the V-1710-E in Reference 4.

The only evidence I've seen for the need of dampers in the Allison engine is Allison's statements to that effect. Reference 5, Dan Whitney's book, sites a crankshaft failure at a location that would correspond to a maximum two node stress area. The actual failure was attributed to a manufacturing defect but the pendulum dampers were installed as a fix.

At this point I will introduce the damper construction used in the later model Allison engines. Some earlier models employed a friction type damper at the output gear. Allison's reason given for abandoning this approach was that it was ineffective for damping two node vibrations where amplitudes are small. Allison's system is shown in Figures 8 and 9.

The hydraulic damper protects the supercharger drive and dampens the one node vibration. The pendulum dampers are tuned for the two node 4½ and 7½ orders, 3 weights for each order. According to Dan Whitney the 4½ order weights were removed in the later V-1710-G engines, which, given the results of my analysis, is really puzzling when you look at Figure 6. I'm assuming here that my composite Allison is close to the G model. Since the pendulum dampers were designed originally for the E model, perhaps the lower amplitude and higher critical speed of my composite engine could justify removing the 4½ damper even though the G model was rated at a higher speed than the E model. On the other hand, the engine ratings were much higher for the "G" model so the excitation torques and hence the vibration amplitudes would have been much higher than those shown in Figures 4 and 6.

To summarize, my analysis of the Merlin seems to agree with the few test results I can check it against. The analysis for the Allison E model shows much more modest torsional activity than for the Merlin yet the Allison was equipped with first and second mode dampers and the Merlin had none. The only factor that can explain this is if the Allison engine had much less natural damping than the Merlin. As I mentioned before this seems very unlikely due to the larger main and crankpin bearings in the Allison and the likely similar friction mean effective pressure (see Reference 6).

The experience with both of these engines post WW2 in aircraft and hydroplane boat racing provides some additional perspective on this question. Both applications resulted in engine speeds well above their rated speeds in military aircraft, routinely into the mid 4,000 rpm range. In the boat application the Merlin failed crankshafts until additional counterweights were added to reduce main bearing loads, particularly the center main. Without the additional counterweighting the center main bearing cap bolts would yield and stretch reducing support for the crank. Once the weights were added no further crankshaft failures were experienced. The stock supercharger drive system in the Merlin had a life of about 15 minutes, apparently due to rapid engine speed changes when the boat's propeller came in and out of the water failing the quill shaft and overrunning clutch. The fix for this was a stiffer and stronger quill shaft. My source for this information was Dixon Smith who had personal experience in this area and is a practicing mechanical engineer. He does not know of anyone installing dampers in Merlins but Dan Whitney has apparently heard of at least one. It should be noted that the torsional characteristics of both engines would have been changed significantly due to the differences in the propeller inertia and the change in the gear box and coupling to accommodate a speed increase in boat racing applications. I doubt that these changes would have changed the two node mode of vibration very much.

Experience with the Allison engine was somewhat different. There apparently were no main bearing cap bolt stretching problems, probably due to the more heavily counter-weighted design

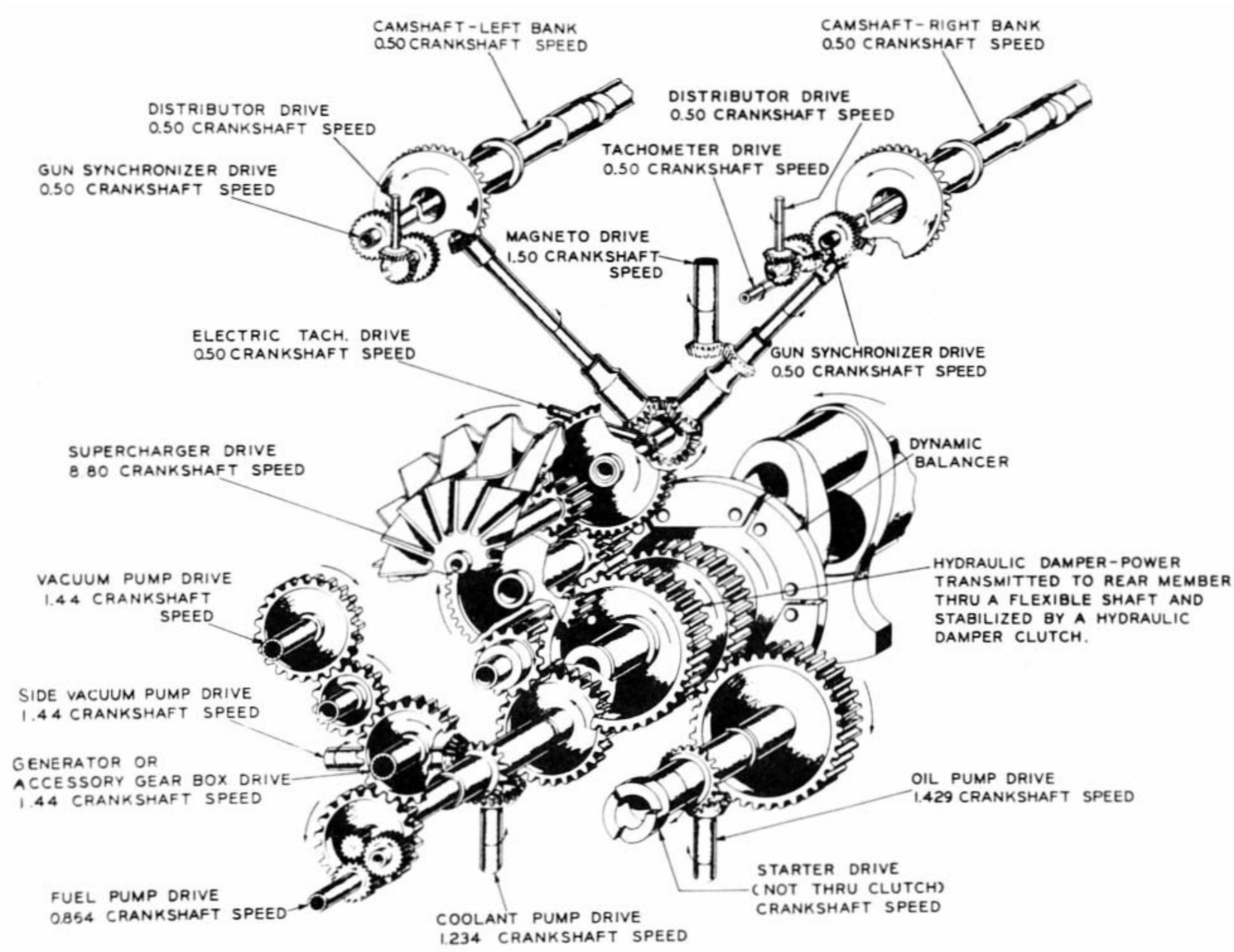


Fig. 8. Allison Gear Train Showing Dynamic and Hydraulic Dampers

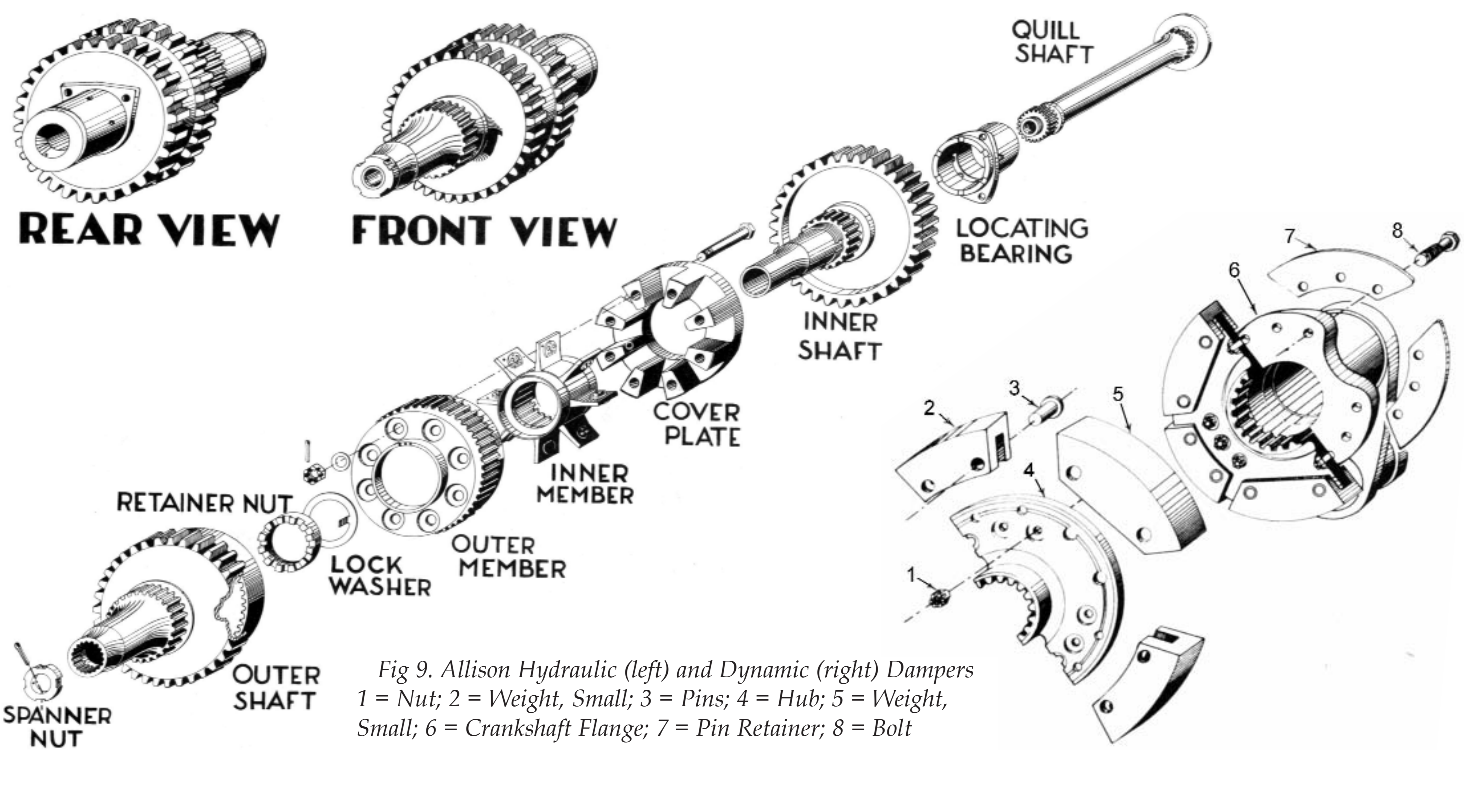


Fig 9. Allison Hydraulic (left) and Dynamic (right) Dampers
1 = Nut; 2 = Weight, Small; 3 = Pins; 4 = Hub; 5 = Weight, Small; 6 = Crankshaft Flange; 7 = Pin Retainer; 8 = Bolt

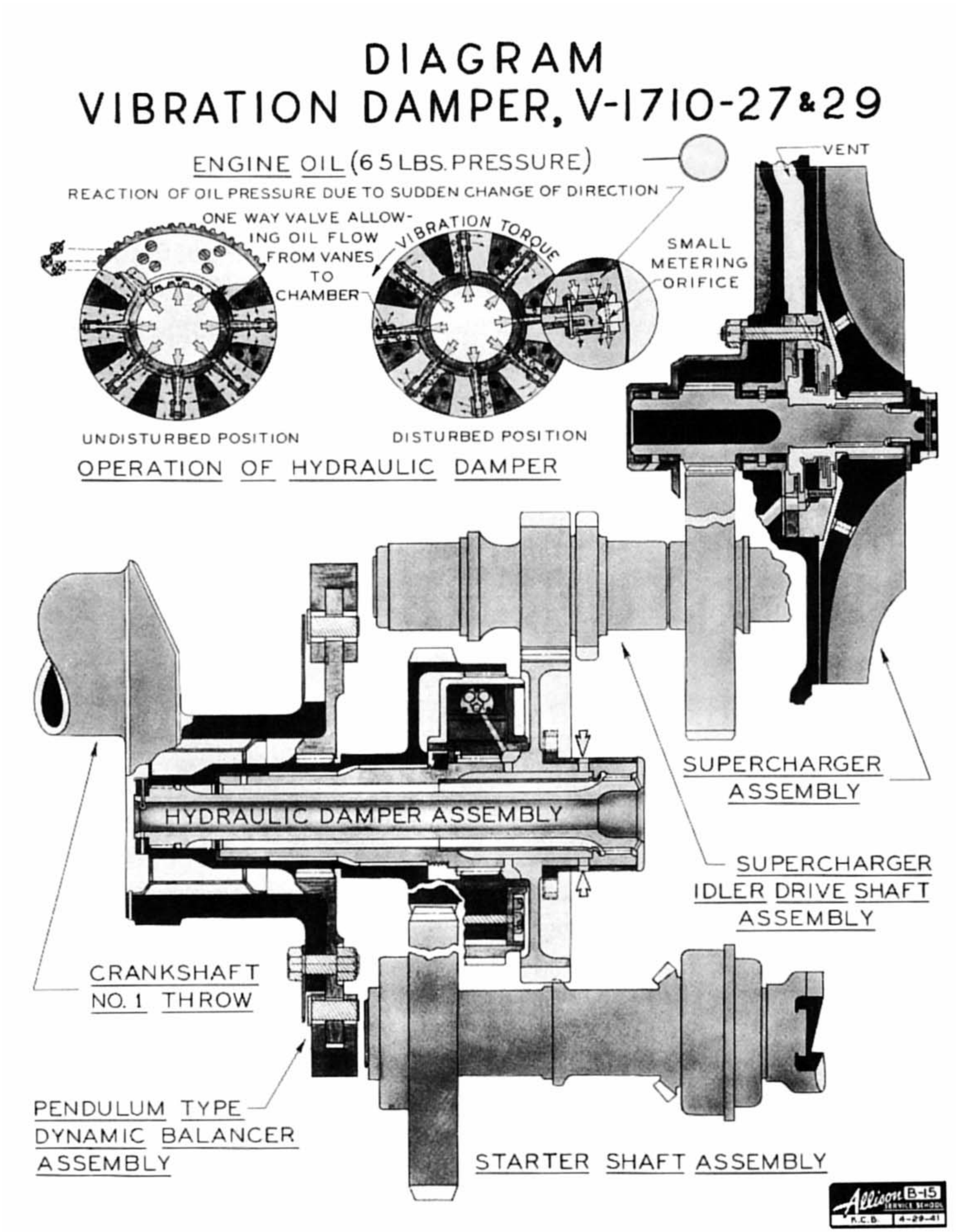


Fig. 10. Allison Hydraulic Damper Operation (from Reference 5)

of its crank. Dixon Smith claims the pendulum dampers would some times fly off at high engine speeds and wreck the engine. Consequently some racers removed the dampers. We do know that wear was a problem with the damper pins and the holes they operated in. These would gall in relatively short periods of time due to high contact (Hertz) stresses. The NACA investigated the problem and reported on it in 1945 (Reference 7). They recommended an increase in the pin and hole size and ran tests to show that it worked. It isn't clear when, if ever, this fix was introduced in production engines. Their design was for a take-off speed of 3,000 rpm. The re-designed pins reduced the stress from 146,000 psi to 119,100 psi. At 4,500 rpm the stress would be back to178,600 psi. The effect of wear on the pins is to de-tune the dampers and enough wear could cause them to break.

I have heard no evidence that the sudden over speeding phenomenon in hydroplane racing caused problems with Allison's supercharger drive system as it did with the Merlin's.

Conclusion

All of the evidence I have been able to assemble and all of the analyses I have carried out lead me to conclude that the Allison dampers were not necessary to protect the crankshaft, gear box or propeller drive systems. The only factor that could explain such a requirement would be inherently less damping in the Allison engine than in the Merlin. This, it seems to me, is highly unlikely for the reasons given above.

The analysis is based on ca.1940 ratings and these two engines ultimately were rated at roughly twice that power. If we double the vibratory excitation torques and look at Figures 5 and 6 we see that we are still at or near the Army-Navy specs at speeds under 3,000 rpm. The damper weights in the Allison were the same in 1945 as they were in 1939, despite the increase in ratings, which implies they had sufficient amplitude to generate an adequate reaction couple to the increased vibratory excitation torque.

Dan Whitney has suggested the most plausible scenario for the adoption of dampers in the V-1710. The V-1710-E was designed for the Bell XP-39 aircraft where the engine was located behind the pilot and a cannon fired through the gearbox and propeller shaft. This was the reason for the long extension shaft (Fig. 10). Testing revealed misfiring above about 2,600 rpm. This was traced to the magneto and attributed to torsional vibrations exciting a resonance in the camshaft and magneto drive system, which was driven off the rear of the crankshaft. This apparently developed into somewhat of a panic situation and, it seems, the initial approach to the problem was to try to dampen the amplitude of vibration at the end of the crank. If we examine Figures 5 and 6 we see that the 1st order one node or the 6th order two node could have induced vibration at about 2,600 rpm, hence the dampers for both modes of vibration?

What finally worked was increasing the wall thickness of the extension shaft. This would have stiffened the shaft and increased the one node

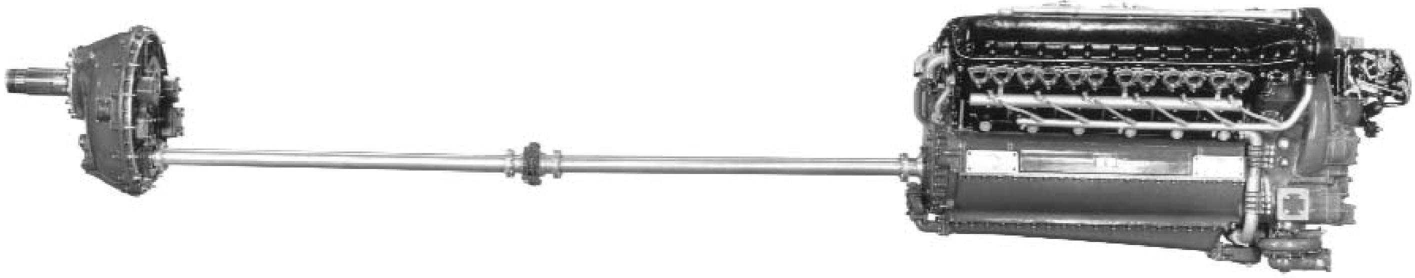


Fig 10. AllisonV-1710-E as used in the Bell XP-39

natural frequency enough to stop it from exciting the resonance in the cam/magneto drive system. Note that in Figures 1 and 3 the one node natural frequency of the V-1710-E was only 44% of the close-coupled V-1710. The change in the extension shaft stiffness would have had a much smaller effect on the two node natural frequency but it's impossible to judge at this point in time which mode of vibration was causing the resonance. My guess is that it was the one node. Changing the natural frequency would seem to have been the obvious thing to do first, rather than trying to reduce the amplitude of vibration at the same frequency but we weren't there and shouldn't judge.

Many engineers, myself included, have been in similar situations where "everything but the kitchen sink" is thrown at a problem in an attempt to get the product to the customer on time and performing as promised.

Once the dampers were designed and installed they were left in all subsequent versions of the V-1710 and provided an additional benefit. The torsional excitation of the crankshaft would have had an equal and opposite reaction on the crankcase and, therefore, on the airframe in which it was installed. A pilot with experience of both the Allison and the Merlin in the same air-frame (if such a person exists) could possibly comment on this.

The fact that Allison was a General Motors company may have contributed to the adoption of dampers. By the late thirties many automotive engines were equipped with dampers and the impetus had probably as much to do with providing a vibration free vehicle as with protecting the crankshaft. We know that the General Motors Research lab was involved in V-1710 development and contributed very significantly to the optimization of the crankshaft design. It seems quite likely that they may have encouraged the adoption of dampers. They are simple and do not add much weight to the engine.

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I wish to thank Dan Whitney and Kim McCutcheon for bringing this topic to my attention and providing encouragement and helpful information along the way. Dan's encyclopedic knowledge of the Allison engine and his collection of otherwise unattainable information made this effort possible.

Dixon Smith's generous contribution of time to relate his racing experience to someone who has never taken a wrench to an aircraft engine is much appreciated and I was very gratified to be able to get it in print.

Dave Piggott of the Rolls-Royce Heritage Trust provided valuable insight into Rolls' activity in the area of torsional vibrations.

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