A German Assessment of the Allison V-1710 Aircraft Engine

by Dan Whitney

Reviewed and Edited by Robert Raymond

During the course of his engine research in the British Archives, AEHS Member Calum Douglas came across a large technical file produced by Daimler-Benz in 1943/1944, reporting on their analysis of a captured Allison V-1710. Their work was focused on the internal components and dynamics of the power train, particularly the crankshaft, reduction gears and supercharger drive. Included are extensive calculations used to determine the critical vibration frequencies inherent in the engine. It is not clear why they felt this was necessary, unless they were seeking vulnerabilities in the engine or its design, or if they were looking for potential solutions to associated issues they may have been having in their engines, or to justify features of their engine design. To this end, the Daimler-Benz V-12s used a hydraulic drive to the supercharger, a device that ran the supercharger at its optimum speed for all conditions, however lost considerable energy into heating of oil, as well as operational problems due to sludging. The amount of effort expended preparing this report was considerable, particularly given the course of the war at the time, suggesting that they had real concerns about their future engine development.

While the report at hand was based on a captured Allison V-1710-39(F3R), it is apparent from the report that they had previously done a similar analysis of the earlier V-1710-33(C15) engine. These engines models powered the Curtiss P-40E and P-40C respectively, and differed not only in detail, but the F3R featured an improved reduction gear incorporating external spur gears rather than the C15s internal spur gear. The supercharger drives were also quite different, as the F3R supercharger is driven from the aft end of the crankshaft while the C15 supercharger, also mounted at the rear of the engine, is driven by a long quill shaft from the propeller reduction gear. The Germans referred to this as a "parallel" drive, while the F3R supercharger drive was described as "series." Interestingly, Daimler-Benz showed that the effects on crankshaft dynamics and torsional vibrations within the engine were quite different.

The following article is the result of translating the original hand written German material into English. As a caution to other historians and researchers seeking to utilize archived German records from this era, the handwritten script used is known as "Suetterlin." This form of handwriting was removed from the German school curriculum during the 1930s as a Jew was involved in its creation, making its teaching unacceptable during this period. Finding an interpreter of this rapidly dying hand becomes more difficult with every passing day. Researchers seeking to access such material are encouraged to hasten!

The following discussion focuses on the torsional vibrations inherent in the Allison V-1710, V-12 engine. These are not unique to the V-1710; rather they are a factor in the design and success of every reciprocating engine. They became the subject of the Daimler-Benz investigation as they were the designers and manufacturers of some of Germany's primary V-12 aircraft engines, specifically the DB 601, DB 603 and DB 605. Obviously, they were interested in why American engines might be different from German engines, and at the same time, was there anything they could learn from the American designs that would be advantageous to their own engines and developments.

The primary moving components within a V-12 engine power train include the propeller, reduction gears, crankshaft and supercharger drive. These components rotate at relatively high speeds and are driven by the considerable forces coming from combustion gasses via the pistons and connecting rods. These forces cause intermittent loading and un-loading of the crankshaft journals, which results in non-uniform torsional inputs to the crankshaft, causing it to load-unload at a frequency proportional to the speed of the engine. This variation creates uneven torque impulses on the entire drive train, which being made of high strength steel, twists. This spring-like twisting and un-twisting induces a vibration within all of the components of the system, which in turn manifests themselves as vibrations.

These vibrations can be thought of as being similar to the vibrations of a violin string. There is a natural frequency of vibration when the string is "plucked," or energized. In addition there are higher order vibrations induced as well. These are multiples of the basic mode of vibration and resolve themselves as more complex shapes. If we think of the first mode having only one "node," then the string is fixed at one end and open to vibrate at its opposite end. An example would be a "tuning fork." Its "second mode" would have two "nodes," nodes being points along the string where the amplitude of vibration is zero, however all of the stresses inducing vibration in the subsequent section of the string pass through this point. A string having three nodes has the appearance of a "sine" wave, that is, half of the string has positive going amplitude while the second half amplitudes are negative going mirror images of the first. Such an occurrence is known as Mode 3, i.e., having three nodes.

Each of these modes has a natural frequency, determined by the stiffness, configuration and shapes of the materials of construction. Within the mode there are other events occurring, these are vibrations which are multiples of the fundamental natural frequency of the mode and occur at specific crankshaft speeds, or rpm. These orders start at ½ times the fundamental, and increase in the series 1, 1½, 2, 2½, 3, 3½, 4, 4½, … Many of these orders have very little energy in them and can be ignored. Some however, often the 4½, 6 and 7½ orders, can be very energetic, and if ignored, can produce stresses that will fatigue and ultimately fracture the metal in the crankshaft or drive element.

Often the way to deal with these effects is to either strengthen the component, not allow operation within the rpm range of the offending mode and order, or to provide a damping device tuned to absorb the offending energy associated with the order. The Germans V-12 engine builders, both the Junkers Engine Company[1] and Daimler-Benz, for whatever reason, chose to use heavy crankshafts in their engines, wherein the resulting stiffness apparently alleviated the need to otherwise resolve and deal with the effects of crankshaft torsional vibration. Allison on the other hand, focused on minimizing the weight of their engines, and as a result designed and fitted dynamic vibration dampers on the crankshaft and drive train to

[1] The Jumo 210 crankshaft weighed 164 pounds (74.5 kg), versus 102 pounds (46.4 kg) for the six-counterweight crank in the Allison V-1710-E/F, and about the same stiffness as the Rolls-Royce Merlin crank. Interestingly, the 12-counterweight crank in the V-1710-G weighed less, but was stronger.

absorb the undesirable torsional energy fluctuations, and thereby protect the components from fatigue and failure.[2]

Contained within the Daimler-Benz report was this excerpt from an earlier German report comparing crankshafts:[3]

> *The crankshafts of the German models are considerably stiffer than those of Rolls-Royce and Allison. A polar moment of inertia of 746 cm^4 for the crankshaft main journals of the DB 605 crankshaft and 618 cm^4 of the Junkers crankshaft compared to 424 cm^4 in the case of the Rolls-Royce Merlin 61 and 571 cm^4 in the Allison.*[4] *Similarly, the ratio in the crank pins and the cheeks – especially the Junkers cheek is very stiff – so that the deflections and twisting, which are the causes of the unfavorable peak loads in the bearings, are much lower in the German models. This also allows larger bearing areas [to] accommodate and thus suppress bearing loads.*
>
> *The structural strength [stiffness] of the crankshaft is positively influenced by overlapping the main and crankpin journals, which Rolls-Royce and Allison renounced in favor of manufacture.*[5]
>
> *The advantages mentioned are achieved by a significant additional weight* [in German engines]. *The larger crank pin also increases the connecting rod weight and thus the reciprocating mass [kinetic] forces, so that the advantage of larger bearing areas is somewhat reduced. The well-known three-piece Rolls-Royce main connecting rod is, in terms of weight, still somewhat lighter than the more conventional design, as shown by comparison with the Allison connecting rod.*

The following Daimler-Benz report shows that their engineers were fully versed in torsional vibration analysis, and evidently routinely preformed it on their engines. That said they were particularly interested in how Allison approached the problem, and specifically, the purpose and design of the two dynamic vibration absorbers built into the V-1710.

Vibration Assessment of Allison Aircraft Engines

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[2] During the course of the war the Allison crankshaft evolved, first being plain, then shot-peened, and finally Nitrided, continually improving its resistance to fatigue. The Packard built Rolls-Royce Merlin crankshaft was nitrided; the Merlin crankshaft journals and crank pins were smaller than the V-1710s, yet it was not more prone to fatigue failure. See "The Metallurgy and Processing of the Packard-Built Rolls-Royce Merlin Crankshaft," Frey, M.L., Proceedings of the Society for Experimental Stress Analysis, Vol. II, No. 2, 1944, p.158.

[3] For clarity, translated German material is shown in *italics*.

[4] See page 44 in DB report for calc of 571 cm^4, (13.8 in^4). The standard formula is $J = \pi/32 (D^4 - d^4)$, where "D" is the main journal outside diameter and "d" is the inside diameter.

[5] Your author and editor disagree with this statement, as both the Merlin and V-1710 had positive pin/journal overlap, while the Jumo 211 had negative overlap, i.e., no overlap. The DB 601 overlap was more than the Merlin and V-1710.

The bulk of the above, prior to Section XI of the report, consists of extensive hand calculations and sketches used to define the mass and shape of the various elements of the components listed in the above Table of Contents. This information was then utilized to determine, the Moment of Inertia and Stiffness of the individual and combined components, and thereby the modes and resonance frequencies, via Holzer Analysis, of the entire power train. We are fortunate to have the similar analyses done by Allison for the same components, and the results will be compared.

The early sections of the report focus on determining the physical and dynamic elements of the various components in the power train, for use in later analysis sections.

> Note: To compare D-B moment inertia values to those calculated by Allison, and shown on some of the following charts, one can convert their kg/cm-sec^2 to Allison inertias, given in terms of lb-in-sec^2, by multiplying by the radius of gyration squared. The radius of gyration for the crankshaft is the crank “throw”, of 3 inches, or 7.62 cm.
>
> The values given in Figure 1 are the Polar Moments of Inertia for the crankshaft and the masses attached to the crankshaft. Allison determined the inertia of the crankshaft using the torsional pendulum method, for which the result was 0.292 lb-in-sec^2 (0.0058 kg/cm-sec^2). To this was added the effect of the loads attached at each crank throw, the values of which are the result of the physical configuration and weights of each component. Allison used the following relationship to determine the resulting inertia values on the journals:
>
> $$I = R^2\,[Wc + 0.5\,Wd\,(1 + 0.25\,y^2)]\,/\,g, \text{ lb-in-sec}^2 \text{ or kg-cm-sec}^2$$
>
> Where: R is the crank “throw,” or radius, 3 inches (7.62 cm), Wc is the rotating weight on each crank throw/journal, 7.63 lb (3.46 kg), Wd is the reciprocating weight on each throw, 12.54 lb (5.69 kg), “y” is the ratio of connecting rod length to throw, (10 inches/3 inches) 3.333, and “g” is the gravitational constant, 386 in/sec^2 (980

cm/sec^2). As described above, multiplying by the crank throw (radius of gyration) squared, and converting units, results in the Daimler-Benz preferred units of kg/cm-sec^2, as presented on Figure 1.

The result of the connecting rods, bearings and pistons connected to each journal are 0.328 lb-in-sec^2 (0.0065 kg/cm-sec^2), and for the crankshaft throw 0.292 lb-in-sec^2 (0.0058 kg/cm-sec^2), for a total of 0.62 lb-in-sec^2 (0.0123 kg/cm-sec^2). These values are shown on Figure 9 and Figure 1 respectively. Note that the inertia on the forward cylinder is slightly greater as it was assigned one-third of the inertia of the extension shaft to the propeller.

Note also, that the Daimler-Benz inertia for the supercharger (Lader) is also much larger than the Allison value. Allison again determined the moment of inertia of the supercharger impeller empirically, by the torsional pendulum method, and then adjusted it for the effect of the step-up gears. In the case, shown in Figure 9, the step-up gear ratio was 6.44:1, while for the V-1710-F3R analyzed by Daimler-Benz the step-up ratio is 8.8:1. Making the adjustment results in a value of 2.87 lb-in-sec^2 (0.0569 kg/cm-sec^2). Curiously, this result is only half of the supercharger inertia determined by D-B.

As you can see, the moments of inertia are the result of physical features of the loads on the crankshaft journals; it is not yet clear why the values determined by Daimler-Benz should be different than those calculated by Allison, but a review of their analysis suggests that they calculated the moment of inertia for the crankshaft rather than measuring it experimentally.

Relating to the above indexed sections of the Daimler-Benz report:

***Section X** , Torsional Vibration Calculation*

This section concludes with a diagram, Figure 1 below [D-B figure W602], showing the component inertias and elastic lengths between components.

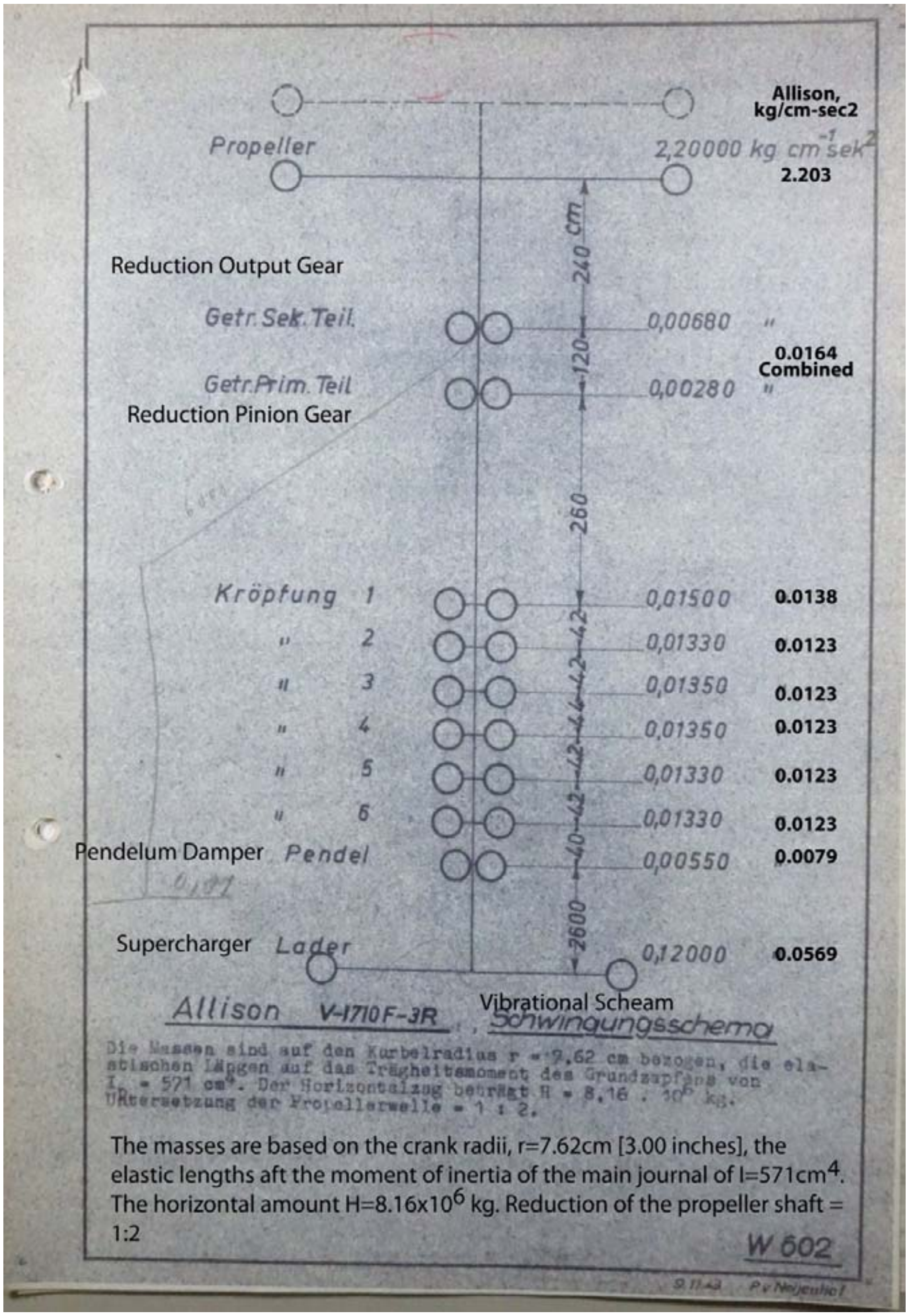

Figure 1. Daimler-Benz calculated Vibration Scheme for V-1710-F3R. Similar values calculated by Allison for the V-1710-E2 are shown in black and presented in Figure 9 following. See also, Figure 10 for a working version of this schematic done by D-B during analysis.

Section XI *Torsional Vibration Calculation*

This section determines the natural frequency and the relative strength of the various vibration orders, and states: *Therein the torsional vibration calculations identified the 3 node, 4½ order, for which the centrifugal pendulum [damper is] designed, having a peak at 3,884* *rpm, while the highest operating speed is at 3,000 rpm. Therefore, no vibrational [difficulties] occurred, the result of the incorporation of dynamic vibration dampers. The following [analysis] is intended to examine the effects of larger masses [weights] and/or larger elastic lengths, and how they might affect the 3 node, 4½ order, when operating at higher speeds.*[6] Figure 2, Daimler-Benz drawing W607 below, plots their results.

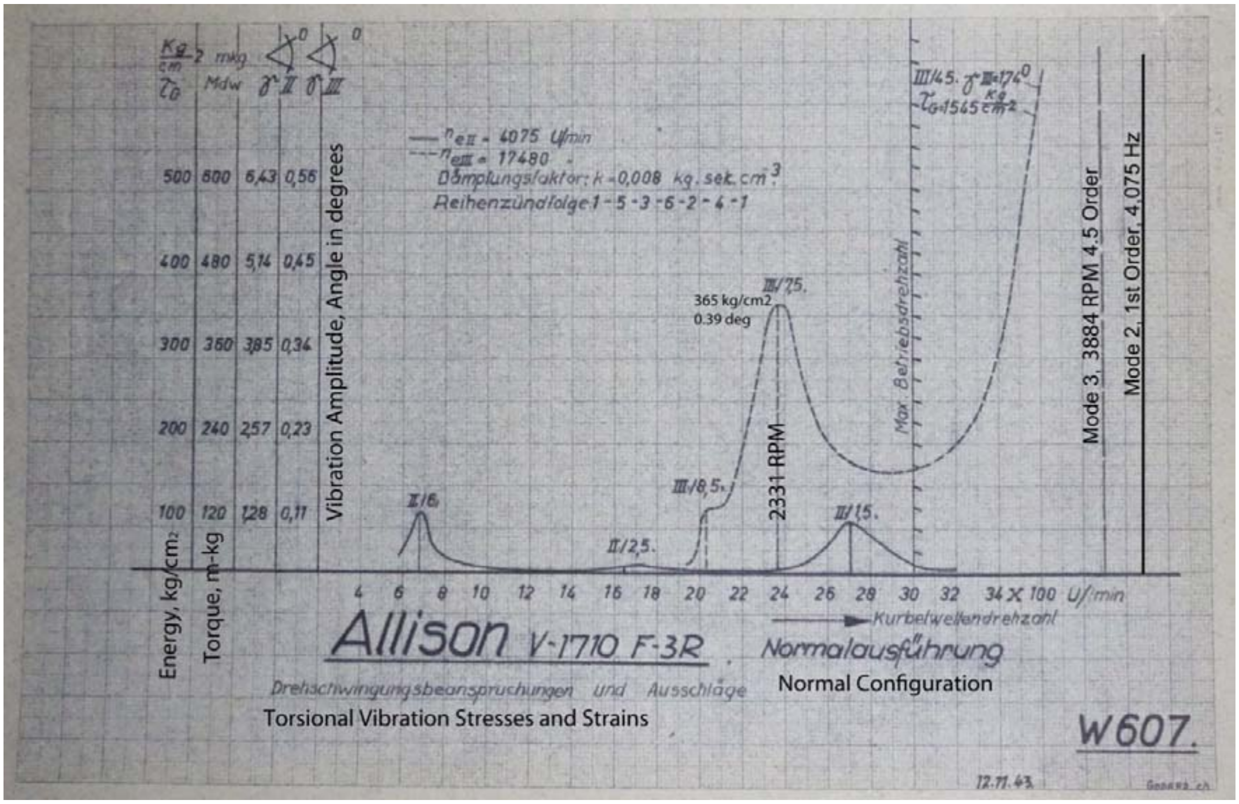

Figure 2. D-B Torsional Vibration Orders for V-1710-F3R Modes 2 and 3. D-B calculated first Order peak for Mode 2 was 4,075 Hz, while the Mode 1, 1st Order natural frequency was determined to be 1,350 Hz. Significantly, the predicted amplitude of the Mode 3 (3-node) peak, 7.5 Order, of 0.39 degrees exceeds the US Army allowable vibration amplitude limit of 0.25 degrees. Allison measured this vibration and met the Army acceptance criteria, hence we have to consider that the Daimler-Benz calculation methods may not have been entirely accurate. Values in excess of 0.25 degrees have been known to result in broken crankshafts.

Section XII. *Torsional vibration calculation with larger masses and increased elastic lengths.*

This investigation showed that with increased masses and elastic lengths, the critical speed is 3,650 rpm, which [for the 4½ order] *is somewhat closer to the 3,000 rpm normal operating speed of the engine, and that the [strong] 3 node, 4½ order peak is present, but has not*

[6] Author's comment: Note that while the three node 4½ order at 3,884 rpm is quite large, the 7½ order peak at 2,381 rpm is also significant. It was because of these two resonances that Allison provided their dynamic pendulum vibration damper, tuned specifically to offset the energy in these peaks.

reached its critical peak [which Allison determined occurs at 3,700 rpm]. See Figure 3, Daimler-Benz drawing W608 below, which shows the results.

Meanwhile, it has been found from the literature [see Section XIV below] *that for the Allison V-1710-F3R, along with the extension shaft to the remote reduction gear engine* [V-1710-E series] *that, for the purpose of uniform equipment, the centrifugal pendulum is also installed. Below is a new investigation* [by Daimler-Benz] *considering a further extension of the quill shaft to the supercharger, with an elastic length of 3,000 cm assumed* [equivalent to 98.4 feet, i.e., a very flexible shaft]. This analysis is based on the original vibration system according to Section X and presented in Section XI above. The apparent purpose of the analysis was to see the effect of a more flexible quill shaft [in the supercharger drive].

Section XIII. *Torsional-Vibration calculation with long Quill Shaft to the Supercharger*

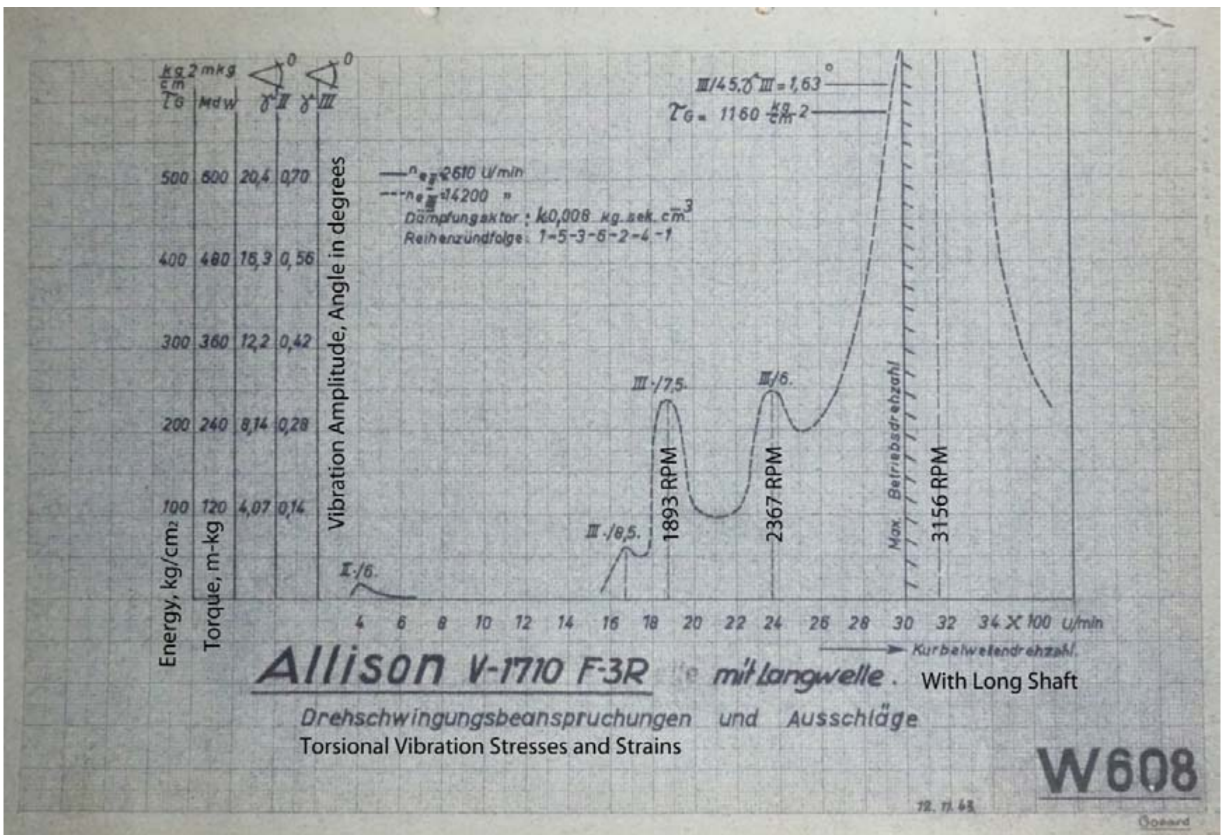

Figure 3. D-B Torsional Vibration Orders for V-1710-F3R with Longer Quill Shaft, Modes 2 and 3

Editors Comment: D-B doesn't clearly state why they did the analysis assuming the longer Quill Shaft. We can speculate that their then current main production engine, the DB 605 may have been having issues and that they were looking to the V-1710 and this study to suggest ways of resolving issues involving development of that engine. Such an effort may have been focused on replacing their hydraulic supercharger drive with a direct mechanical drive.

As described in Section XII above, the longer and more flexible quill shaft has the adverse effect of bringing the troublesome Mode 3, 3-node, 4½ order peak closer to the engines *normal operating speed of 3,000 rpm, with the "shoulder" area well in the operating range, and additionally, reintroducing the 6th order as a troublesome harmonic.*

Section XIV. *The Development of Allison Aircraft Engines (from No. 295 reprint from BMW archive, Translation from SAE Journal Vol. 46, No. 5, page 488-500).*

> Editor's note: all of the material in this section was translated by BMW from the SAE Journal and made available to D-B. It is reproduced here as it was a component of the Daimler-Benz report. The original SAE Journal report is available, in its original English, in numerous archives and on the Internet.

The Allison Engineering Co. was, up until 1930, fully engaged with the production of lead bronze on steel support shell plain bearings and with other various high-quality development and specialized manufacturing work. In 1930, the company was acquired by General Motors Corporation and began the development for the Navy Department an order for one liquid-cooled aircraft engine.

1) In August 1931, the first embodiment of the V-1710 was completed. The exact name was GV-1710-A. Key Facts:

> *650 HP at 2,400 rpm. compression ratio 5.8:1, 80 octane fuel, supercharger gear ratio 7.3:1 and 209.6 mm [8.25 inches]supercharger impeller diameter, weight 436 kg [959 pounds] reduction gear ratio of 1.5:1.*
>
> *The reduction gear pinion was mounted directly on the crankshaft, where it drove the internally toothed bull gear on the propeller shaft, and fitted in the front part of the crankcase. From the bull gear a long compliant auxiliary equipment drive shaft drove the camshaft, the supercharger and all other accessories which were mounted at the rear of the engine.*
>
> *Cylinders were in a pair of mono blocks, and were glycol-cooled for the first time. The first 50-hour endurance run with 650 HP found that some parts needed to be changed; at the same time the supercharger ratio was increased to 8:1, so that in 1932 a 50-hour run with 750 HP at 2,400 rpm was accomplished. (Type V-1710-A2)*

2) In 1933 the above basic engine was redesigned as the V-1710-B as a reversible engine (reversal from full load to full load in 8 seconds) and passed the type test in 1936 for the airships Akron and Macon. Further development to model V-1710-C4 with gear reduction of 2:1, diameter of the supercharger increased to 241.3 mm [9.5 inches], the Weight 510 kg [1,122 pounds], delivered for 700 HP at 2,400 rpm. In 1934 it achieved a 50-hour Endurance test at 800 HP at 2,400 rpm. In the spring of 1935 it was further improved and tested at 1,000 HP, achieved at 2,650 rpm.

3) The V-1710-C1 experiments which now together with propeller in the provided engine mount made on the test bench where varied faults [were found], which eventually became a 1936 redesign and strengthening of the engine to resolve effects of the first mode, single node, 1½ order vibrations.

The original simple propeller shaft was redesigned as a two-piece, with the inner, elastic shaft serving to transmit the torque, the outer to accommodate the propeller bending moment. The inner shaft twists with torsional vibrations compared to the outer shaft. Between the two was placed a [friction type] torsional vibration damper.

Furthermore, the front crank cheek and the front crankpin were strengthened to support the [overhung] reduction gear pinion. The following long trial runs revealed various additional shortcomings.

The induction manifolds and cooling lines were changed, as was the shape of the combustion chamber, with the compression ratio increased to 6:1. The pistons were redesigned for better heat conduction and the piston rings changed. This was done for the desired power of 1,000 hp the speed reduced from 2,650 to 2,600 rpm; as well as the supercharger gears changed from 8.77:1 to 8.0:1. Orifices for metering the coolant between [the] liner and cooling jackets allowed exact adjustment of the coolant flow to the individual cylinders. With the redesign it was possible to use the engines for either Tractor or Pusher installations.

After the above redesign the engine weighed 560 kg [1,232 pounds] (Type V-1710-C6). Following further endurance testing the cylinder head was found to require more reinforcement, while the cylinder head hold-down bolts were redesigned to dampen vibrations within them by close fitting in the holes. The thus developed V-1710-C8 engine in 1937 successfully passed a 150-hour type test at 1,000 HP.

Supercharger improvements made possible an increase in takeoff power to 1,150 HP at 2,950 rpm. Developments led to a reduction of the supercharger gears from 8:1 to 6.23:1 [for turbosupercharged installations] *with simultaneous increase of the cylinder compression ratio from 6.0:1 to 6.65:1 when running on 87 octane fuel, and at the same ratings. Motor weight was 585 kg [1,287 pounds] with the newly introduced Bendix Stromberg injection pressure carburetor instead of the former float carburetor, [Type V-1710-C10].*

New variants emerged:

The V-1710-C9, left-hand running [for the XP-38] *and the V-1710-C13 rebuilt as an altitude rated engine* [for the Curtiss XP-40] *with supercharger gear ratio increased to 8.77:1, [and at this point the engine had] already [been] fitted with the oscillation pendulum* [dynamic vibration damper].

4) Variant V-1710-D with a 1.52 m [five foot long] *propeller speed extension shaft, [which] brought new vibration difficulties, since the previous vibration damper [in the propeller shaft] was [intended only] for Mode 2, 2-node, vibrations, and was not effective on the new engine.*

This resulted in fitting a centrifugal pendulum damper at the rear of the crankshaft. Because of the desire for interchangeability of all parts among models, this change was included in the V-1710-C, beginning with the C13.

Variant V-1710-D [in pusher configuration] *was for the second airplane to fly with the engine, the Bell "Airacuda," while the model C became the standard type as the V-1710-C15 with 100 octane fuel in the Curtiss P-40, an altitude engine with only engine-driven supercharging.* [All previous models had both engine stage and an exhaust turbocharger.]

All further developments were based on the principle that all working parts be as simple and robust as possible, and built as a special unit where possible. So the torsional vibration damper was installed in any case, to the basic engine model both with and without an extension shaft. The crankshaft was in terms of lubrication, completely reversible and both ends were fitted with the same flange.

5) Double motor V-3420, 24 cylinders. By the above measures [standardized interchangeable parts] the assembly was very simplified. The completely reversible crankshaft made it possible, to turn the crankshaft through 180°, whereby when running in opposite direction the same firing order results, so that the same camshafts and induction manifolds can be used for both directions.

Not exchangeable here was the crankshaft housing [crankcase] parts and the main bearings, which differ only in finishing. Even the drive sprocket between the crankshaft and reduction gear pinion is retained.

First the reduction bull gear and the propeller shaft must be capable of double the power. Both of the crankshafts have their own vibration dampers. While the engine worked satisfactorily it was not further developed, as the Plant was fully engaged with serial production of the 12 cylinder engine.

6) Model V-1710-E for 100 octane fuel with 3 meter [ten foot] long extension shaft and remote reduction gear. Again, the vibrational damper worked perfectly for the rated speed of 3,000 rpm.

7) Model V-1710-F with full interchangeability as the E, and with the same supercharger ratio and rated speed 3,000 [rpm], as a "sea level" engine with either right or left-handed rotation, and as an altitude rated engine only with clockwise air screw. Compared to Type C10 it was 13.5 kg [30 pounds] lighter with a 10% smaller cross section.

The propeller axis is in the middle the engine front face. This reduces the size of the engine cowling and places it lower in the chassis. Overall length is 250 mm [ten inches] shorter than for the Model V-1710-C15. In this article it is not mentioned that the Model F, in contrast to the Model C, again has one simple propeller shaft and no torsional vibration dampers within the airscrew shaft. Also the internally toothed ring gear in the Model C [reduction gear] is replaced by an externally driven spur gear on the propeller shaft. The pinion is not mounted directly on the crankshaft but is connected by a short coupling.

The E and F series are being mass-produced since the summer of 1941, replacing the V-1710-C15 at the same time. In the Pre-Report No. 7 of the RLM will be the Two-speed supercharger for Model F mentioned, however, the existing captured engine had only one speed and a single entry supercharger.

Section XV. *Allison Parallel and Series Supercharger Drives*

From an included Daimler-Benz letter of 3-18-1944 from Fauradae:

Subject: ***The influence of the supercharger on the apparatus [accessories] side and propeller side of the engine determines the vibration behavior of aircraft engines.***

Comparing the apparatus-side [accessory section driven, aft end of engine] supercharger drive to the arrangement where the supercharger is driven from the propeller reduction gear, i.e. connected in parallel with the crankshaft. This causes significant differences in the torsional vibration behavior of not only the supercharger itself, but also the total [system], which will be briefly described below.

As is well known, in multi-mass systems, multiple levels of vibration can occur, each of which can be energized by a large number of harmonic forces of gas and mass [dynamic] pressures. The degree of vibration is given by the number of nodes, and in both cases, the nodes pass through the supercharger drive.

Important for the torsional vibration assessment of an engine are usually only the two and three node vibrations; as the one node frequency is too low, [and] the others too high to be dangerous in the speed range. In twelve-cylinder V engines the two-node frequencies between 3,000 and 6,000 cycles/min [50 Hz to 100 Hz], and the three node frequencies, between 10,000 and 20,000 cycles/min [167 Hz to 333 Hz] are considered, in installations where the supercharger is driven from the accessory end of the crankshaft.

The American V-1710 aircraft engine, which has both types of supercharger drives in its various construction patterns [V-1710-C15 long nose and V-1710-E/F], has the two node in series connection [V-1710-F] of supercharger and crankshaft at [a natural frequency of] 4,075 cycles/min [67.8 Hz], and with the parallel connection [V-1710-C15] at 3,600 cycles/min [60 Hz]. In both cases there is a node in the propeller drive, which is different in the supercharger drive.

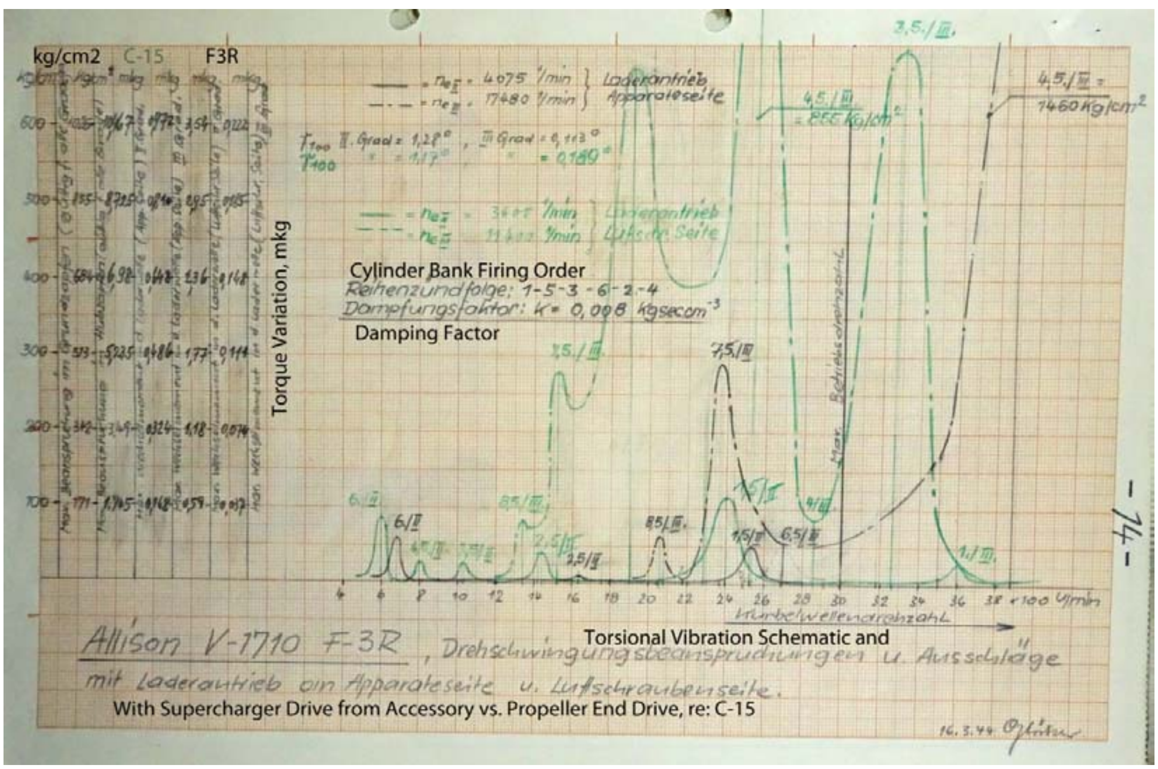

Figure 4. D-B calculated 2-node & 3-node Torsional Vibration Orders in V-1710-C15 and V-1710-F3R

For the three node vibrations, the corresponding [natural frequency] numbers are 17,480 cycles/min [291 Hz] at the rear end of the V-1710-F3R and 11,400 cycles/min [190 Hz] for the parallel [V-1710-C15] arrangement.

As a result of the lower frequencies produced in the case of the propeller end supercharger drive, which further causes low order and often very strongly excited orders in the operating speed range, results in considerably greater vibration stresses for the crankshaft and propeller shaft, compared to the case with the apparatus [accessory section] *end drive* [see Figure 5 below for a depiction]. *In addition, the lower frequencies from other forms of vibration, which can considerably change individual orders in their strength; this is especially true of the three node vibrations in the 12-cylinder V-engine as it places a node at approximately in the middle of the crankshaft, while the propeller end drive has a node between the 1 and 2 crank throws* [note – German practice was to number the cylinders from the propeller end (front) of the engine, while American practice is to number from the rear]. *As a result, the critical order is particularly strongly influenced, which in the first case* [propeller end drive] *produces only slight, but in the second case* [accessory end drive] *very considerable, vibrational stresses. This can readily be seen in the various development stages of the Allison V-1710.*

The V-1710-C15, which began series production in 1939, drove the supercharger from the propeller reduction gear via a long and flexible torsion shaft. At the same time, this model for torsional crankshaft vibration had centrifugal pendulums [at the rear of the crankshaft], which were tuned partly to the 4½ order, and partly to the 6th order, both of the third mode. For new designs, e.g. the V-1710-F3R, they went away from the propeller-end supercharger drive and instead drove it from the rear end of the crankshaft. At the same time, the pendulum 6th order weights were replaced by 7½ order weights, which, even in the earlier V-1710-C15 with a parallel [driven] supercharger, also provided significant oscillatory stresses.

When propeller-end supercharger drive can be accommodated, the torsion bar is almost the full length of the engine and has a large elasticity, whereby the supercharger driven [directly] from the crankshaft experiences torsional vibration stresses. However, a substantial relief is obtained when the torsion shaft rotates at propeller speed, that is not translated, and when the decrease takes place directly on the propeller, for example from the inside [as the internal spur reduction gears of the C15]. *Mostly, however, when driven from the propeller gearbox, which itself still undergoes significant deflections, with a translation into the supercharger drive. In these circumstances, it is therefore questionable whether the not too great improvement in the torsional vibration conditions in the supercharger will justify a sizeable deterioration of the torsional conditions in the crank and propeller shafts.*

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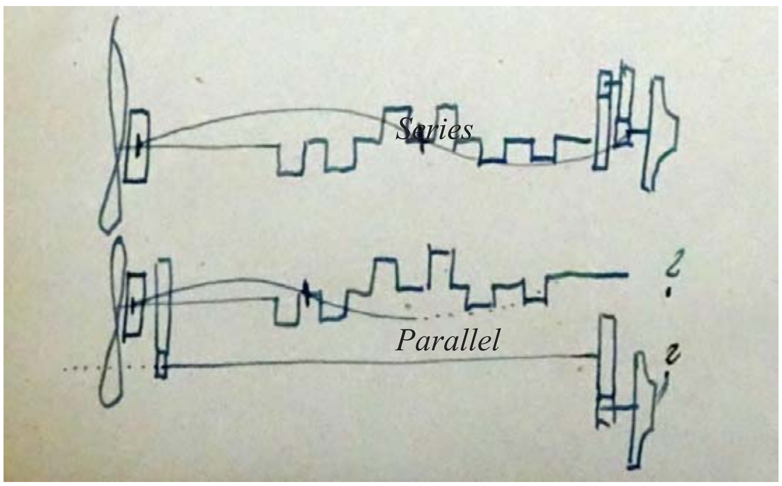
The above letter was followed by one of 30 March 1944 from a Dr. Engr A. Berger:

Dear Mr. Erberhard!
Thank you for your very detailed presentation of the vibrations in parallel and series connection of crankshaft and supercharger.
But I have them intimidations about the supercharger drive at Allison old [V-1710-C15 series] and Griffon [both driven from the propeller end, i.e. parallel drive] *and Merlin and Allison now* [V-1710-E/F], *series drive even further condensed, since I'm not fully knowledgeable about the directions of development of enemy flying machines in ½ to ¾ hours can speak.*
But I notice that you (line 21) say that the second mode [2-node] frequency is higher in a series connection, while the third mode [3-node] frequencies are lower in the parallel connection (line 24ff.).
Is that really so?



Also, I would like to know, where is the third node, mode three, in the propeller-end [parallel] drive since it is probably in the accessory-side in the supercharger gearbox?
I would be grateful if you were the bearer of this return the same day would give the answer.
Best Regards
Your A. Berger

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Dear Doctor! 31March 1944
In my last message of 18 March 1944 on line 24 is actually parallel connection versus connected in series. The higher frequencies are consistently in the series connection, as well as from the following text emerged. The finding in general, the frequencies are lower than in systems in which the branches are connected in series.
In a series arrangement the third mode has one node in the propeller shaft, the second between the 3rd and 4th crank throws, and the last in the supercharger drive [as shown in your earlier sketch]. With a parallel arrangement, the first node is in the propeller shaft, the

second between the 1st and 2nd bearings [i.e. near the propeller pinion gear], and the 3rd in the long shaft driving the supercharger[7].
I expect to arrive on Monday to Mr. Witzky after K.T., so that at this opportunity to answer any other questions could be spoken.

[Signature]

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Comments by Author:

Figure 5 following is from the Daimler-Benz report and provides specifics to the points discussed in the above letters, and clearly shows that first there are very significant differences between the V-1710-C15 vibration environment and that of the later V-1710-F3R with its "series" driven supercharger. For each of these engines the 2-node and 3-node curves are given and the peaks for the various orders plotted.

From Figure 4 above we see that in the V-1710-C15 the 6th and 4½ order peaks, the most energetic of the 3-node peaks, and that the 3-node critical [natural] frequency is 11,400 cycles per minute (190 Hz), hence the peaks occur respectively at 1,900 and 2,533 crankshaft rpm. For this reason Allison tuned the dynamic pendulum vibration damper for these 3-node orders. Two damper weights were provided to dampen the 6th order and four for the 4½ order.

[7] NOTE: the parallel arrangement they are speaking of is as in the V-1710-C15, where the supercharger is driven from the reduction gear. The series arrangement is as in the V-1710-F3R, where the supercharger is driven through a quill shaft from the rear of the crankshaft. Author

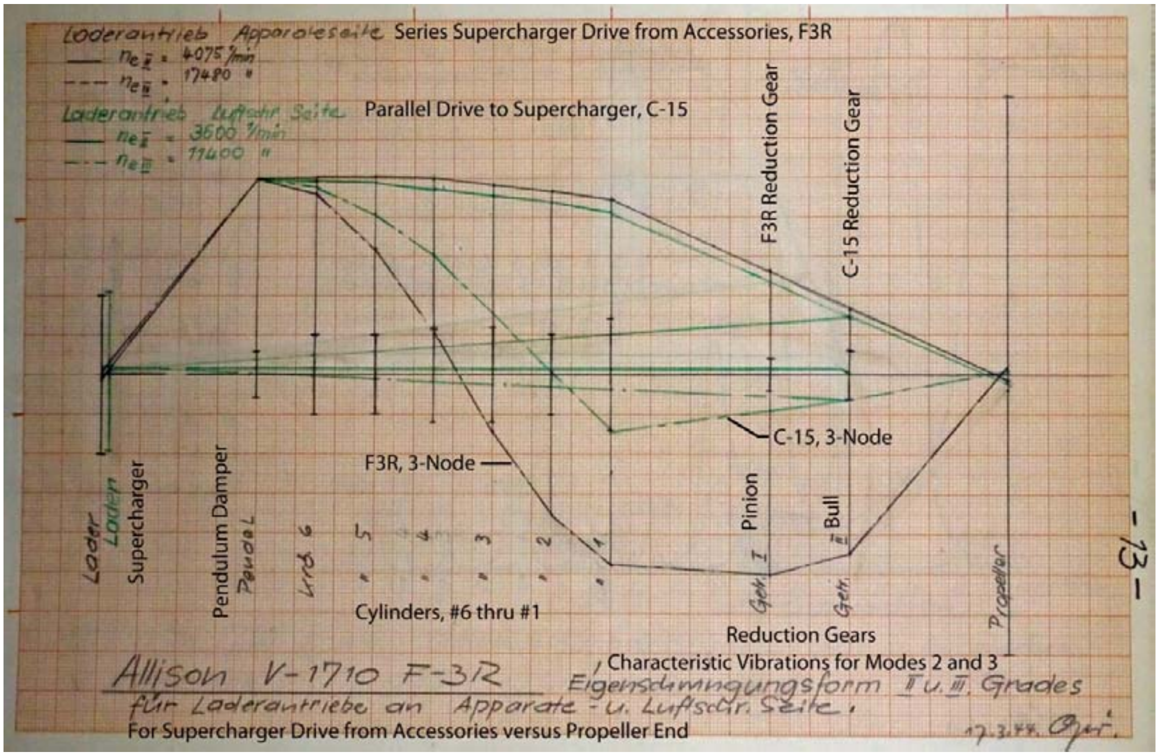

Figure 5. D-B Schematic showing 2-node and 3-node locations for V-1710-C15 and V-1710-F3R. Significantly, the slope of the curve at the intercepts (at the nodes) determines the level of maximum vibratory stress. For the 3-node mode the C-15 stresses are less than for the –F3R. Since the vibratory stress goes as the frequency squared, indicating that the C-15 had a much lower natural frequency for the 3-node case.

Figure 5 shows the Daimler-Benz calculated "wave forms" and nodes [the "zero" crossing points] of the vibrating crankshaft for 2-node and 3-node harmonics, for both the C-15 and F3R engines. Note that for the C-15 the center node lies between the 1st and 2nd cylinders, while for the F3R it is in the center of the crankshaft, between cylinder 3 and 4, as described in the above letters.

For the V-1710-F3R, with its supercharger driven from the rear of the crankshaft, Figure 4 shows that the 3 node 6th order is completely gone, replaced by a 7½ order peak at 2,330 crankshaft rpm, and that the 4½ order has become very energetic, peaking at 3,884 rpm. Even though this peak is far outside the normal operating range of the engine (limited to 3,120 rpm) it is so energetic, though of low amplitude, that its shoulders extend down into the operating range. The mode 3 critical natural frequency for the V-1710-F3R configuration is 17,480 cycles per minute (291.3 Hz), while the mode 2 natural frequency is 4,075 cycles per minute (67.9 Hz). Incidentally, Daimler-Benz calculated that the mode 1 natural frequency was 1,350 cycles per minute (2.25 Hz). For the F3R Allison changed the weights on the dynamic pendulum damper to tune it so that there are three for each order to be dampened, the 4½ order weights are each 1.18 pounds (0.535 kg) and the 7½ order weights are 0.88 pounds (0.399 kg) each. When the conditions are such that damping is occurring the 4½ order weights oscillate as much as +/- 15.5 degrees, and the 7½ order weights as much as +/- 12.5 degrees.

The V-1710-E/F series engines also are fitted with a hydraulic vibration damper, located between the pendulum damper at the rear of the crankshaft and the accessories, including the supercharger. This device only mitigates 1st and 2nd mode vibrations, doing nothing for the powerful high frequency 3rd mode.

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The Daimler-Benz analysis of the V-1710-F3R determined its vibration characteristics, and then repeated the analysis assuming an elongated quill shaft in the series supercharger drive. For specifics, refer to Figures 2 and 3 above, i.e., Daimler-Benz drawings W607 and W608 in Sections X and XI respectively.

Several things become quite apparent when the two cases are compared. Daimler-Benz found that the large 3 mode 4½ order peak at 3,884 rpm shifted to 3,156 rpm and that the contained energy dropped from 1,565 to 1,160 kg/cm^2. This is not particularly good, as the peak is now closer to the normal operating range of the engine, and the shoulder is still very energetic. Also, they found that the 6th order had reappeared at a similar magnitude as the 7½ order. In this case the 6th order peak was found to be at 2,367 rpm, right in the heart of the operating range for the engine.

Allison Vibration Analysis

The Allison Engineering Company cut its teeth designing and building sophisticated gearing and drive systems for industry in the early 1900s, and with the advent of the V-1710 in 1929, focused these skills on its design. From the earliest versions of the engine the design and engineering of every part and component was analyzed for strength and reliability. Subsequent analyses focused on engine and component vibrations and their mitigation. Your author is fortunate to have a copy of Allison Engineering Department Report No. 285, "Detail Design Analysis and Torsional Vibration Analysis of the Allison Model V-1710-E2 Aircraft Engine (Air Corps Type and Model V-1710-17)," dated August 15, 1939.

This engine was the first of the redesigned V-1710-E/F models and was the engine with the ten foot extension shaft that powered the Bell XP-39 aircraft. Significantly, Allison ran an early "workhorse" V-1710-E engine, without dampers installed, at its mode 2 critical speed and after only 55 minutes the crankshaft broke at the #6 main journal.

The following Figure 8 is Allison's chart for the "Harmonic Excitation for Three Modes of Vibration as a function of Crankshaft Speed." This chart is directly comparable to Figure 2 above, Daimler-Benz chart W607. It clearly shows that the single node vibration orders contain very little energy, while the two node orders are likewise of minimal interest, excepting the 6th order at 470 rpm. Normal idling speed for the engine is greater than 600 rpm, so the effects of this order are easily accommodated by the hydraulic vibration damper.

The mode 3, three node natural frequency is 16,680 cycles per minute (278 Hz) and results in vibrations that are significant, specifically the 7½ and 4½ orders, where the 7½ order peaks at 2,220 rpm, with an energy component of 1,320 in-lb, while the 4½ order peaks at 3,700 rpm and has an energy input component of 5,650 in-lb. Note the 6th order peak at 2,780 rpm, but its energy input is only 600 in-lb, and clearly is not an issue, though it was significant in the earlier V-1710-C15.

Allison calculated that with an assumed amplitude of oscillation of 10 degrees, at 3,000 rpm, the three 4½ order weights (each 1.178 pounds, 0.534 kg) developed and offsetting torque of 2,428 in-lbs, and the three 7½ order weights (each 0.871 pounds, 0.395 kg) a combined 1,994 in-lbs. Completely offsetting the peak energy required oscillations of +/- 15.5 degrees and +/- 12.5 degrees respectively.

Allison used these same weights and supporting pins through all of the V-1710-E/F production, on both the six counterweight and 12-counterweight crankshafts, however for the post-war V-1710-G6 engine the 4½ order weight was slightly redesigned to use larger mounting pins. This was done to provide more contact area between the weights and the pins, thereby resolving the occasional occurrence of galling of the seating surface.

Summary Table: 3 Node Vibration Harmonics for Allison V-1710-F3R							
	Order =>	4½		6		7½	
	Natural Frequency	RPM	Energy	RPM	Energy	RPM	Energy
DB, W607, F3R	17,480 cpm	3,884	1,545, kg/cm^2		None	2,331	365, kg/cm^2
DB, W608, long shaft	14,200 cpm	3,156	1,160, kg/cm^2	2,367	245, kg/cm^2	1,893	240, kg/cm^2
Allison, Rpt#285	16,680 cpm	3,700	5,650 in-lb	2,780	600 in-lb	2,220	1,320 in-lb

The "units" associated with the above energy values are not directly equivalent. Daimler-Benz calculates vibration energy in terms of kg/cm^2, which is usually thought of as a pressure or stress value, however this value is arrived at by taking their "mkg" value and dividing it by the Polar Moment of Inertia, measured in cm^4, of the crankshaft main journal and multiplying by other dimensional factors and a constant. The term "mkg" is equivalent to meter-kg, which is a torque moment, distance times force. In this respect it is similar to the value Allison determines for vibrational energy, inch-pounds (in-lb), which is also distance times force, a torque or moment. The kg/cm^2 term is unique in that it is relative to a particular volume, or mass of steel, however it is not necessarily equivalent to normal handbook material specifications.

Figure 9 following, shows schematically the equivalent inertias and rigidities of the Allison power train for the V-1710-E2, an engine model very similar to the V-1710-F3R, the major differences being the presence of the extension shaft in the former, while the latter had an integral reduction gear, and a change in supercharger gear ratios from 6.44:1 to 8.80:1 respectively. Otherwise the supercharger drive, dynamic vibration dampers and crankshafts are identical, and the reduction gears, propeller shaft and propeller all very similar.

Allison presents its "rigidities" in terms of "millions of inch-pounds per radian," i.e. the amount of applied torque needed to "twist" the shaft one radian; as there are 2 pi radians per 360 degrees, one radian would involve twisting through an angle of 57.3 degrees. Air Corps design criteria typically limited the twisting due to harmonic vibrations to less than +/- 1 degree at the aft end of the crankshaft. To twist just the crankshaft, with a rigidity of 8.5

million in-lb per radian: one degree would require a torque of 148,353 in-lb. At 1,000 bhp the average crankshaft output torque is 24,240 in-lb, however the 3-node harmonic orders can add to this value. Allison determined the rigidity of the crankshaft by empirical means so as to eliminate the need to measure shapes and estimate properties.

Figure 10 below shows the Daimler-Benz calculated elastic lengths and moments of inertia for the V-1710-F3R engine power train. While the numerical values are obviously different than those from Allison, and shown in Figure 8, much of the difference is due to the different units used to express the results. Allison units for inertia are pound-inches per second squared, lb-in-sec^2, while German practice is to use kg-cm/sec^2. The difference being that Daimler-Benz used mass terms rather than force; with these adjustments the results are directly proportional, excepting that the Daimler-Benz results differ somewhat, primarily due to the values they calculated for the modal natural frequency of the crankshaft and shown in the above Summary table.

In early 1943 the U.S. Army Air Corps measured the torsional vibration characteristics of the Allison V-1710-73(F4R), the follow-on to the V-1710-39(F3R), and similarly, fitted with 8.8:1 supercharger gears.[8] From a dynamics and ratings perspective the two models are identical, with the exception of a shorter coupling, PN 35472, between the crankshaft and reduction gear pinion.

Figure 6 below shows the results of the testing, and in particular, the fact that compared to the V-1710-F3R analysis with undampened 4½ and 7½ orders, the operational engine, fitted with the tuned pendulum dynamic absorber, has eliminated the potentially damaging effects of the mode 3 torsional vibrations. The Air Corps assessed the remaining 1½ and 2½ orders, noting that the natural frequency for the 2½ order at 2,300 rpm was 5,760 cycles per minute [96 Hz] and the 1½ order peaks at 3,050 rpm at 1,056 cycles per minute [7.6 Hz]. Given these frequencies it is apparent that they are specific to the 2-node and 1-node modes of vibration, for which the hydraulic damper is responsible to mitigate, saving the accessories, including the magneto, camshafts, generator and supercharger from excess torsional vibration. Note that the torsional vibrations were measured on the starter shaft, which is connected directly to the crankshaft, i.e., it is does not benefit from the hydraulic damper.

[8] Torsional Vibration Survey of the Allison V-1710-73 Engine in Combination with Curtiss Propeller, Drawing No. 89301-3. Power Plant Laboratory Memorandum Report Serial No. ENG-M-57-503-797. February 9, 1943.

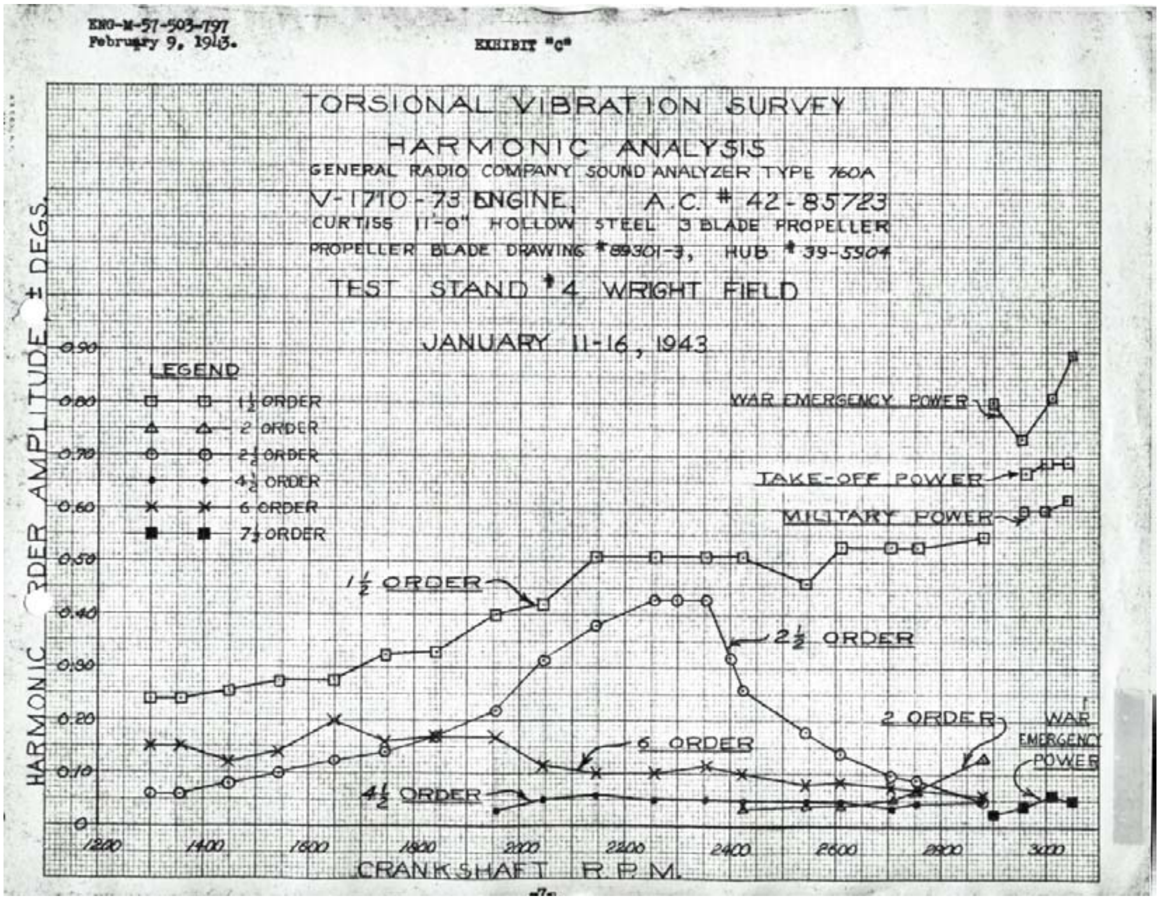

Figure 6. V-1710-73 Engine Vibration Spectrum, Note low amplitude 4½ and 7½ order harmonics

Rolls-Royce Merlin Vibration Analysis

Unfortunately, we have very little specific information about the design of the Rolls-Royce Merlin regarding torsional vibration. An archivist for the Rolls-Royce Heritage Trust says that in all of his years of work with the Rolls-Royce archives, he has yet to come across any documents dealing with the issue. In 1943 the U.S. Army Air Corps was interested in putting the Merlin in the Bell XP-63 airplane. To this end they tested a V-1650-3 two-stage Merlin engine connected to a P-39 drive train and did perform a series of vibration tests. Concurrently, they had requested that Packard/Rolls-Royce provide their torsional vibration analyses, however none was forthcoming.

We do have the information provided by Daimler-Benz, given earlier in this paper, that shows, of the four engines they considered, the Merlin 61 had the least rigid crankshaft, by a large margin, with a polar moment of inertia of 424 cm^4 compared to the V-1710-E/F series at 571 cm^4. The Air Corps testing found that the Merlin did have an undamped mode 3, 3-node, 7½ order harmonic at 2,350 rpm, and that the natural frequency of the crankshaft was 17,640 cycles per minute [294 Hz].

The following schematic, Figure 7, shows the torsional modes for the Packard/Rolls-Royce Merlin V-1650-3, when tested driving a P-39 extension shaft reduction gear and propeller.[9] The similarities to the V-1710-F3R modes, as shown in Figure 5 above, are striking, and the effects of the 3 meter [ten foot] extension shaft are clearly seen.

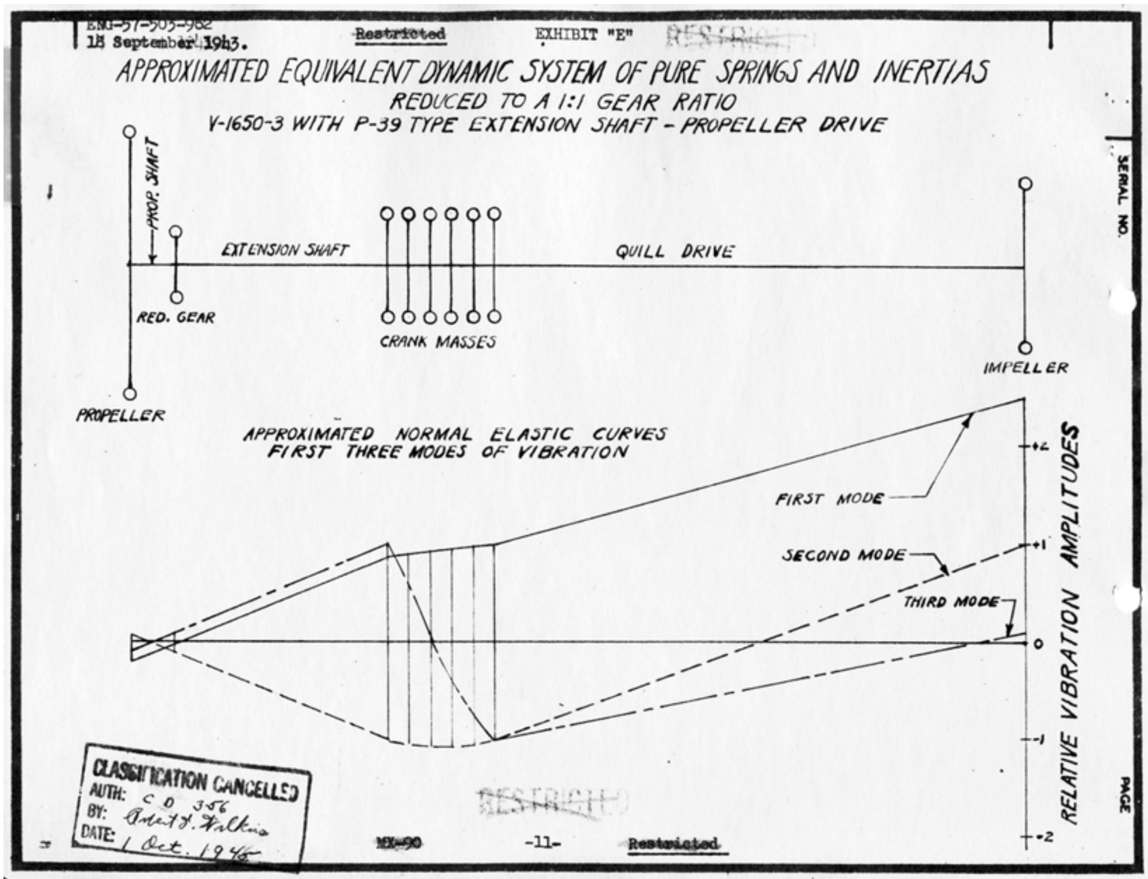

Figure 7. Rolls-Royce 2-Stage V-1650-3 Merlin Torsional System, with Modes

In more recent work, AEHS member Robert Raymond has performed Holzer analysis on the Merlin and his comprehensive report is available on the AEHS website, see www.enginehistory.org.[10]

SUMMARY

The Daimler-Benz analysis and report provides an interesting and independent assessment of the vibration issues inherent in the Allison V-1710 V-12 engine. The V-1710 engine is successful, and is acknowledged to be one of the smoothest running of the V-12s, (which are categorically known to be inherently smooth running). The additional effort expended by Allison to assure that torsional vibrations were positively resolved is quite apparent.

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[9] Torsional Vibration Survey Conducted on the Packard V-1650-3 Engine Equipped with P-39 Type Propeller Drive, Power Plant Laboratory Memorandum Report ENG-57-503-962, 18 September 1943.

[10] An Examination of the Torsional Vibration Characteristics of the Allison V-1710 and Rolls-Royce Merlin Aircraft Engines by Robert J. Raymond, 2016.

The German V-12s began as large capacity and relatively slow running engines. To produce the needed horsepower the result was that pistons were large, strokes long and the overall engines were heavy, largely due to the large and heavy reciprocating parts. Crankshafts had relatively large bearings to support the heavy kinematic and gas pressure loads, and this resulted in large diameter journals. By way of comparison, the post-war Rolls-Royce Merlin 130 at takeoff rating had a maximum crankpin bearing load of 3,398 psi, while the early Jumo 211B bearing pressure was 3,019 psi, and the Allison V-1710 at post-war ratings was a respectable 2,875 psi.

Crankshaft stiffness was largely a result of these factors affecting the dimensions of internal components. A primary driver for these engine designs was the requirement to use 87 octane fuels, as it was only late in the war when Germany began using, in limited quantities, 100 octane fuel. Engines using these later fuels were run at higher crankshaft speeds, however the engines had very low time between overhaul, largely due to the adverse bearing and friction loads with the engine when operating at the higher ratings. With the low specific ratings of the early engines the only way to make power was to increase the size of the engine.

As to vibration issues, natural frequencies are proportional to the square root of stiffness divided by mass. Stiffness is higher with a larger crank, but so is mass and no crankshaft can survive running at a strong critical speed.

It is still speculative as to how and why the Merlin was not hindered by torsional vibrations. It is entirely possible that their built-in crankshaft flexibility accommodated torsional vibrations without exceeding the fatigue limits or overloading the engines components, and thus they built the reputation of the Rolls-Royce organization and the Merlin.

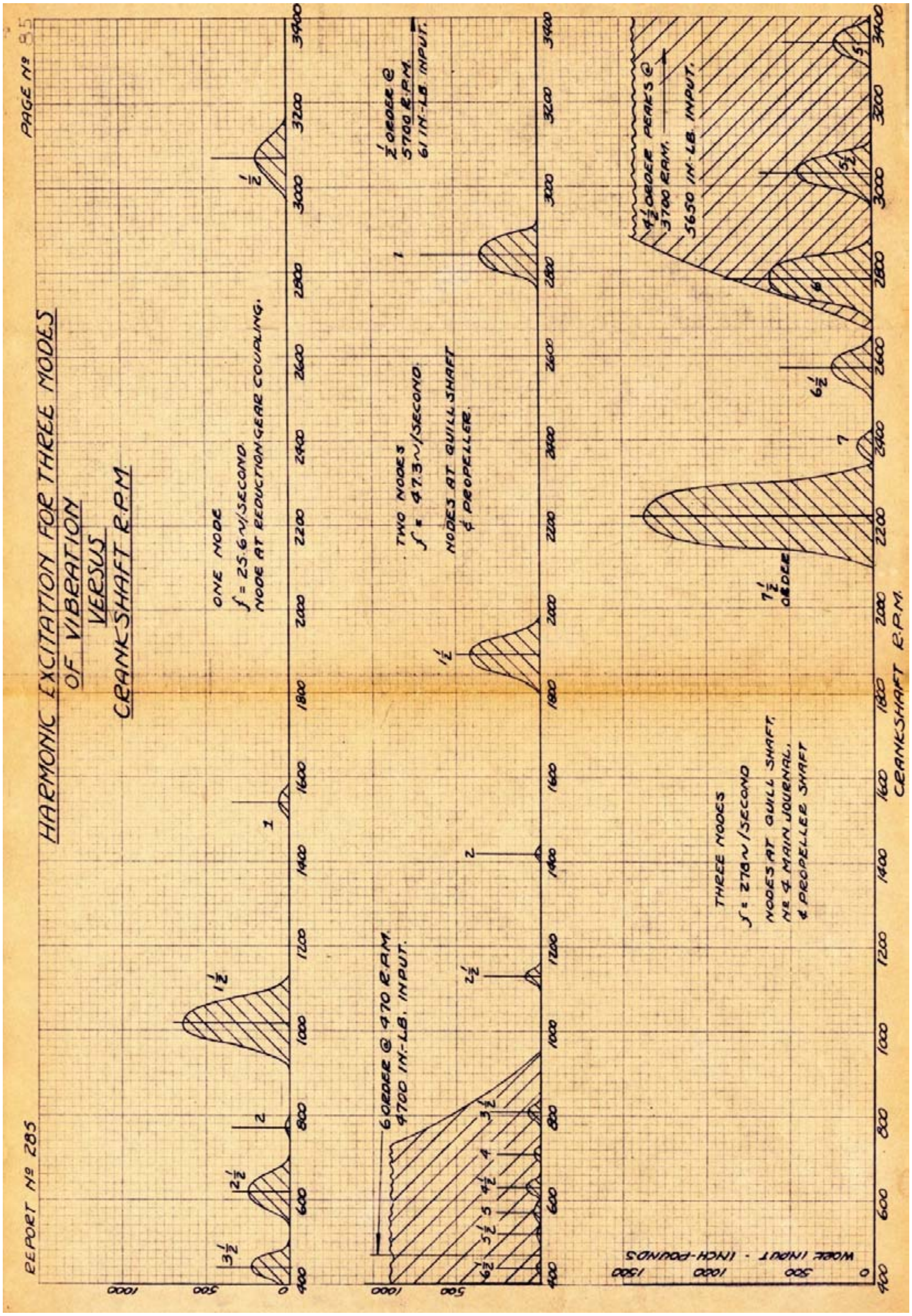

Figure 8. Allison calculated Harmonic Vibration Spectrums for V-1710-E2, with Extension Shaft

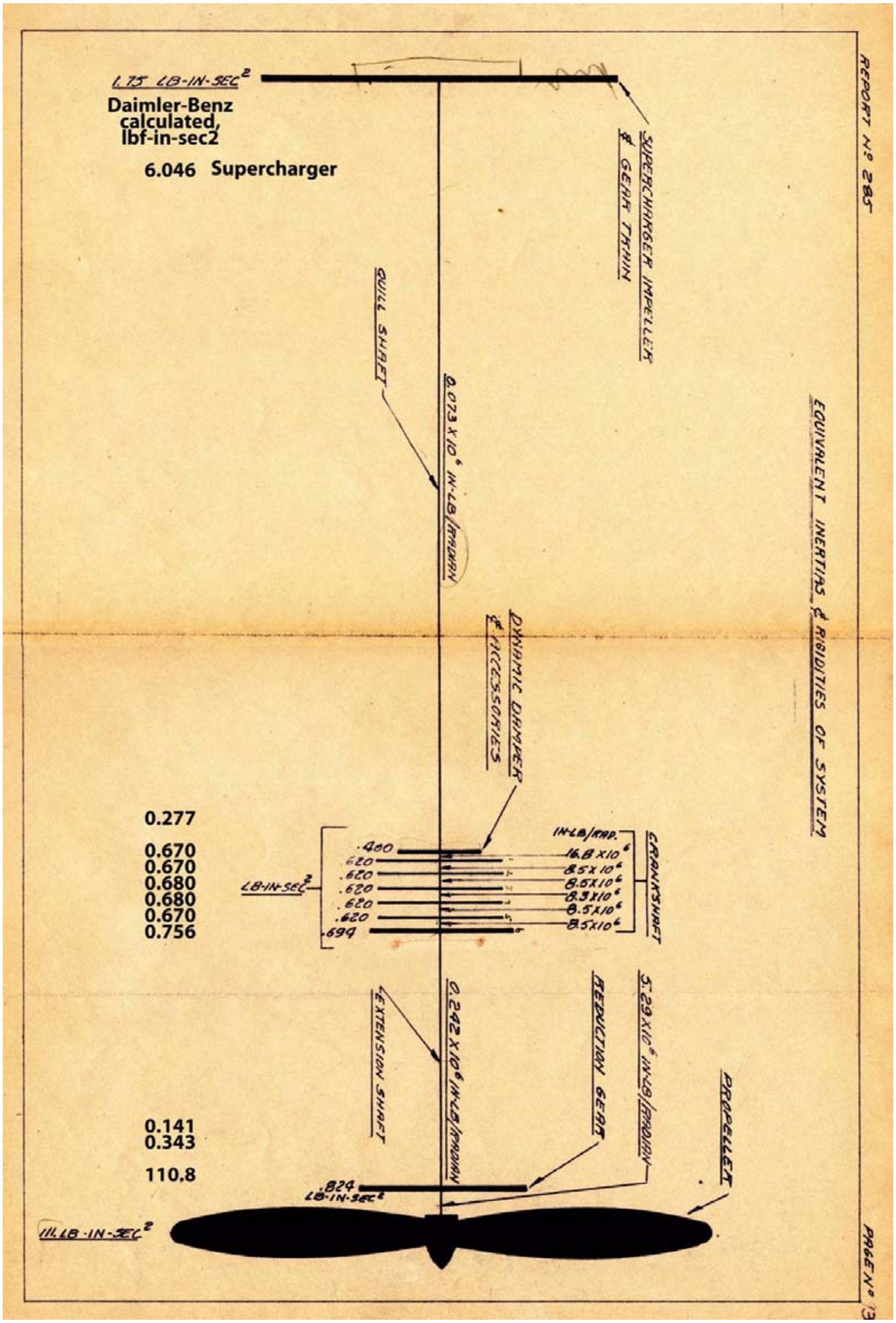

Figure 9. Allison calculated Inertia and Rigidities for Allison V-1710-E2 Power Train. Daimler-Benz calculated moments of inertia tabulated for comparison

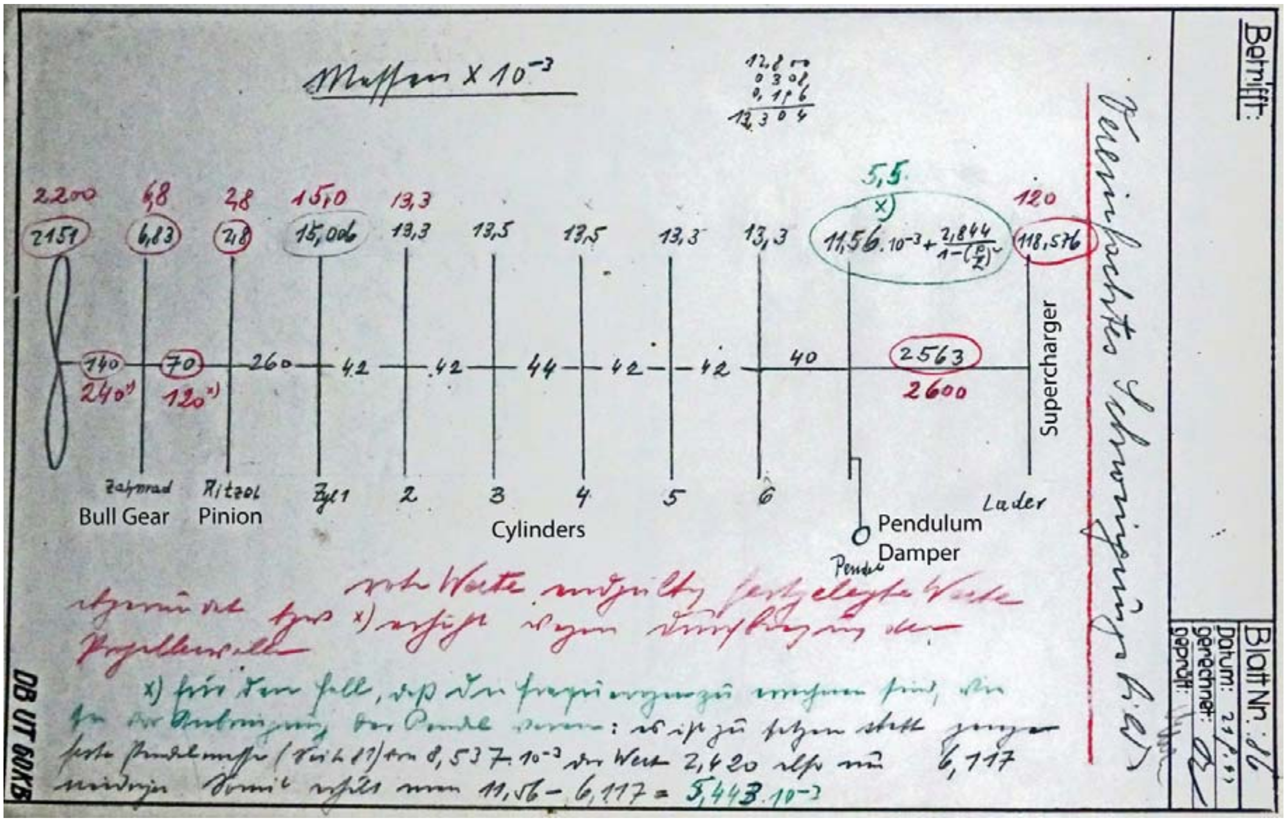

Figure 10. D-B Working schematic of V-1710-F3R Drive Train showing Elastic Lengths and Component Inertias