3 Torsional Vibration

Crankshaft torsional vibration has been a problem with aircraft engines since before World War I. Crankshaft torsional vibration happens because each power stroke tends to slightly twist the shaft. When the power stroke subsides, the crankshaft untwists. One would think that something as massive as a crankshaft would not twist significantly, but any piece of metal always deflects a bit when a force is applied, and when large amounts of power are generated, the forces can become huge indeed. The effects of torsional vibration can be amplified by a phenomenon called torsional resonance. Each crankshaft design has a natural torsional frequency like the note of a ringing bell or sound of a vibrating guitar string. If this natural frequency coincides with the torsional frequency of the crankshaft, the effects can be devastating, resulting in broken crankshafts, lost propeller blades, sheared accessories, and stripped gear trains.

One of the first major scandals in British aviation began in April of 1917 and involved torsional vibration. Granville Bradshaw, chief designer of ABC Motors, Ltd., secured a production contract from the British Air Board for a new engine, the Dragonfly. Bradshaw was a better salesman than engine designer. The Dragonfly had not even run at the time it was procured. When it did run, it was a miserable failure because Bradshaw had managed to design its crankshaft with a resonance exactly in the operating range. By the time the contract was cancelled, 1147 of the engines had been built. This episode upset British air-cooled engine development for years.[1]

The problem of crankshaft torsional vibration in American radial engines appeared almost simultaneously in Curtiss-Wright, Pratt & Whitney, and Lycoming radial engines. This was due to the use of controllable-pitch propellers that were heavier than previous wood and fixed-pitch metal propellers. This increased the effective propeller inertia and brought the crankshaft resonant frequency down into the engine operating range. Lieutenants Howard Couch, Orval Cook and Turner A. Sims, working at Wright Field in Dayton, Ohio, first identified the difficulty.

The problem became really serious in 1934 when the geared Wright R-1820 began breaking propeller shafts. E. S. Taylor of Massachusetts Institute of Technology became involved in the problem and in 1934 and proposed the puck-type damper to Curtiss-Wright. This damper, depicted in Figure 3.1, has a thick disk resembling a hockey puck rolling inside a large hole in the fixed counterweight.

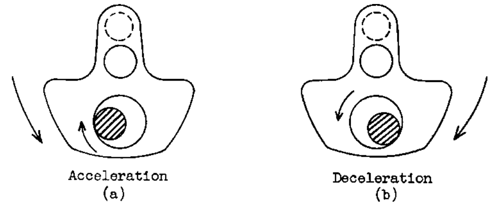

Figure 3.1 Puck-type Damper **(Pratt & Whitney)**

Curtiss-Wright employed Roland Chilton, a prolific designer of many aviation engine and accessory mechanisms. Chilton immediately designed a pendulum mechanism that was vastly superior to Taylor's puck-type damper. See Figure 3.2. Chilton received a U. S. patent for his design, which is called variously the "Chilton damper" or "bifilar damper". Three months after Taylor proposed the damper to Curtiss-Wright, they were delivering engines equipped with it.

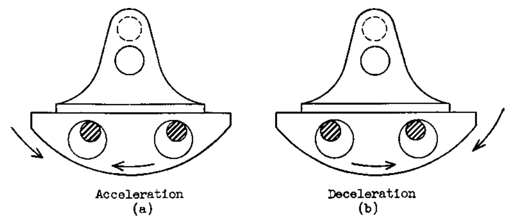

Figure 3.2 Chilton Damper **(Pratt & Whitney)**

The patent situation, however, turned out to be most involved since two French engineers, Salomon and Sarazin, working independently, were earlier in conception. According to Taylor, "Salomon was the first to understand the principle of the pendulum damper." Also, "Sarazin had designed a device almost identical with Chilton's and was in contact with Hispano-Suiza."[2]

The Chilton damper had much better vibration-reducing characteristics, but this would not be evident for years. Since Curtiss-Wright held the patent for the Chilton damper, Pratt & Whitney was left with the Salomon or puck-type damper. This was suitable for the earlier, smaller radials but would be pushed to its limits in the R-2800 and eventually replaced entirely.

Just as E. S. Taylor became the principal vibration consultant to Curtiss-Wright, another M.I.T. Professor, J. P. Den Hartog, became a consultant to Pratt & Whitney. Den Hartog who would later literally write the book on mechanical vibrations, contributed both theoretical knowledge and instrumentation experience. Den Hartog also insisted that the correct terminology was "tuned absorber" instead of "damper". A damper converts movement to heat, while a tuned absorber temporarily stores energy, and then later returns it to the system without producing any significant heat.

When work began on the R-2800, torsional vibration was becoming better understood. The Army had even issued a Torsional Vibration Specification that set a maximum value of 0.50 degrees. Engine designers had learned to make crankshafts large enough so that natural resonance would fall outside the engine operating range. But as engine power increased, even a small percentage of total engine power that became resonant could do damage. Initially, the R-2800 design lacked any mechanism for addressing torsional vibration. One can only guess that the designers chose the simplest configuration, hoped for the best, but were prepared to redesign if necessary. And redesign they did. Trouble appeared almost immediately.

Robert E. "Bob" Gorton got in on the ground floor of R-2800 vibration problems. Gorton was born December 5, 1915 in Norwich, New York where he grew up and attended Norwich High School. Like many of boys of his era, Gorton had been inspired by Charles Lindbergh's solo flight from New York to Paris in 1927. Gorton had a keen interest in aviation and built model airplanes in high school. Also like many boys of his era, Gorton was faced with real challenges when it came time for college – the country was in the midst of the Great Depression. Fortunately, Gorton placed well in the Regents' examination and was awarded a tuition scholarship to Rensselaer Polytechnic Institute.

Toward the end of his senior year at RPI, Gorton was again faced with a shortage of money. He had a summer job at Pratt & Whitney, but needed support to complete his Masters degree. Gorton did something that was unprecedented for the time – he convinced Pratt & Whitney to finance his Masters study in vibration, and in return, agreed to a work-study program. Pratt & Whitney got its very first engineer with actual college training in vibration issues. The relationship was destined to be long and fruitful. Gorton's diligent testing and instrumentation contributed greatly to getting all of Pratt & Whitney's reciprocating engines developed. He and his team invented new types of instrumentation to meet the challenge of each new problem. When jets arrived, Gorton continued to develop innovative approaches to instrumentation of turbine wheels and other gas turbine components.[3]

Gorton initially worked with W. H. Sprenkle in the Test and Instrumentation Department. When Sprenkle moved on to other things in 1939, Gorton took over the department and grew it into a large organization. Test engineers had to be quite creative in the design and implementation of vibration instrumentation. It was a science in its infancy, and the problems had to be solved as they went along. Gorton joined Pratt & Whitney at the same time it acquired a Sperry-MIT torsiograph, serial number 2.[4]

The torsiograph, depicted in Figures 3.3 and 3.4, consisted of a lightweight axle that was attached directly to the vibrating shaft, usually at the rear end of the engine crankshaft. Suspended on ball bearings around the axle was a heavy seismic element that, except for very light springs, was free to rotate. The relative angle between the axle and seismic element was measured electrically. Once in motion, the seismic element tended to stay in constant motion. If the axle were undergoing torsional vibration, the positional difference between the axle and seismic element would be recorded on a 35mm filmstrip.[5] Later analysis of the record could isolate individual frequency and amplitude of torsional vibration. A typical statement from this analysis would be something like "a 4.5X torsional resonance of +/-1.36 degrees was detected at 2000 RPM". This means that when the engine was run at 2000 RPM, the crankshaft twisted 1.36 degrees back and forth at a frequency four and one-half times the rate of crankshaft revolution.

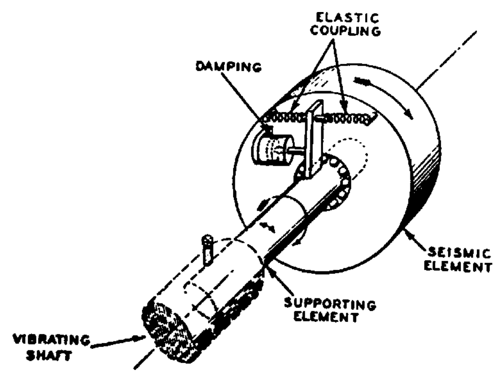

Figure 3.3 Torsiograph Mechanical Components (Draper [6])

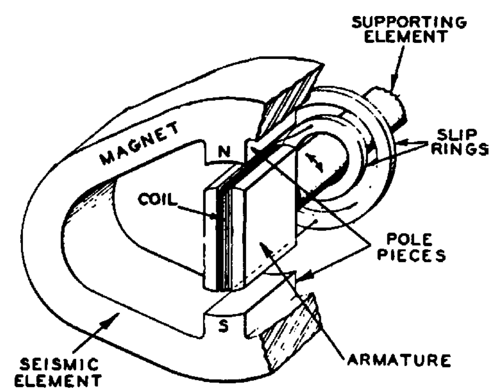

Figure 3.4 Torsiograph Electrical Components (Draper [7])

The Discovery of Torsional Vibration Problems

Most torsional vibration problems occurred either on the propeller or accessory end of the crankshaft. It was here that large inertia loads from the propeller or supercharger and their associated gear trains reacted with the natural torsional variations of the crankshaft. The new R-2800 was about to start high-power runs, and the test engineers wanted to assure that as more and more power was extracted, the engine would stay together. To do this, it would be run with the torsiograph attached to investigate its vibration characteristics. This was done with a wooden test club, a large propeller calibrated to dissipate a given horsepower at a certain engine RPM. Similar tests would be done when metal flight propellers were eventually fitted. Each combination of engine, propellers, and reduction gear had to be tested, since it was impossible to predict when or how a particular vibration problem would be encountered. Nearly all of the vibration testing was done on just three experimental R-2800 engines – Experimental Serial Numbers X-78, X-79, and X-83.

To clarify the rather complicated discussion of R-2800 torsional vibration issues, the story of problem identification is presented chronologically while the solution to each of these problems will be discussed separately.

Sprenkle and Gorton started their investigation of the vibration characteristics of engine X-78 on the last day of January 1938. Everything looked good up through 2000 RPM, but a bearing failure prevented completion of the test.[8] In Gorton's words, "Everything worked fine as long as we stuck with the wooden clubs."[9]By February 16, the engine had been rebuilt and the test was continued. Now a slight crankshaft torsional resonance was observed, but Sprenkle thought it safe to operate up to 2400 RPM. Sprenkle's concluding paragraph would prove prophetic: "The natural frequency of the system is not sharply defined, although it appears to be approximately 90 cycles per second. Vibration frequencies from 5.5 cycles per revolution to 2.5 cycles appeared in order over the speed range, indicating the presence of all orders with no [resonant] excitation at any frequency."[10] When the engines were later run on dynamometers and when metal props were tried, all orders of vibration present would be troublesome.

Sure enough, by the middle of March, one of the test engines had sheared the accessory drive shaft. This shaft connected the rear of the crankshaft through a gear train to the supercharger, oil pumps, magnetos, starter, generator, and everything else behind the power section of the engine. This particular failure had happened on a test dynamometer, a large electric motor that absorbed and measured engine power. The dynamometer also had the ability to drive, or "motor" the engine without the engine actually running. When the engine drove the dynamometer, it was called "firing". Each dynamometer had a unique set of vibration characteristics. It was not unusual that vibration problems would arise when the engine was coupled to the dynamometer. It was Sprenkle and Gorton's job to find an acceptable operating range that would allow testing to continue without destroying the engine.

This activity got under way on March 22, 1938 using the standard 2:1 propeller reduction gear. Very serious resonant vibration existed at speeds below 1500 RPM, making it unsafe to operate the engine on the dynamometer below this speed.[11] With the need to continue testing looming over everyone, it was decided to remove the propeller reduction gear and see if the same vibration difficulties persisted when the engine was connected directly to the dynamometer. No vibration improvement was realized. More work-arounds were suggested, including the installation of pendulum dampers on the dynamometer drive shaft coupling and placing master rods twenty degrees apart[12]. Neither was very appealing. The pendulum damper would be another thing to design, test, and debug. Further, it would be specific to the R-2800 requiring installation and removal from the dynamometer as other engines were tested. The alternative rod placement would have required tearing the engine down and rebuilding it for each dynamometer run. As a result, it would have been a different engine altogether, with different internal organization and vibration characteristics.[13]

In spite of the engine/dynamometer interface problems, other testing proceeded, including testing of different reduction gear construction.[14] The engineers were still at a loss to explain vibration in some operating modes when others were so trouble-free. Runs with the wooden test club continued to indicate very little vibration, but this was decidedly not the case when metal flight propellers were fitted. Neither was it the case when second-order linear vibration difficulties began to surface. Some of the fixes proposed for the linear vibration problems affected the torsional behavior of the engine. Each new idea had to be investigated from the point of view of both torsional and linear vibration modes[15]. While safe to operate, this prop/engine combination when run above 2100 RPM, exhibited excessive first-order torsional vibration as well as a decreased crankshaft natural frequency.[16]

On July 2, 1938, a test was run to determine the effects of relocating the master rod spacing to 180 degrees (cylinders 6 and 15). This was a shot in the dark done in conjunction with linear vibration tests in an effort to reduce the excessive second order linear vibration. Not only was linear vibration unimproved, but second-order torsional vibration became excessive.[17] The original 100-degree master rod spacing had been selected to reduce second-order torsional excitation from unbalanced inertia torque. It is not surprising that the 180-degree master rod spacing failed.

Initial tests using both wooden test clubs and metal flight propellers were done with S.A.E. No. 60 propeller shaft size. In an effort to reduce weight, the propeller shaft was redesigned for a S.A.E. No. 50 shaft size. This was disastrous from the start. Running with a wooden test club, the crankshaft natural frequency deteriorated from 5200 cpm to 4600 cpm. First-order torsional amplitude went from 0.30 degree to 1.02 degrees. With the metal flight propeller, vibration was even worse. Crankshaft natural frequency was reduced to 4400 cpm and troublesome 1X, 1.5X and 2X torsional resonance peaks appeared. This was all the result of reduced stiffness in the smaller propeller shaft.[18] But the weight reduction afforded by the smaller propeller shaft was important and the change was there to stay. In addition to all their other troubles, the engineers now had yet another problem.

Despite the torsional vibration difficulties that continued to unfold, some progress was being made on the linear vibration front. Experiments with counterbalance weights running at twice crankshaft speed were bearing fruit. [19] But crankshaft torsional vibration was making the task of designing suitable drives for these counterbalances exceedingly difficult. In an effort to isolate the counterbalances from the crankshaft, a drive train featuring a number of rubber buttons had been designed. Unfortunately, this addition of the second order counterbalances had increased the crankshaft torsional vibration values at some speeds and had further deteriorated the crankshaft natural frequency to 4000 cpm. The most troublesome was a 1X vibration that peaked at 2300 RPM.[20]

From September 2 through 10 of 1938, a series of tests were conducted on a new counterbalance drive incorporating leaf spring to isolate crankshaft torsional vibration. The leaf spring drives, while an improvement over the ones with rubber buttons, were ultimately not successful. However, important headway was made during these tests toward understanding some of the vibration. For the first time, it was postulated that a three-blade propeller running at one-half engine speed caused the 1.5X torsional vibration. There also seemed to be some contribution from the test house itself, because vibration measurements were inconsistent when different engines were run at the same time as this X-78 R-2800 test engine. Hoping that some of the torsional vibration that had been observed was vibration of the engine as a whole, someone finally got around to measuring the torsional behavior of the entire engine. The results of this, however, were not good. It was found that all of the vibration was in the crankshaft, reduction gearing, and propeller shaft. The engine itself was only exhibiting 0.10 degree of torsional vibration.[21]

During this same testing period, engineers from the Hamilton Standard Propeller Division of United Aircraft conducted the first metal flight propeller blade stress measurements. Hamilton Standard had pioneered the use of carbon strain gages in the study of propeller vibration. Carbon composition radio resistors had been ground into thin sections that could be cemented to propeller blades. Hamilton Standard engineers had developed the bonding techniques and slip rings necessary to collect dynamic vibration data from rotating propeller components. It was upon this basis that R. E. Gorton and his team later developed instrumentation for internal components on operating engines.[22] The propeller blade stress measurements were not at all good. It was found that strong 4.5X resonance existed with this engine/propeller combination. The vibration gave rise to propeller blade stresses in excess of 11,500 PSI, nearly three times the maximum acceptable value.[23]

In early January of 1939, W. H. Sprenkle moved on to other duties at Pratt & Whitney, leaving R. E. Gorton in charge of all vibration testing. Fortunately, Gorton had gotten his first assistant, Albert R. (Al) Crocker the month before.

Al Crocker was born on May 28, 1914 in Higganum, Connecticut, the son of a power company electrician. Crocker always had an interest in aviation, and by the time he got to East Hartford High School, knew he wanted a career in either aviation or radio. In spite of the guidance counselor's advice otherwise, Crocker pursued his aeronautical dreams. After graduating from high school in 1931, he visited the Pratt & Whitney employment office two or three times per week. Finally in December, Crocker was given a job polishing rocker arm adjustment screws. Meanwhile, Crocker had gotten a scholarship to New York University. He continued to work summers in the Pratt & Whitney Assembly and Test Departments. Crocker graduated in 1936 with a degree in Aeronautical Engineering and in 1937 obtained a Master's Degree for his work on the problems of radio shielding aircraft spark plugs. As a full-fledged engineer, Crocker continued at Pratt & Whitney in Production Test and eventually Experimental Test with Gorton. He worked R-2800 valve-bounce problems, instrumentation of supercharger impellers, and vibration problems on both air-cooled radial engines and liquid-cooled experimental sleeve-valve engines. Crocker left Pratt & Whitney late in 1939 to join the vibration group at Martin Aircraft.[24]

After nine months of vibration testing, Sprenkle and Gorton had established that the R-2800 had unacceptable torsional vibration at 1X, 1.5X, 2X, and 4.5X. The good news is that things would get better from this point as engineers methodically found solutions to each problem. There would be false starts and bad assumptions, but the job would get done.

Solution of the 1X and 2X Torsional Vibration Problem

Since solutions to 1X and 2X torsional vibration problems are related, both vibration modes are discussed together.

Although the 1X torsional vibration had primarily been a problem when operating the R-2800 on the dynamometer, it was large enough in magnitude to potentially damage propellers and engine accessories. Thus, a solution was sought which would reduce the magnitude of 1X torsional vibration, if not eliminate it outright. By October 11, 1938, a double-link pendulum damper, presumably of the sort proposed by Taylor[25], had been constructed and was ready for testing. The double-link damper lends itself mechanically to lower 1X frequency. Since Rolland Chilton of Curtiss Wright owned the U. S. patents for the slickest pendulum damper available at that time (the bifilar damper), it is reasonable to assume that Pratt & Whitney had to make do with the double-link damper. Unfortunately, this damper design was a waste of time. It had persistent problems with link bearings that quickly galled and produced enough friction to render the damper inoperative. Improved oil supply and increased bearing clearance did not help.[26]The plain bearings in the links were replaced with needle bearings, but tests in early November yielded no better results. Improper assembly of the rear second-order counterbalance and failure of the front second-order counterbalance drive hampered these tests. The 1X damper was completely ineffective in diminishing 1X torsional vibration and was abandoned. No satisfactory explanation was advanced for its failure.[27]

As it turned out, the main factor contributing to the 1X torsional vibration was master rod spacing. In the original experimental test engines as well as the "A" and "B" series production engines, master rods were positioned 100 degrees apart (in cylinders 8 and 13) to reduce the effects of second-order inertia torque. While this was advantageous from the perspective of reducing 2X torsional vibration, it was the worst possible master rod location for 1X torsional vibration. In spite of this, 1X torsional vibration in the "A" and "B" engines came in just under the limit imposed by the Army's specification. As engine power increased in the later models, this was no longer the case.

Beginning with the "C" models, master rods were located 20 degrees apart (in cylinders 8 and 9) and the crankshaft was fitted with a 2X torsional vibration damper on the front crank cheek. The 20-degree rod placement is best for reduction of 1X torsional excitation, and the 2X torsional damper removes the unwanted effects of secondary inertia torques.

Solution of the 1.5X Torsional Vibration Problem

Testing in early September of 1938 began to shed light on the nature of 1.5X torsional vibration. This particular harmonic had been especially elusive. It would appear in a test, and then be absent in a nearly identical test. A number of theories were advanced to account for the 1X behavior. A prime candidate was propeller blade interference of a three-blade propeller running at one-half engine speed. There was also speculation that interference from other engines operating in the test house was affecting vibration measurements of the experimental R-2800s being tested. On October 20, 1938, an engine was run outside the test house, but the 1.5X vibration remained. While this test ruled out test house effects, there was still doubt about whether propeller interference with the ground and engine was the main cause of vibration.[28]

On February 24, 1939, a serendipitous thing happened. During a routine torsional vibration run on a new propeller, a large 1.5X torsional vibration suddenly appeared. The engine was checked, and it was discovered that the #5 cylinder was misfiring. The spark plugs were replaced, and the 1.5X vibration disappeared. This was the first hard evidence that misfiring could cause the quirky 1.5X vibration that had come and gone in the past [29] In later tests, engines were routinely fitted with individual temperature probes on each cylinder to detect misfire.

In the final analysis, there was also merit to the argument of interference between the propeller and engine. Later engines abandoned the 2:1 reduction gearing for uneven ratios that eliminated the problem of a propeller blade interference frequency resonating with an engine vibration frequency.

Solution of the 4.5X Torsional Vibration Problem

During the first week of October, 1938, additional propeller blade stress measurements showed conclusively that the most troublesome 4.5X vibration was the result of unequal crankshaft windup at the firing frequency of the two 9-cylinder banks. Several solutions were proposed and analyzed. The most obvious solution was the inclusion of a 4.5X crankshaft torsional vibration damper, but there was some concern that while this would remove the 4.5X vibration component, it would worsen the 3.5X, 4X, 5X, and 5.5X components. Also proposed was a scheme to isolate the propeller and crankshaft using a flexible coupling and another scheme to centrifugally couple the crankshaft to the propeller, thus isolating crankshaft vibration from the propeller. This heavy and complicated approach was never implemented. [30] A third proposal was to investigate the possibility that excessive rear propeller shaft bearing clearance was allowing the propeller shaft to whirl, exacerbating the 4.5X vibration and hence the propeller blade stress. Parallel efforts were begun to explore all three threads

On December 21, 1938, tests were run on an engine with a quill shaft[31] installed between the reduction gear and propeller shaft. It was hoped that by flexibly coupling the propeller and crankshaft 4.5X crankshaft vibration could be isolated from the propeller. This was not to be. In addition to very high torsional vibration on the order of nine to ten degrees, propeller blade stress at a frequency 4.5 times crankshaft speed was still present and unacceptably high. Gorton proposed an innovative solution consisting of a tuned leaf-spring drive for the accessory section tuned to the natural frequency of the propeller quill drive that would allow the accessory section to act as a dynamic vibration absorber. [32] While clever, another solution was ultimately developed, and this proposal was never implemented. However, the concept would prove useful during several other tests that Gorton oversaw.

The R-2800 propeller shaft is supported at two points – at the front in the thrust bearing and at the rear in a plain bronze tail bearing inside the front main journal of the crankshaft. The front crankshaft journal has a 0.005-inch cold clearance in its bearing and can be driven about inside the bearing in a whirling motion. It was thought that this whirling motion might be transferred to the propeller shaft, not only causing the propeller to constantly change planes of rotation, but also resulting in uneven meshing of the gear teeth in the planetary reduction gear. It was conjectured that supporting the rear of the propeller shaft on the engine crankcase would stop this whirling.

Several schemes to eliminate the supposed problem were tried in January of 1939. Although a slight decrease in torsional vibration was achieved, propeller blade stress remained unaffected. It was decided that no benefits were obtained that warranted the added mechanical complication.[33]

By December 3, 1938, the crankshaft on engine X-78 had been reworked to include a 4.5X torsional vibration damper of the single spool type in the rear counterweight. This design along with a variation that included a 4.5X damper in the front counterweight as well, was run for a period of 85 hours from December 3, 1938 through February 14, 1939.[34]. The record differs as to the effectiveness of this arrangement. Meloy states that “Torsiograph and blade stress data showed that the 4.5X damper installed in the rear crankshaft counterweight proved slightly effective”[35]. Gorton is less generous, stating that “The R-2800 engine with 4.5X torsional vibration dampers in the rear counterweight gave lower measured values of 4.X crankshaft torsion than did the engine with no dampers or with the 4.5X dampers in both front and rear counterweights. The reduction in amplitude caused by the dampers was only slightly greater than the magnitude of experimental variations found on successive runs with the standard no-damper engine”. He continues, “None of the 4.5X damper arrangements tested were successful in reducing the 4.5X propeller tip stresses below those measured with the no-damper engine”.[36]

Irrespective of the apparent damper effectiveness, it was a variation of this damper style utilizing two spools that was ultimately installed in all R-2800 “A” and “B” series engines. See Figure 3.5.

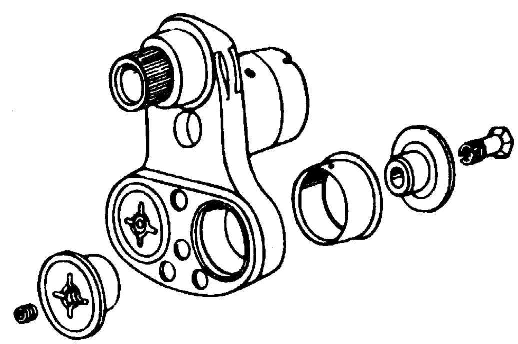

Figure 3.5 Two-spool Damper (Pratt & Whitney)

The fact they were changed for the “C” engines indicates they were less than ideal. Indeed, a test comparing the effects of three types of dampers was conducted in July and August of 1941. In this test, standard Pratt & Whitney spool type dampers were compared with specially built geared-spool dampers and “Chilton” dampers (Pratt & Whitney had not yet established a corporate policy of referring to them as “bifilar” dampers). The geared dampers were used to check the tuning of the spool-type dampers. Since the gear-type dampers were forced to roll and not slide, they gave a check on how well the standard spool-type dampers were performing. Test results indicated performance of both spool-type and geared-spool dampers to be nearly identical. The bifilar dampers were better in both at reducing 4.5X torsional vibration as well as reducing propeller blade stress to acceptable levels.[37]

On May 15, 1939, an engine called “Army No. 1” was delivered for type testing by the Army. The crankshaft of this engine included the twin spool-type 4.5X vibration dampers described above. The Type Test was successfully completed on June 30, 1939, and this included meeting the AN-9504 torsional vibration specification of 0.50 degrees.

When work began on the “C” engine, a new approach was chosen to deal with torsional vibration. The four-counterweight crankshaft of the “A” and “B” series was replaced with a lighter two-counterweight crankshaft. The spool-type 4.5X vibration dampers in the rear counterweight of the “A” and “B” series were replaced with a 4.5X bifilar torsional vibration damper on the rear counterweight and a 2X torsional vibration damper on the front counterweight. Both of these changes were necessary to reliably deliver the higher horsepower of the “C” series. Pratt & Whitney had experimented with the “Chilton” bifilar damper for more than two years before it ever saw its way in to a production engine. The reason for this is unclear, especially in view of the rapidity with which Curtiss-Wright had fielded it in their R-1820 “Cyclone”. One assumes the patent situation clouded the issue and prevented Pratt & Whitney from implementing a clearly superior technology. R. E Gorton recollects a lengthy patent argument between Pratt & Whitney and Curtiss Wright over vibration dampers.[38] In any case, Pratt & Whitney successfully introduced the bifilar damper into the “C” engine and used it thereafter. See Figure 3.6.

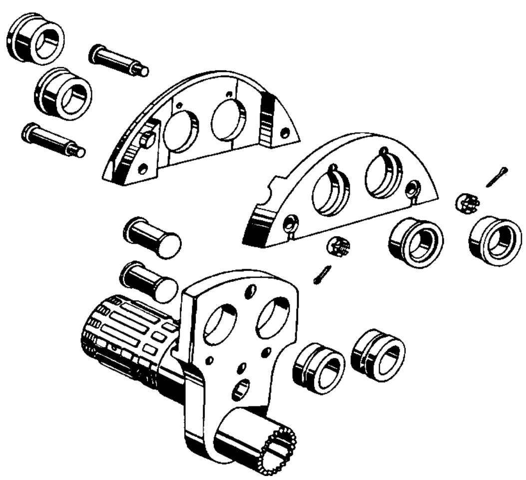

Figure 3.6 "C"-series Damper (Pratt & Whitney)

After the Second World War, Pratt & Whitney was anxious to get back into civilian aviation, and wanted to offer something better than war-surplus engines. The advent of the “CA” series and its corresponding higher horsepower and greater reliability resulted in yet another redesign of its torsional vibration dampers. See Figure 3.7.

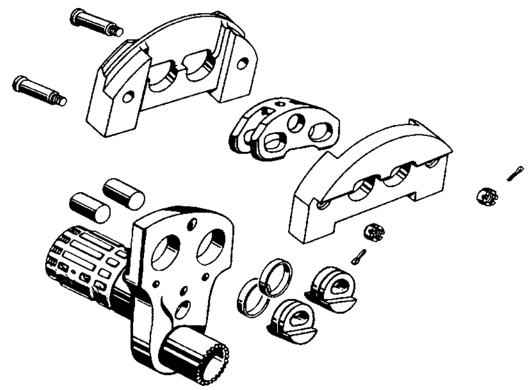

Figure 3.7 "CA"-series Damper (Pratt & Whitney)

Rather than loosely suspending the entire counterweight as had been done in the “C” series, the “CA” engines loosely suspended a much lighter portion of the counterweight mass. This change greatly improved the life of both the damper and of the support pins.[39] This change was particularly useful in assuring that the 4.5X dampers remained tuned throughout their service life, and continued to reduce propeller blade stress as the engine aged.

It is interesting to speculate that much of the torsional vibration trouble in the early R-2800s was a result of the 2:1 reduction. Nearly all 2:1 reduction ratio engines had torsional vibration difficulties, while nearly none of the ones with 20:9, 16:9 or 5:2 had any difficulty. None of the later engines had the 2:1 option. Although the author has never gotten corroboration of this from anyone at Pratt & Whitney, the conclusion is an easy one to draw.

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[1] See Robert Schlaifer, *Development of Aircraft Engines* (Cambridge: Harvard University Press, 1950), 129-131.
[2] See S. D. Heron, *History of the Aircraft Piston Engine* (New York: Ethyl Corporation, 1961), 96-97.
[3] See Robert E. Gorton and R. W. Pratt, “Strain Measurements on Rotating Parts, *SAE Quarterly Transactions Vol. 3, No. 4,* (October 1949).
[4] Robert E. Gorton, telephone interview by author, Huntsville, Alabama, January 11, 1999.
[5] See C. S. Draper, G. P. Bentley, and H. H Willis, “The M.I.T.-Sperry Apparatus for Measuring Vibration, *Journal of the Aeronautical Sciences* Volume 4, Number 7 (May 1937), 282.
[6] Ibid.
[7] Ibid.
[8] W.H. Sprenkle, “Crankshaft Torsional Vibration on R-2800 Engine X-78, *SMR (SMR) No. 393* (February 7, 1938).
[9] Robert E. Gorton, telephone interview by author, Huntsville, Alabama, January 11, 1999.
[10] W. H. Sprenkle and R. E. Gorton, “”Crankshaft Torsional Vibration R-2800 Engine X-78”, *SMR No. 408* (March 11, 1938).
[11] W. H. Sprenkle, “Dynamometer Torsional Vibration on R-2800 Engine X-79”, *SMR No. 410* (March 25, 1938).
[12] Master rods for this series of engine were normally located 100 degrees apart, in cylinders 8 and 13. The suggestion of 20 degree

spacing would have reduced first order inertia torque excitation at the expense of second order torque excitation.

[13] W. H. Sprenkle, "Dynamometer Torsional Vibration of R-2800 Engine X-79 with Direct Drive", *SMR No. 415* (April 12, 1938).

[14] W. H. Sprenkle, "Crankshaft Torsional Vibration of R-2800 Engine X-79 with Wooden Test Club", *SMR No. 418* (April 27, 1938).

[15] W. H. Sprenkle and R. E. Gorton, "Torque Stand Vibration of R-2800 Engine X-79", *SMR No. 420* (April 29, 1938).

[16] W. H. Sprenkle and R. E. Gorton, "Crankshaft Torsional Vibration of R-2800 Engine X-79 with Metal Flight Propeller", *SMR No. 431* (May 17, 1938).

[17] W. H. Sprenkle and R. E. Gorton, "Crankshaft Torsional Vibration of R-2800 Engine X-78 with 180° Master Rod Position", *SMR No. 449* (August 15, 1938).

[18] W. H. Sprenkle and R. E. Gorton, "Crankshaft Torsional Vibration of R-2800 Engine X-78 with 50 Spline Propeller Shaft", *SMR No. 455* (September 9, 1938).

[19] W. H. Sprenkle and R. E. Gorton, "Linear Vibration of R-2800 Engine X-78 with Rubber Drive Second Order Counterweights", *SMR No. 462* (August 22, 1938).

[20] W. H. Sprenkle and R. E. Gorton, "Crankshaft Torsional Vibration Characteristics of R-2800 Engine X-78 with Rubber Drive Second Order Counterweights and Hydromatic 6159-0 Propeller", *SMR No. 474* (September 17, 1938).

[21] W. H. Sprenkle and R. E. Gorton, "Crankshaft Torsional Vibration of R-2800 Engine X-78 with Spring Drive Secondary Counterweights and Hydromatic 6159-0 Propeller", *SMR No. 476* (September 28, 1938).

[22] See Robert E. Gorton and R. W. Pratt, "Strain Measurements on Rotating Parts, *SAE Quarterly Transactions Vol. 3, No. 4* (October 1949).

[23] W. H. Sprenkle and R. E. Gorton, "Linear Vibration of R-2800 Engine X-78 with Spring Drive Secondary Counterweights and Hydromatic 6159-0 Propeller", *SMR No. 479* (October 7, 1938).

[24] A.R. Crocker, telephone interview with the author, (Huntsville, Al, August 2, 1999).

[25] E. S. Taylor, "Eliminating Crankshaft Torsional Vibration in Radial Aircraft Engines, *SAE Journal, Vol. 38, No. 3* (March, 1936).

[26] W. H. Sprenkle and R. E. Gorton, "Crankshaft Vibration Characteristics of R-2800 Engine X-78 with First Order Crankshaft Damper", *SMR No. 488* (November 8, 1938).

[27] W. H. Sprenkle and R. E. Gorton, "Test of First Order Crankshaft Damper with Needle Bearing Links on R-2800 Engine X-78", *SMR No. 490* (November 10, 1938).

[28] W. H. Sprenkle and R. E. Gorton, "Linear Vibration of R-2800 Engine X-79 Mounted on 'O' (outside) Stand", *SMR No. 489* (November 8, 1938).

[29] R. E. Gorton and A. R. Crocker, "Torsional Vibration or the R-2800 Engine with Hydromatic 6159-0 Propeller", *SMR No. 530* (March 14, 1939).

[30] W. H. Sprenkle and R. E. Gorton, "Investigation of the Source of High Propeller Tip Stresses on the R-2800 Engine X-79 with Hydromatic 6159-0 Propeller", *SMR No. 487* (October 31, 1938).

[31] A quill shaft is a torsionally flexible reduced diameter shaft used to isolate torsional vibration from components.

[32] R. E. Gorton, "Vibration Characteristics of R-2800 Engine X-83 with Quill Shaft Drive LE-3162", *SMR No. 519* (February 15, 1939).

[33] R. E. Gorton and A. R. Crocker, "Torsional and Linear Vibration of the R-2800 Engine X-83 with Independently Supported Propeller Shaft Assemblies LE-3174 and LR-3260", *SMR* No. 531 (March 14, 1939).

[34] See George E. Meloy, "Report on History of R-2800 Engine Development, (Pratt & Whitney Aircraft Report No. PWA-192, May 30, 1939), 6.

[35] Ibid.

[36] R. E. Gorton and A. R. Crocker, "Torsional and Linear Vibration of R-2800 Engine X-78 with 4 ½X Crankshaft Dampers Mounted Rigidly on Radial Rubber Engine Mounts", *SMR* No. 544 (March 24, 1939), 2.

[37] R. W. Pratt, "Effect of Various Dynamic Dampers on 4 ½ Order Crankshaft Torsional Vibration of the Two Speed, Single Stage, R-2800-2SBG Engine X-79 with 2:1 Nose", SMR No 790 (December 18, 1941).

[38] Robert E. Gorton, telephone interview by author, (Huntsville, Alabama, January 11, 1999).

[39] W. J. Closs, "Development of the R-2800 Engine, *Service School Handbook*, (Pratt & Whitney Aircraft, date unknown), 29.