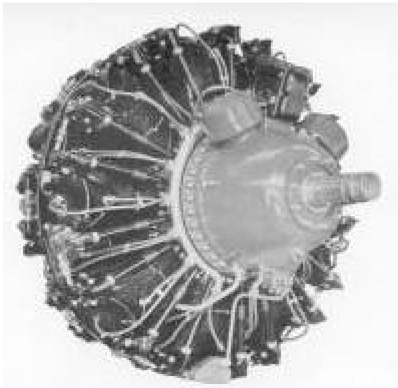


No Short Days:

The Struggle to Develop the R-2800

"Double Wasp" Crankshaft

By Kimble D. McCutcheon

Just prior to World War II, engineers at both Pratt & Whitney and Curtiss-Wright worked feverishly to produce the first air-cooled engine capable of more than 2,000 horsepower. The efforts of both teams were nearly thwarted by severe vibration from unexpected sources. This is the story of how the Pratt & Whitney team, through hard labor and persistence, identified and solved the problems with vibration. The result was one of the most successful engines of all times - the R-2800.

Source:
http://www.enginehistory.org

1 Introduction

Acknowledgements

This article is possible only because of the kind assistance of Mr. Graham White, Aviation Historian, and the many current and former employees of Pratt & Whitney Aircraft. I would like to thank Mr. Richard C. Wellman and Mr. Jesse Hendershot, who made available archival material from which this article is drawn. Thanks are also due to the retired Pratt & Whitney engineers who took the time to undergo my telephone interrogation. In this category are Mr. Albert Crocker, Mr. Gordon Beckwith, Mr. Robert Gorton, Mr. George Meloy, Mr. Elton Sceggel, Mr. Frank Walker, and Mr. Dana Waring,

Dr. A. Gordon Emslie, Mr. James Howell, Ms. Margaret Jacobs, Dr. Greg Swanson, and my wife Shirley were among those who reviewed the early drafts and made many valuable comments. I am very grateful for their efforts.

Preface

All manufactured products, no matter how simple, are the result of considerable engineering effort. In a product as complex as an aircraft engine, there is an enormous amount of engineering, testing, and refinement before the engine meets the combined goals of being light, powerful, fuel-efficient, and reliable. Much of the credit for the success of a new engine goes to the test engineers who methodically and often with great ingenuity, identify and fix the problems with a new engine.

While the iterative work of engine designers may be lost in obscurity, the work of test engineers is nearly always recorded, and provides the best record available of the process of engine perfection. This is the story of a group of dedicated test engineers who took a design loaded with problems and refined it into one of the finest aircraft engines ever built - the Pratt & Whitney R-2800. This is not a criticism of the designers, but rather simply the nature of the engine development process itself. Contemporary engine designers make heavy use of computer simulation and they still rarely get it right the first time. Simulation only works when all of the unknowns are accounted for, and that is rarely the case when something truly revolutionary is being developed. Designers in the thirties had none of these tools, and had to depend on trial-and-error techniques of the test engineers to perfect revolutionary concepts. While this is hardly the complete account of R-2800 development, it does cover an important and historically significant story: the test engineers' efforts in perfection of the crankshaft and the triumph over vibration.

Gordon Beckwith is famous for leading the team that brought Pratt & Whitney's first commercial jet engine, the JT8D, to market. Despite the fact that Rolls Royce was already flying a competitive engine, Beckwith's team produced a better engine that met the customer's noise, bleed air purity, thrust, fuel consumption, weight, and interchangeability guarantees. The engine came through ahead of schedule and under budget.

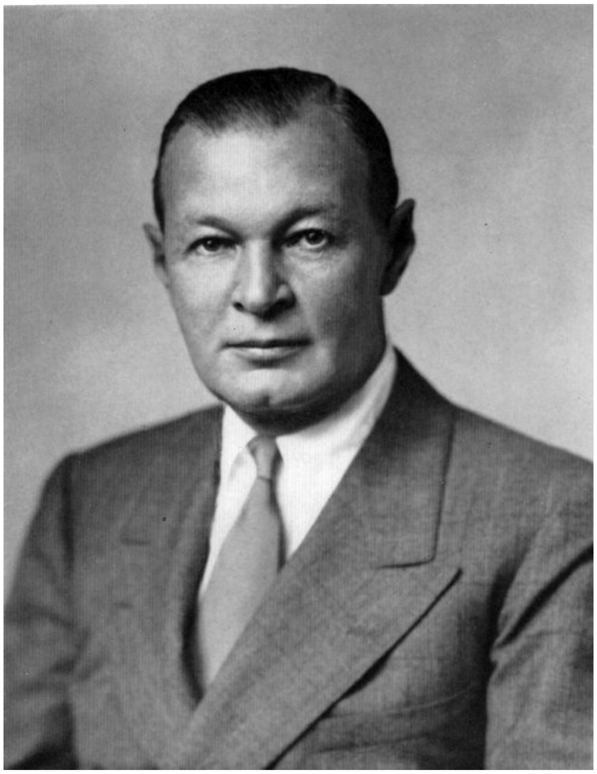
Beckwith came to Pratt & Whitney during the summer of 1939 before his senior year in college and worked in the stock room delivering parts to the engine assembly department. When he returned the next summer as a degreed engineer, he was told he would be working as a designer. The interviewer told him, "If you take this job, you will first become a draftsman. You will have a black oilcloth cover for your drafting table. At the end of the day, when the horn blows, your left hand will roll that cover across your board, you will get up from your stool, and you will go down the stairs and be done for the day." Beckwith objected, "Wait a minute, isn't there any place around here where I can work with the engines, you know, feel them and smell them?" "Oh, you wouldn't want to do that", said the interviewer, "that's the group of people they call test engineers and they don't even know when to go home at night - they stay here for all hours." Beckwith replied, "Hey, that sounds great! How do I get a job down there? So he let me go and talk to Joe Ballard, the head of the Experimental Test Department where I got a job actually working with engines."

Beckwith's story is typical of the many bright and innovative test engineers with abiding interests in aviation who brought the R-2800 to life.

Politics, Management, Corporate Culture, and Competition at the time the R-2800 project was begun

When development of the R-2800 began in March of 1937, Pratt & Whitney Aircraft was a company in trouble, both internally and externally. Pratt & Whitney badly needed a big success, and the R-2800 had to be it. In order to understand the motivations that drove test engineers to solve the complex problems of the R-2800 crankshaft, it is enlightening to review the political landscape of the times, as well as the corporate culture of Pratt & Whitney and the key engine architects' personalities.

The Formation and Spectacular Success of Pratt & Whitney Aircraft

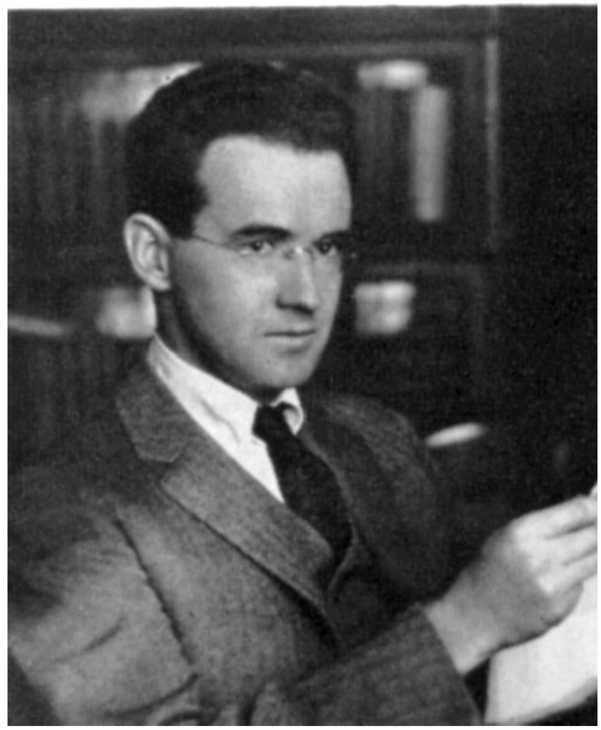


Frederick B. Rentschler, founder of Pratt & Whitney Aircraft (Pratt & Whitney)

Frederick Brant Rentschler turned his back on the family foundry business to become an Army engine inspector at the Wright-Martin Company during World War I. When the war ended, Wright-Martin was reorganized, and Rentschler stayed on to become President of Wright Aeronautical Corporation. But Rentschler was unable to convince the Wright Board of Directors, composed largely of investment bankers with little knowledge of aviation, to fund research necessary for continued improvement of air-cooled engines.[1] Such philosophical differences led Rentschler to resign from Wright on September 1, 1924[2].

By July 23, 1925[3], Rentschler had struck a deal with the Pratt & Whitney Tool Company of Hartford, Connecticut to fund the development of a high-powered air-cooled engine for the Navy.

He had secured the support of his old friend George Mead, the well-respected and highly capable chief engineer of Wright Aeronautical Corporation, as well as a promise from Admiral William A. Moffett to buy such an engine if it were successful. Thus, Rentschler and Mead obtained a controlling position in the Pratt & Whitney Aircraft Company. Other key people at Wright quickly joined forces with Rentschler and Mead. On December 28, 1925, the first Pratt & Whitney "Wasp" ran.[4] The Navy was overjoyed with the result, and, as promised, began to buy engines.



George J. Mead, Vice President of Engineering (Pratt & Whitney)

Bill Boeing also took note of the new engine and had his engineers design the Wasp-powered Model 40A mail plane. When the US Post Office privatized the carriage of airmail in 1927, Boeing won a large part of the transcontinental postal route by underbidding all others by nearly half. His competitors expected him to be bankrupt within months, but instead he made a sizeable profit and assured another market for Pratt & Whitney engines.

By the middle of 1929, Rentschler had organized the United Aircraft and Transport Corporation, a holding company for Boeing and Pratt & Whitney, as well as a number of other aircraft manufacturers, airlines, aviation schools, airports and export trading companies. In a series of stock transactions that were perfectly legal for the pre-crash "Roaring Twenties", Rentschler parlayed his and Mead's meager $500.00 investment into personal fortunes exceeding sixty million dollars. The other participants in United Aircraft also became fabulously wealthy.[5]

The Political Climate Turns Chilly

Healthy profits from exports made up for reduced military spending brought on by the Great Depression. But several factors were to present a real challenge during the 1930s. The McNary-Watres bill forced all airmail carriers to also provide seats for passengers after May 1, 1930. This was no major blow for United

Airlines, but it eroded the bottom line and started United Aircraft thinking of aircraft that could serve both roles. George Mead and Boeing engineers designed the Model 247,a sleek twelve-passenger all-metal monoplane that was to have been powered by two Pratt & Whitney Hornets. United pilots claimed the new design was too heavy and too powerful. In the end, Rentschler settled the dispute in favor of the pilots, and in doing so alienated Mead. To make matters worse, TWA commissioned Douglas Aircraft to build the DC-2 powered by Wright Cyclone engines of the same power class as Hornets. The DC-2 was superior to the 247 in all respects, and its successor, the DC-3 put Boeing out of the transport business for decades and Pratt & Whitney out of the transport engine business for a good part of the 1930s. Faced with lost transport sales, reduced military spending, and unused capacity, United Aircraft's bottom line began to reflect the deepening depression. Only the export picture remained highly profitable.

In 1934, real trouble began in the form of one Senator Hugo Black, a Democrat from Alabama. Black was troubled that in the middle of the depression, with rampant unemployment and hungry children, companies like United Aircraft were making huge profits on ventures originally funded by public money. Never mind that the government had deliberately tried to stimulate an American airmail and aviation transport industry through the Kelly Act of 1925 and McNary-Watres bill. Black thought everyone should suffer, even if they had done nothing illegal. Black publicly embarrassed Fred Rentschler and Bill Boeing on the floor of the U.S. Senate, and ultimately orchestrated the dissolution of United Aircraft. The Black-McKellar bill made it illegal for aviation manufacturers to be involved in air transport or the carriage of mail. Bill Boeing was so angry that he withdrew from aviation endeavors permanently.

Just as the feeding frenzy of the Black Committee was peaking, another grandstander, Representative John Delaney of New York, began investigating alleged profiteering on Navy contracts. Again, while nothing illegal was ever demonstrated, he singled out Pratt & Whitney's 1926 profit margins when research and development for the original Wasp and Hornet was being paid for by Navy contracts. In the end, Congress passed the Vinson-Trammel Act that limited profits to ten percent.

As if Black and Delaney were not bad enough, Congress authorized Senator Gerald Nye of North Dakota to investigate aircraft industry export policies. Again, Pratt & Whitney executives were roasted on the floor of the U. S. Senate, but no wrongdoing was proved. Still, a bill in the form of a neutrality act required exporters to obtain licenses before selling abroad.

To Pratt & Whitney owners and managers, all of this legislation seemed like a way for the Roosevelt New Deal to nationalize the aircraft industry. In Rentschler's view, their "only crime was earning a reasonable profit in a field where most others had lost their shirt"[6]

For the remainder of his life, Rentschler was haunted by accusations, largely by those unschooled in the intricacies of government contracting, of "profiteering" and "treason". The record, however, shows otherwise. Pratt & Whitney, under Rentschler's leadership, quite simply built the engines that won World War II. Pratt & Whitney and its licensees delivered 363,619 engines – fifty percent of all engines produced. Curtiss-Wright contributed thirty-five percent, with the balance coming from all other sources.[7]

While Curtiss-Wright had to be dragged kicking and screaming into subcontracting arrangements, Rentschler agreed to license Pratt & Whitney engines[8] for $1.00 each, and even this paltry fee was later waved[9]. While the papers accused the aircraft industry of "too little, too late", Pratt & Whitney, already producing **four million** horsepower per month from its East Hartford plant alone, was building a new plant in Kansas City, Missouri, and training 400 Missourians to produce aircraft engines.[10] While Curtiss-Wright President Guy Vaughn was defending the R-3350 before the U. S. Senate, aircraft powered by Pratt & Whitney R-2800s were dropping bombs on Europe. This was despite the fact that the R-3350 had almost a two-year development lead over the R-2800.

Rentschler was a shrewd, tough businessman. This attribute saved Pratt & Whitney from a fate like the post-war demise of Curtiss-Wright[11]. Rentschler was also a patriot who just wanted to make a fair profit building a product he believed in – air-cooled engines.

Internal Disharmony

In addition to external problems from the halls of Congress, Pratt & Whitney by 1935 was suffering internally as well. The key people had worked together for years, many having been handpicked by Frederick Rentschler during their previous time of service at Wright. They formed a very capable core group that shared a common philosophy about the superiority of air-cooled engines. When Rentschler focused on running United Aircraft, he appointed Don Brown, who had been with the company since August of 1925, to the Pratt & Whitney Presidency. This act, in conjunction with the Boeing 247 deal, had incensed George Mead to the point he was no longer providing suitable engineering leadership. An additional problem was that the engineering team was spread too thin by the myriad engines under development in the early-to-mid 1930s –the R-985 "Wasp Junior", the R-1340 "Wasp", the R-1535 "Twin Wasp Junior", the R-1690 "Hornet", the R-1830 "Twin Wasp", and the R-2180

"Twin Hornet". A further complication was Mead's fascination with high-speed liquid-cooled sleeve-valve engine, a complete departure from the air-cooled engine roots of Pratt & Whitney.

Stiff Competition from Curtiss-Wright

In the mean time, Curtiss-Wright had fully recovered from the hemorrhage of talent to Pratt & Whitney. Curtiss-Wright had introduced the very good R-1820 "Cyclone", and was progressing well with development of the R-2600 and R-3350, both of which were intended to produce more power than anything Pratt & Whitney had on the drawing boards. In order to restore some order to Pratt & Whitney engine development, Don Brown appointed Leonard S. "Luke" Hobbs to the position of Engineering Manager in 1935.

Technology Dictators: Army and Navy

On the technology front, both Army and Navy were convinced in 1937 that the only really high-powered engines on the horizon would be liquid-cooled. When R-2800 design began, the most powerful air-cooled engine in production was rated at just over 1000 HP. The biggest air-cooled engine even planned at that point was the Curtiss-Wright R-3350, initially rated at 2000 HP. In late 1936 or early 1937, the Navy had issued a request for an engine with a take-off rating of 2300 HP. Pratt & Whitney believed this rating would eventually be reached with an air-cooled engine[12], but knew it would take years to reach the 2300 HP mark.

Then in 1939, the entire Army Air Corps appropriation for fighter engine procurement was given to the Allison Division of General Motors for an order of V-1710s. This came as a devastating surprise to Pratt & Whitney. General Hap Arnold had given Pratt & Whitney a verbal contract, and operating solely on Arnold's word, Pratt & Whitney had started producing an order for R-1830s. Some months into the verbal order, Louis Johnson, the Assistant Secretary of War, actually awarded the order to Allison. According to Arnold, someone at Wright Field had made such a good case for Army support of liquid-cooled engines in fighters that the decision to buy from Pratt & Whitney had been reversed. Worse, it was anticipated that engine power for bombers would go the same route during the following fiscal year. Arnold advised Pratt & Whitney that if it wanted Army business, it had better develop a liquid-cooled engine of its own. Industry rumors attributed the reversal to a "procurement man who had never lost a game of poker to a General Motors representative". The bribery charge was never substantiated, but it made little difference to Pratt & Whitney, who laid off twenty percent of the work force and was seriously considering closing down the entire operation.[13]

Pratt & Whitney had been engaged in experimentation with large, high-powered liquid cooled engines under the direction of George Mead since about 1932. But Mead was disillusioned with Pratt & Whitney, and his health was failing. The liquid-cooled program languished to such an extent that no complete engine ran before testing began on the first experimental R-2800s. Ultimately, Pratt & Whitney would back out of all liquid-cooled contracts with both Army and Navy, freeing engineering resources to perfect air-cooled engines that would far exceed the original 2300 HP Navy liquid-cooled requirement.

The Architects of the R-2800: Hobbs, Parkins, and Willgoos

Hobbs

Leonard S. "Luke" Hobbs was born in Carbon, a Wyoming boom town that no longer exists. He spent his boyhood in Texas, graduated from Texas Agricultural and Mechanical College, and served as an engineering officer with the 42nd "Rainbow" Division in France. After WWI, Hobbs attended Kansas State College for further engineering studies before joining the Army Air Corp as a civilian experimental engineer at McCook Field, Ohio. In a later stint at Stromberg Carburetor, Hobbs' contributions to the modern aircraft carburetor established his reputation in aviation circles.

"Luke" Hobbs (Pratt & Whitney)

Hobbs came to Pratt & Whitney in 1927 where he rapidly gained the reputation as a brilliant engineer and capable manager. He proved his mettle by leading the team that solved the very serious master rod bearing problems that plagued the R-1535 and R-1830. When Pratt & Whitney's bearing vendors were unable to find a solution, Hobbs organized virtually the entire Pratt & Whitney engineering staff to solve the problem. Through much experimentation, hard work, and perseverance, the team invented the lead-silver-indium bearing that was so good it was even adopted by Pratt & Whitney's rival, Curtiss-Wright. After the Second World War, Hobbs would again lead a jet engine development team that would earn the prestigious Collier Trophy in 1953.

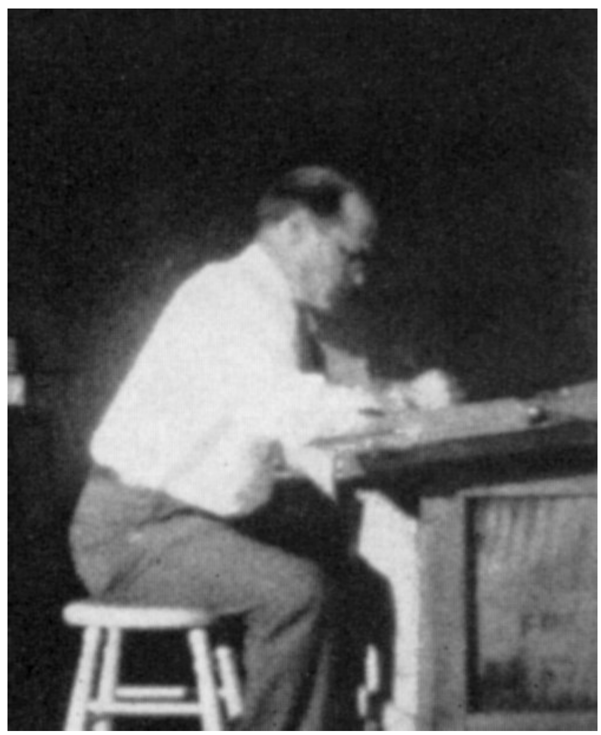
Parkins

Wright Parkins (Pratt & Whiney)

Wright A. Parkins was one of the development engineers who made his mark at Pratt & Whitney during Hobbs' master rod bearing campaign. Parkins, destined to eventually succeed Hobbs as engineering manager, was out of North Dakota by way of Manitoba and Seattle. He left high school to enlist and fight as a doughboy during the First World War. Afterward, he worked his way through engineering school at the University of Washington. Parkins met Hobbs while serving on the engineering staff at McCook Field, and followed him to Pratt & Whitney in 1928.[14] Without exception, those who worked for Parkins vividly remembered him nearly sixty years later. George Meloy described him as "a dynamo" who "made everyone cringe when you couldn't give him a [task completion] date he liked. He wanted two weeks work over a weekend"[15]. Both Frank Walker[16] and Elton Sceggel[17] said that Parkins "struck terror in the hearts of experimental engineers", but Sceggel continues "His bark was worse than his bite, he was strict but tough and that was what was needed". Gordon Beckwith remembers that Parkins was the "star of the show, with ideas, motivation, and leadership. You could leave the plant on Friday, and everyone would be down in the dumps with some terrible problem. Parkins would come in on Monday with eleven new things to try. One might not work, but another would. You didn't know until you tried."[18] One of Parkins' favorite targets was Joe Ballard, who ran the experimental assembly and machine shop. "Parkins made his life miserable", remembers Meloy. But it was Ballard's shop where the problems were and Parkins was a consummate problem-solver. It is no surprise that Parkins pushed Ballard hard to get results.

In addition to his energy, Parkins was well known for his wit. Parkins once guided a visitor on a tour of a test house. The visitor, while watching the blue exhaust flame of an engine under test, brightly remarked, "Actually, Mr. Parkins, you people simply are trying to contain and control fire, aren't you?" "Yes", said Parkins who was having his usual troubles, "and that's simply all the devil has to do in hell, too, as I understand it."[19]

Willgoos



Andy Willgoos in his garage during the summer of 1925 at work on drawings for the "Wasp" (Pratt & Whitney)

A. V. D. "Andy" Willgoos worked with Mead on the initial "Wasp" even before Pratt & Whitney Aircraft was incorporated. Willgoos left Wright Aeronautical Corporation in the summer of 1925 and worked without pay, enthralled by the task of creating a completely new engine in a new organization. [20] It is Willgoos' name in the title block of the first “Wasp” cylinder drawing dated July 25, 1925.[21]

It was Willgoos whose “calm and gentle”[22] nature moderated the high-strung Mead. And it is Willgoos who is copied on nearly all of the early R-2800 experimental test reports. It is likely, but not verifiable, that Willgoos was heavily involved in at least the initial R-2800 crankshaft designs.

Others

There were numerous other people who made significant contributions to high-level R-2800 design. Men like Earl Ryder and Val Cronstedt brought many skills from varied backgrounds. They were key to solution of problems regarding the crankshaft design and vibration issues. They were also unique in that they were consultants during R-2800 development. Unlike Wright, who used a large number of “hired guns” for their engine designs, Pratt & Whitney cultivated people from within who often made long careers of service to the company. Pratt & Whitney also had aggressive recruiting and work-study programs with numerous local colleges and universities. Pratt & Whitney recruited only the best, and summer internships allowed the company to identify those persons who were exceptional. Several of the test engineers came from these summer internships.

The Development Environment: Ideas into Metal

It took considerable fortitude to launch into a new engine development in such a political and corporate climate. New engine designs are always problematical endeavors. This is especially true of high-powered reciprocating engines, with literally thousands of moving, wearing, and vibrating parts. It seems simple to just add another couple of cylinders to each row of an already proven engine such as the R-2180 (a seven cylinder per row engine). In truth, this seemingly simple enlargement of seven to nine cylinders per row was to prove especially difficult for both Pratt & Whitney and Curtiss-Wright.

Luke Hobbs is usually credited with responsibility with the R-2800, and there can be little doubt that he led the team. Mead was almost certainly involved[23], especially in the early studies, but probably became absorbed with his passion for liquid-cooled sleeve-valve engines before the first R-2800 was actually built. Willgoos was heavily involved from conception through at least May of 1942, being copied on numerous reports from the experimental test department. According to George Meloy, who was closely involved with R-2800 development almost from the beginning, Hobbs had “overall responsibility, participated in the development of certain features, but primarily helped to develop solutions to problems”[24]. Hobbs would often show up unannounced to observe the results of an important test. When knotty problems were being solved, engineers would often report test results directly to Hobbs and get direction for the next phase of testing. But after the successful Type Test of the “A” model, Hobbs’ attention was diverted from R-2800 issues to more general management of a progressively larger engineering team.

In Hobbs’ place, Wright Parkins was the “idea man” with the tough job of achieving impossible schedules prior to and during World War II. He drove the team hard and though he was a fair and reasonable manager, the test engineers and personnel in the Experimental Department lived in constant fear of him.

Unlike engine development today, there was no modeling and very little analytical work at the time the R-2800 was developed. According to retired Pratt & Whitney Engineer Larry Carlson, engine designer L. Morgan Porter, using a 20-inch slide rule, did all of the analytical “high science” for the entire company.[25]

Instead, Pratt & Whitney used the time-honored “Run ‘um, bust ’um, and fix ’um” development philosophy. This technique, still an important part of today’s development practices, involves building a device using all available design tools and experience, running the device to destruction, analyzing what broke and why, designing a better part, and repeating the process until the desired level of reliability is achieved. Often, solutions would require the resolution of conflicting objectives. A fix to one part of an engine might introduce an unexpected problem elsewhere. But the ability of Joe Ballard’s experimental test and assembly department to rapidly produce and test new parts allowed these conflicting objectives to be sorted out with great rapidity.

Morgan Porter was instrumental in this process as well. George Meloy recalls Porter’s “unique ability to conceptualize engine configurations, sometimes using both hands as he drew illustrations on the blackboard”[26]. “Porter had a bachelor’s, master’s and professional degrees in mechanical engineering; had been professor of mechanical engineering and taught advanced college courses in aircraft engine design before joining Pratt & Whitney”[27]

The R-2800 crankshaft started out much simpler than it ended up. We can speculate that designers initially built the simplest engine that their vast experience dictated. With testing came problems, the solution of which necessitated complication of the design to include dynamic torsional vibration dampers, second-order counterbalances and the like.

As important as R-2800 development was, one must also bear in mind that other projects were under way which competed for the test engineers’ time. These included: R-1340 and R-1830 vibration tests; testing of the liquid-cooled X-1800 and H-3130 engines; and development and testing of the XR-4360, Also there were many miscellaneous studies and experiments of spark plugs, superchargers, valve mechanisms, and nose gear cases. The engineering and test team had just successfully completed an exhaustive program to eliminate master rod bearing failures, not to mention the R-2180, which was abandoned just prior to production.

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[1] See Ronald Fernandez, *Excess Profits* (Reading: Addison-Wesley, 1983), 27.

[2] See Robert Schlaifer, *Development of Aircraft Engines* (Boston: Harvard, 1950), 185.

[3] See Harvey Lippincott, "The Navy Gets an Engine", *American Aviation Historical Journal* (Winter, 1961), 258.

[4] *Ibid*.

[5] See Ronald Fernandez, 39 - 63.

[6] Ibid, 95.

[7] See*The Pratt & Whitney Aircraft Story*, 141-143.

[8] See Ronald Fernandez, 128.

[9] *The Pratt & Whitney Aircraft Story*, 132.

[10] See Ronald Fernandez, 144.

[11] At the end of World War II, Curtiss-Wright was the second richest company on earth. It never succeeded in gas turbines and ultimately sold the engine business to Deere in 1985.

[12] When work began on the R-2800, Pratt & Whitney had a 1400 HP engine, the R-2180, almost ready for production. Parts from R-2180s were used in early R-2800s.

[13] Ronald Fernandez, 113.

[14] See "The Pratt & Whitney Aircraft Story (East Hartford, CT: United Aircraft Corporation, August, 1950), 114-115.

[15] George E. Meloy, telephone interview by author, Huntsville, Alabama, December 14, 1998.

[16] Frank Walker, telephone interview by author, Huntsville, Alabama, March 29, 1999.

[17] Elton Sceggel, telephone interview by author, Huntsville, Alabama, March 22, 1999.

[18] Gordon Beckwith, telephone interview by author, Huntsville, Alabama, March 22, 1999.

[19] *The Pratt & Whitney Aircraft Story*, 115.

[20] Lippincott, 258.

[21] This drawing, on Willgoos' original drawing board, is on display along with his drafting tools at the New England Air Museum in Windsor Locks, CT.

[22] *The Pratt & Whitney Aircraft Story*, 95.

[23] See Cary Hoge Mead, *Wings Over the World, The Life of George Jackson Mead* (Wauwatosa: Swannet Press, 1971), 183.

[24] George Meloy, telephone conversation during meeting with R-2800 developers, East Hartford, CT, November 5, 1998.

[25] Larry Carlson, meeting with R-2800 developers, East Hartford, CT, November 5, 1998.

[26] George Meloy, electronic mail to the author, October 8, 1999.

[27] Albert R. Crocker, letter to the author, November 1, 1999.

2 Technical Background

Vibration

In order to understand some of the most difficult R-2800 development issues, we must first briefly digress for a quick vibration tutorial. The literature concerning engine vibration is a literal Tower of Babel because each writer has invented his own terminology to describe the phenomenon. Despite the fact it is dated, the author has elected to use the same terminology used in the reports of the engine developers. This terminology is defined below.

Vibration is a motion repeated at regular intervals. It is expressed in terms of frequency or order.
Cycle is a single complete repetition of a vibratory motion.
Period is the time required to complete one cycle
Frequency is the number of cycles completed in a given interval of time, usually one second, but occasionally, one minute.
Order is a convenient means of denoting frequency in terms of crankshaft revolution. For example, a first-order vibration has one period per crankshaft revolution, a second-order vibration has two periods per crankshaft revolution, etc.
Amplitude is the maximum displacement of a vibrating object from its initial position.
Torque is an action tending to produce rotation of an object.
Torsional Vibration is the twisting and untwisting of a shaft resulting from the periodic application of torque.
Linear Vibration is “shake” of the entire engine.
Damp is to dissipate energy from a vibrating system.

For this study of the R-2800 crankshaft, we are concerned with both linear vibration and torsional vibration. In order to understand vibration, one must first be familiar with the forces at work that cause vibration. Most engine vibration is a result of unbalanced forces inside the engine, predominately inertial forces arising from non-rotating parts as they change direction, or the power pulses from each cylinder as it fires. Both are of interest in exploring the problems of crankshaft development in large engines.

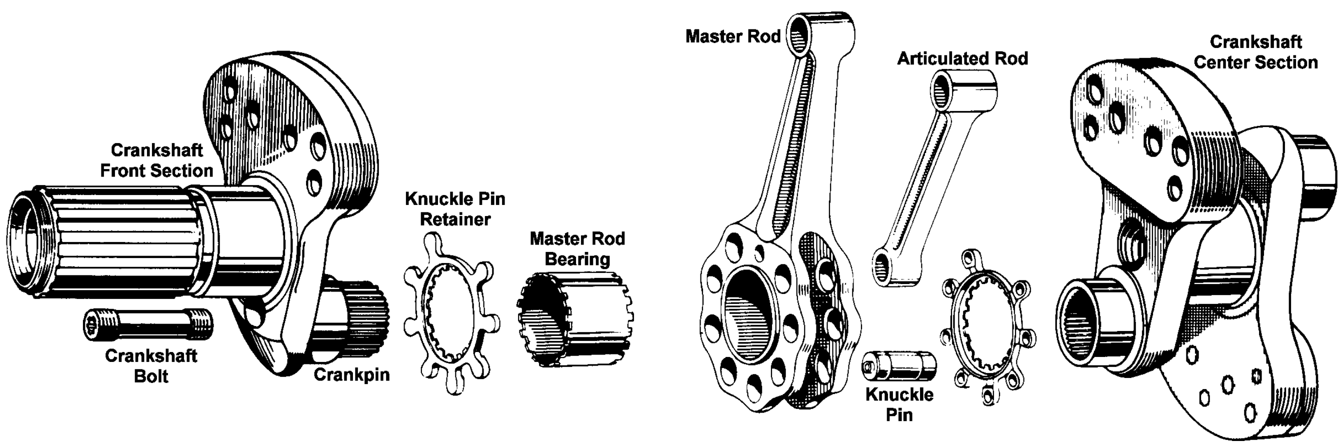

Figure 2.1 Radial Engine Crankshaft Showing Master/Articulated Rod Construction **(Pratt & Whitney)**

Second-Order Inertial Forces and Linear Vibration

Radial engines are almost always constructed around a crankshaft system using a master rod and articulated rods attached to the master rod via knuckle pins. See Figure 2.1. Other schemes have been tried, but were mechanically complex and fragile. The master rod concept, though imperfect, is good enough. Note that the big end of the master rod moves in a circle on the crankpin, while the small ends of the master and articulated rods, each attached to a piston, move in straight lines.

Engine designers learned long ago to do a good job of balancing the moving parts of a crankshaft system with counterweights. These counterweights are of sufficient mass to balance all of the rotating mass plus one-half of the reciprocating mass. For most engines, this technique results in very good balance. The master rod construction of radial engines poses a special set of problems. See Figure 2.2. None of the knuckle pins move in a circular path, and no single knuckle pin has exactly the same path as any other. In an effort to compensate for piston stroke variation, each knuckle pin is at a slightly different distance from the crankpin center. All these factors conspire to give each piston a unique motion.

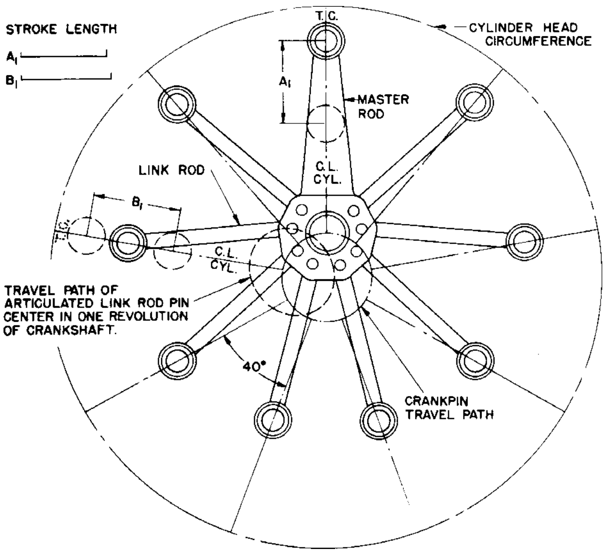

Figure 2.2 Elliptical Knuckle Pin Paths **(Naval Air Training Command)**

As a result, the counterweight of a radial engine can only be made to balance the "average" of all the inertial forces arising from variations in piston and articulated rod motion. When any given piston is at top dead center, the counterweight is too heavy, and when that same piston is at bottom dead center, the counterweight is too light. Note that this unbalanced force occurs twice for each piston for each revolution of the crankshaft. It can be thought of as a force vector that rotates at twice crankshaft speed in the same direction as the crankshaft. This force shakes the entire cylinder row in a whirling motion at twice crankshaft speed, and was referred to by Pratt & Whitney as "second-order linear vibration". Other orders of linear vibration are produced as well, but they are small enough to be insignificant for engines the size of the R-2800.

Second-order inertial forces were never important until the advent of large double-row radial engines with nine cylinders per row. Large single-row radials still have this linear vibration, but the entire engine and propeller whirl together and good engine mount vibration isolators render the vibration unobjectionable.
Double-row radial engines such as the R-2800 have two-throw crankshafts with the throws spaced 180 degrees apart. See Figure 2.3.

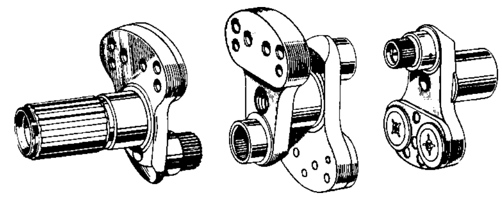

Figure 2.3 Two-row Radial Crankshaft **(Pratt & Whitney)**

One throw and its associated master rod assembly is dedicated to the front bank of cylinders, and the other throw to the rear bank. The 180-degree orientation of the crankshaft causes the unbalanced second-order forces to add in phase, doubling the force acting on the engine. Additionally, since the two crankshaft throws are separated by several inches, the forces form a couple that tends to wobble the entire engine about its center main bearing. This phenomenon was to prove troublesome to both Pratt & Whitney and Curtiss-Wright as they developed the R-2800 and R-3350. Both companies had built a number of successful double-row engines in the past. However, in all cases, prior engines had either smaller cylinders or fewer than nine cylinders per row. The number of cylinders per row is important because as this number increases, the size of the circle of knuckle pins on the master rod becomes larger, exacerbating the effects of their elliptical paths.

Inertia Torques
In addition to the second-order inertial forces discussed above, radial engine master rod construction also gives rise to second-order inertial torques. Unlike torque applied to the crankshaft by the power pulses of individual cylinders, inertia torque results from internal dynamic imbalances and is present any time the engine is rotating. Figure 2.4 depicts a representative radial engine crankshaft arrangement. When the master rod is at top or bottom dead center, all articulated rods are symmetric about the master rod centerline. All forces resulting from the acceleration and deceleration of reciprocating components cancel and no torque is applied to the crankshaft. See Figure 2.5. When any of the other pistons are at top or bottom dead center, the articulated rods are not symmetric. Forces resulting from the acceleration and deceleration of reciprocating components do **not** cancel, resulting in the application of torque to the crankshaft.

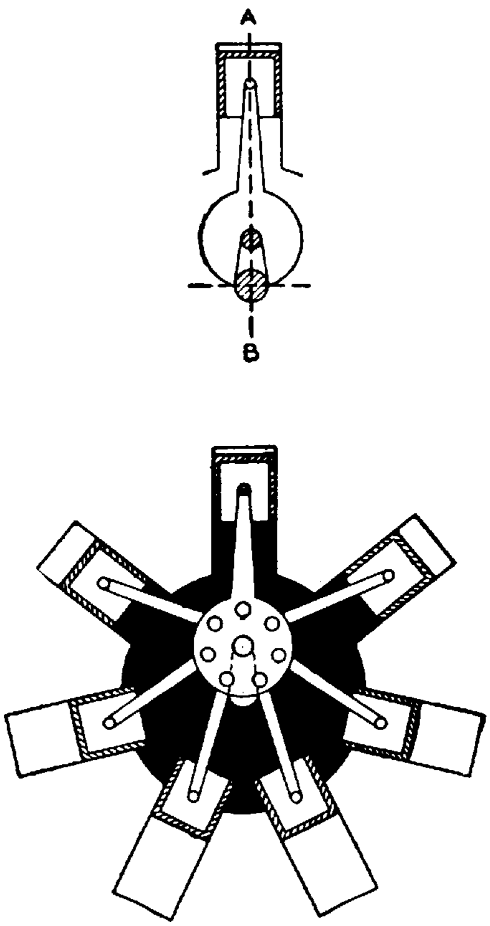

Figure 2.4 Master Rod at Top Dead Center **(adapted from *Aircraft Power Plants*, McGraw-Hill, 1955)**

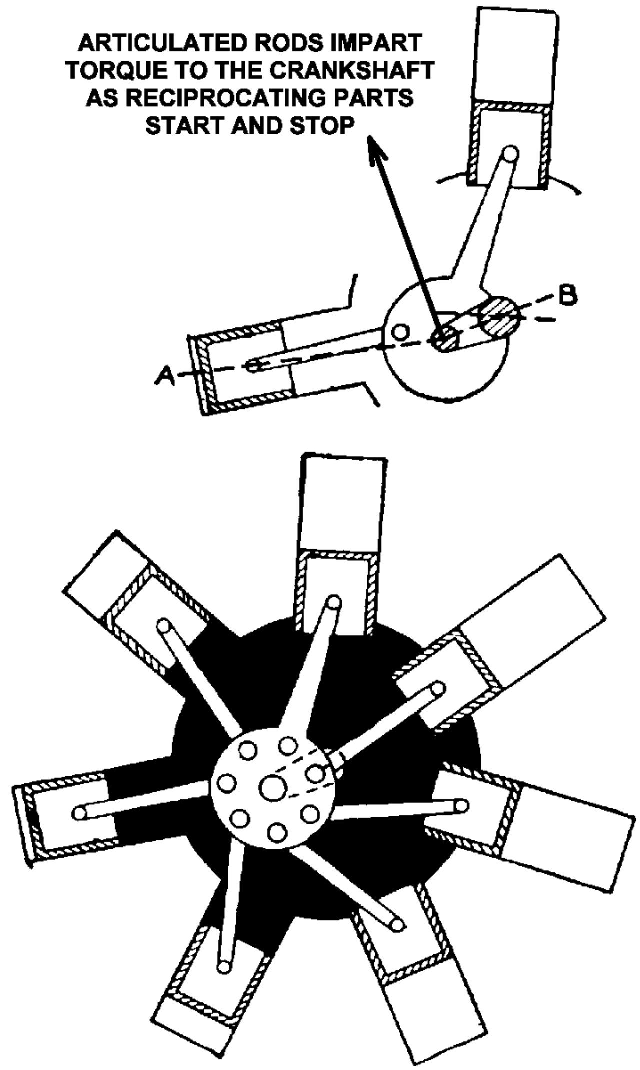

Figure 2.5 Articulated Rod at Top Dead Center **(adapted from *Aircraft Power Plants*, McGraw-Hill, 1955)**

Figure 2.6 shows the total inertia torque variation for one bank of nine cylinders during one revolution of the crankshaft, starting at top dead center of the master rod. This is composed of first and second-order torques. In an engine with nine cylinders per row, third, fourth, and higher orders are small enough to be neglected. This same pattern exists for each bank of cylinders in a multi-row radial engine. By changing the relative position of the master rods, it is possible to vary the overall effect of the inertia torques. If the master rods are placed 180 degrees apart, inertia torques add in phase and produce a torque diagram like Figure 2.6, but with twice the amplitude. If the master rods are placed at 90 degrees, the torque diagram looks like Figure 2.7, which is pure first-order torque. All second-order torque is canceled out.

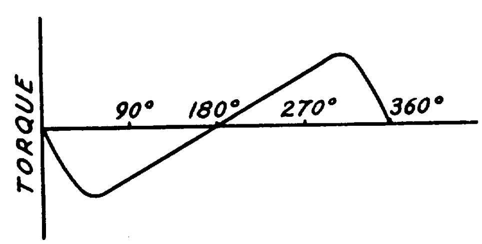

Figure 2.6 Total Inertial Torque Variation **(Pratt & Whitney)**

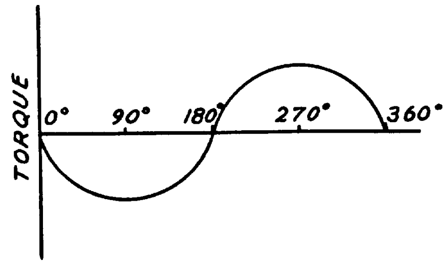

Figure 2.7 First-Order Inertia Torque (Pratt & Whitney)

If the master rods are placed 0 degrees apart, the torque diagram shown in Figure 2.8 results. This pattern is pure second-order torque. All first-order torque is cancelled out.

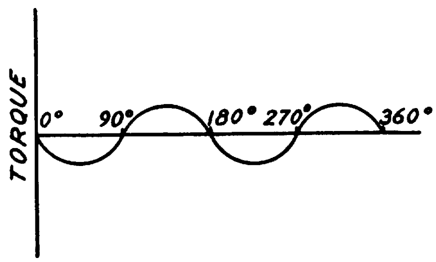

Figure 2.8 Second-Order Inertia Torque (Pratt & Whitney)

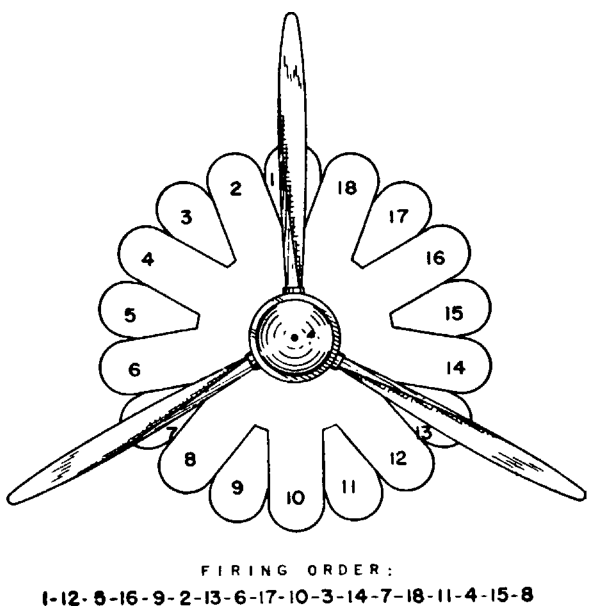

Figure 2.9 Radial Cylinder Indexing

In the real world, cylinders in the front row are staggered to fall between those in the rear row. See Figure 2.9. This improves cooling and makes for more numerous and even firing impulses. It also makes orientation of the master rods at zero or 90 degrees impossible. Instead, they are placed as near to these values as practicable.

Combustion Effects
Another major source of torsional vibration is the force imposed on the crankshaft by the regular, evenly spaced firing of cylinders in a multi-cylinder engine. The R-2800 has one such event every 40 degrees of crankshaft rotation. In such an engine, where each cylinder delivers more than 100 horsepower, this vibration can be quite serious.

Resonance
Reciprocating engines consist of a large number of individual parts, each with its own natural frequency of vibration. These parts are coupled in ways that may cause other parts to vibrate as well, forming a system of vibrations with several natural frequencies. If a regular periodic force is delivered to the system, for example, each time a cylinder fires, and this force happens to be at the natural frequency of some engine part or system of parts, the deflection of the part will be very large. See Figure 2.10.
Since most of the forces inside an engine are delivered to the crankshaft, most of the problems will have to be solved at this point. It is important to bear in mind that all material is flexible. Even a heavy, sturdy steel crankshaft may give several thousandths of an inch or twist several degrees under heavy loads or conditions of resonance.

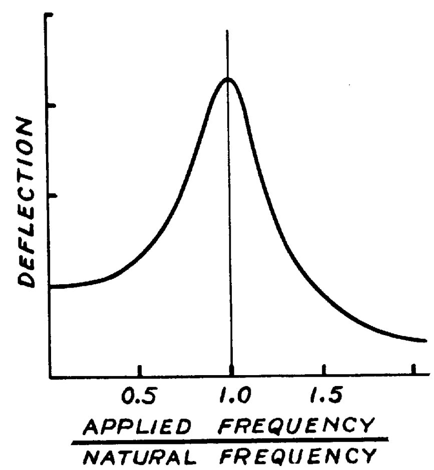

Figure 2.10 Part Behavior in Resonance (Pratt & Whitney)

Engine Anatomy
Figure 2.11 shows the general layout of the R-2800 engine. The parts identified will be discussed in later chapters.

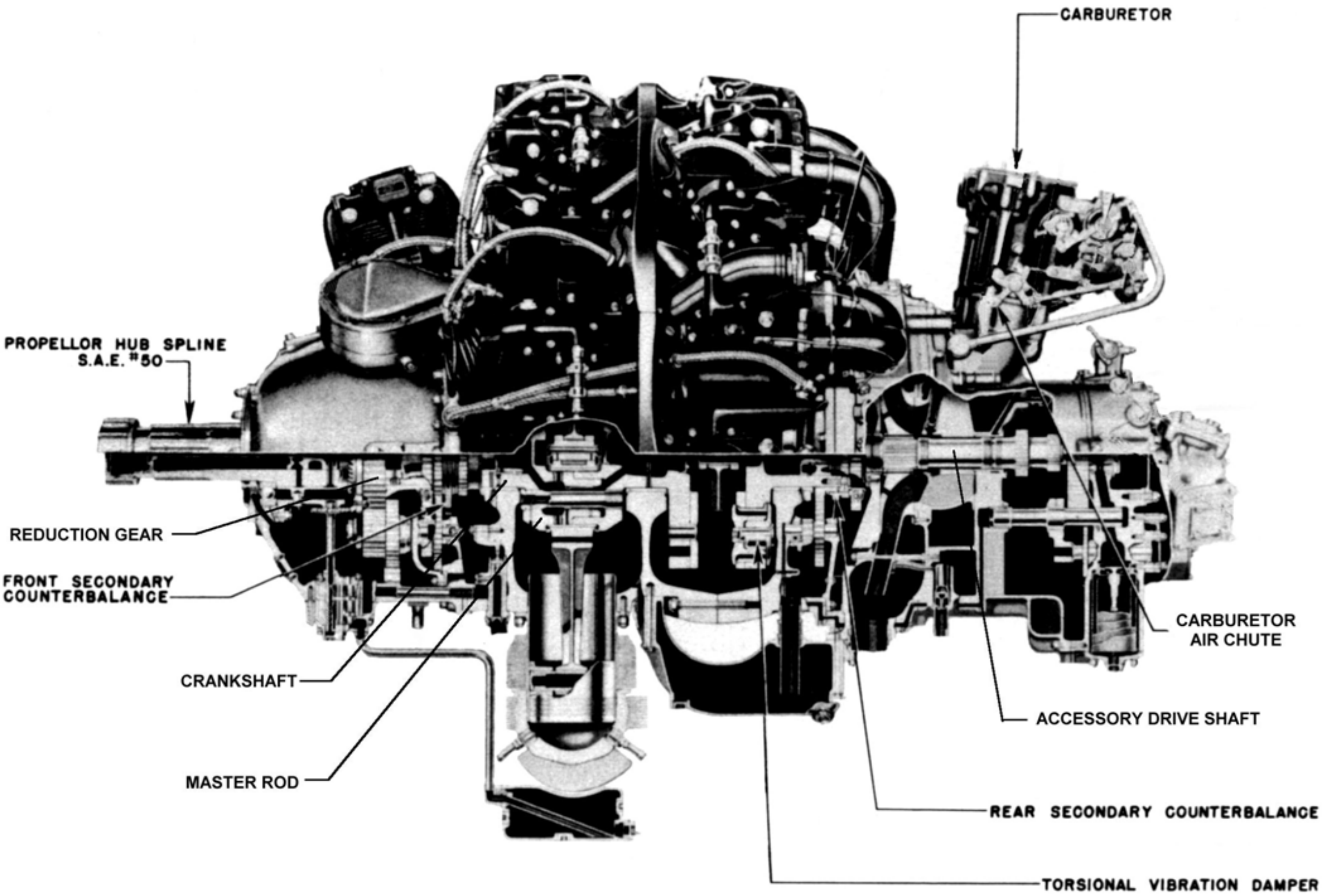

Figure 2.11 R-2800 Left Side View Showing Location of Parts Discussed in Later Chapters (Pratt & Whitney)

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.

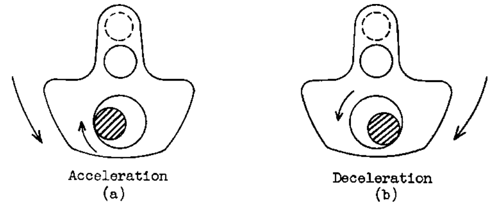

Figure 3.1 Puck-type Damper **(Pratt & Whitney)**

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.

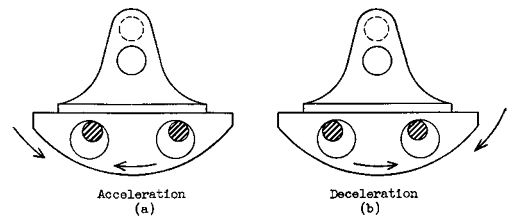

Figure 3.2 Chilton Damper **(Pratt & Whitney)**

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.

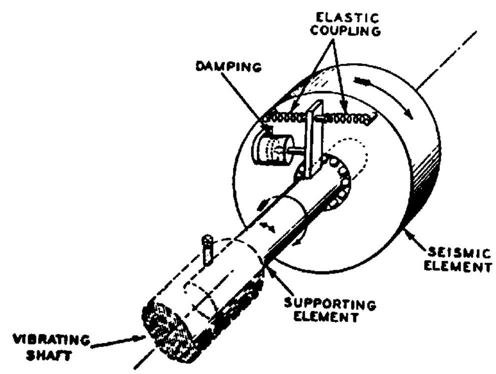

Figure 3.3 Torsiograph Mechanical Components (Draper [6])

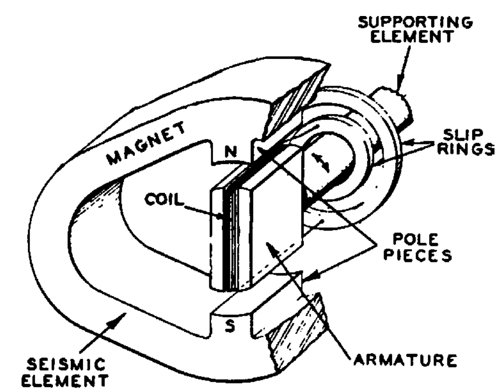

Figure 3.4 Torsiograph Electrical Components (Draper [7])

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.

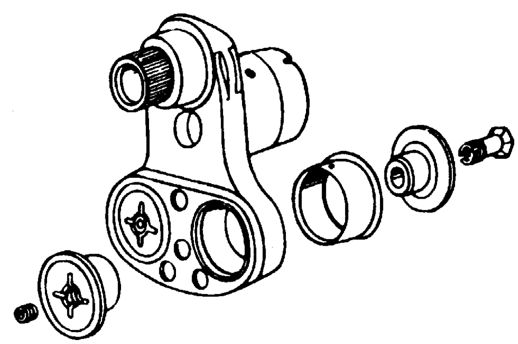

Figure 3.5 Two-spool Damper (Pratt & Whitney)

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.

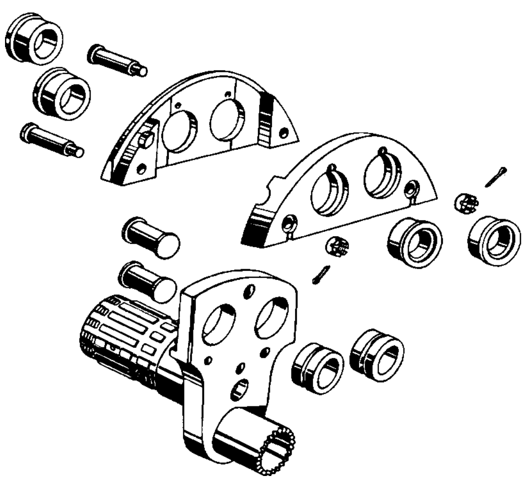

Figure 3.6 "C"-series Damper (Pratt & Whitney)

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.

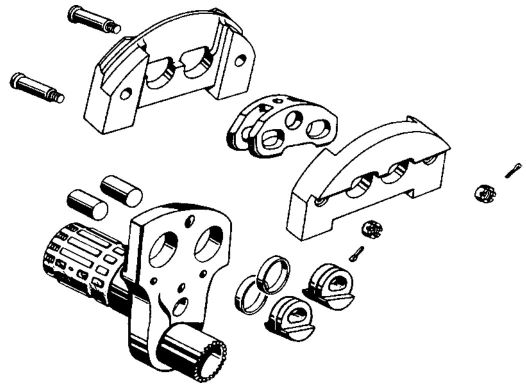

Figure 3.7 "CA"-series Damper (Pratt & Whitney)

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.

4 Linear Vibration

Overload endurance testing is a valuable technique extensively used by engine manufacturers to determine weak points in engines. The process consists of running the engine at high power settings, sometimes even higher than the rated power of the engine, until something breaks. The defective part is then redesigned, the engine rebuilt incorporating the new part, and the endurance run repeated. This procedure, though time consuming and painful, results in robust and reliable engines.

When overload endurance testing was begun on engine X-79, strong vibration began breaking engine parts. W. H. Sprenkle and R. E. Gorton began a series of tests using engine X-78 on April 26, 1938 to investigate the nature of this destructive vibration that had resulted in carburetor mount, air chute, and exhaust stack failures. As was the usual practice, obvious “easy” solutions had already been exhausted: Steel and aluminum air chutes, each with different vibration characteristics, had been tried unsuccessfully. Both metal and wood props were tried, but to no avail. Maximum amplitude of the vibration was at 2600 RPM, right at the take-off power setting for the early “A” engines. Whirling motion at twice engine speed with a node at the center main bearing indicated unbalanced second-order inertia forces. Several suggestions were made to solve the problem, including variation in piston weights between cylinders or the use of three master rods on each crankpin spaced at 120 degrees.[1] While the piston weight variation was tried, no record exists to indicate that the use of three master rods ever received serious consideration.

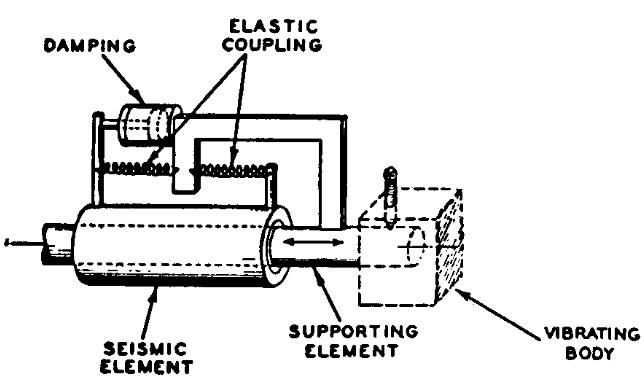


Figure 4.1 Linear Vibration Pickup Mechanical Components (Draper[2])

Figures 4.1 and 4.2 depict the mechanical and electrical components of a linear vibration pickup. The vibrating body is the engine. Two pickups typically measure motion along vertical and lateral engine axes. The electrical output is fed to a multi-channel recording oscillograph, which simultaneously records on a 35mm filmstrip the instantaneous position of the pickups as a function of time. Later analysis allows correlation of relative phase and frequency of the pickups.

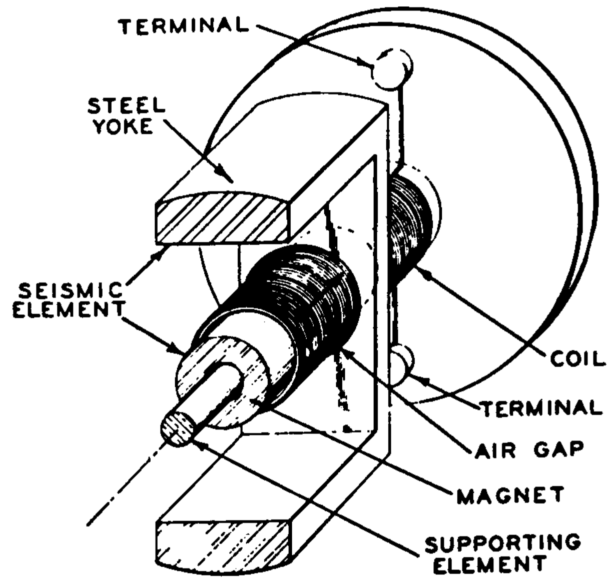


Figure 4.2 Linear Vibration Pickup Electrical Components (Draper[3])

On May 11, 1938, a test was conducted to investigate one of the proposed fixes. Heavier pistons with solid pins and bronze end plugs were installed in the each master cylinder and three adjacent cylinders on each side. Primary balance was maintained by increasing the counterweight mass. In the final analysis, this approach was not practical for achieving secondary inertia balance. Original calculations predicted an 80% improvement. Only a 10% improvement was realized, and this with a 75-lb weight penalty. In spite of its inherent mechanical complexity, secondary counterbalances[4] seemed to be the only remaining solution.[5] These consisted of counterbalance weights mounted concentric with the front and rear crankshaft main journals. These counterbalances were driven by a gear train at twice crankshaft speed, and were phased to properly counteract the inertia forces. See Figures 4.3 and 4.4.

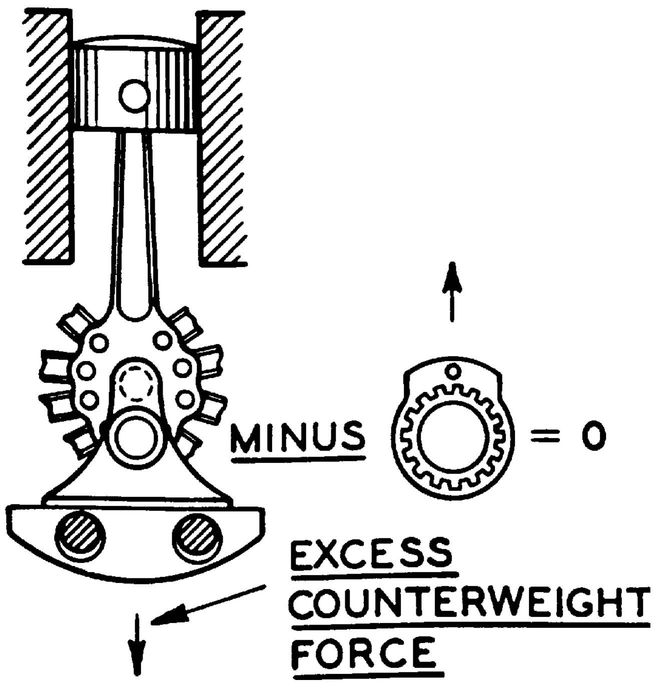


Figure 4.3 Counterbalance Action at TDC.
(Pratt & Whitney)

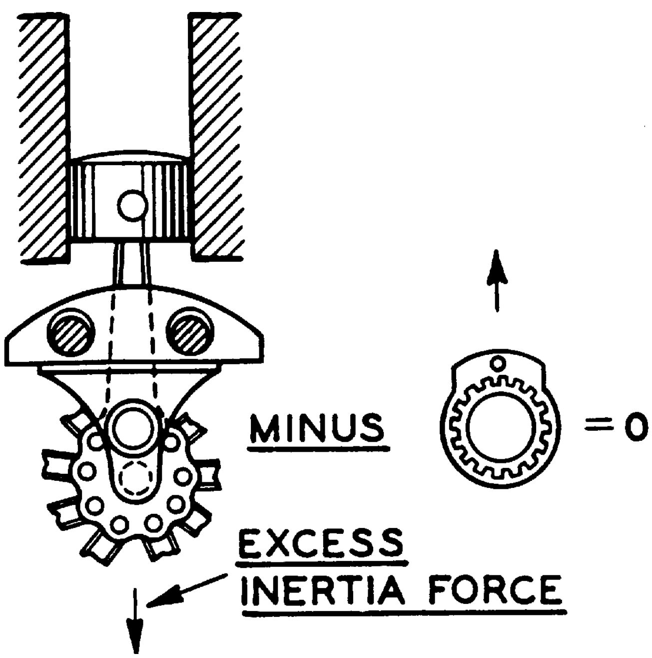


Figure 4.4 Counterbalance Action at BDC
(Pratt & Whitney)

It is no small wonder this solution was a last resort. Designing and producing such a mechanism was a difficult undertaking, and would present an ongoing series of challenges.

By June 17, 1939, secondary counterbalances had been designed and fitted to the experimental engine. Since the R-2800 was rich in torsional as well as linear vibration, the front secondary counterbalance drive gears stripped their teeth before testing could be completed. The results, however, looked promising, having produced a six-fold decrease in vibration during the short test period before the counterbalance drive broke. While it seemed probable that these secondary counterbalances would eventually solve the linear vibration problem, it was also obvious that a long and painful development cycle lay ahead.

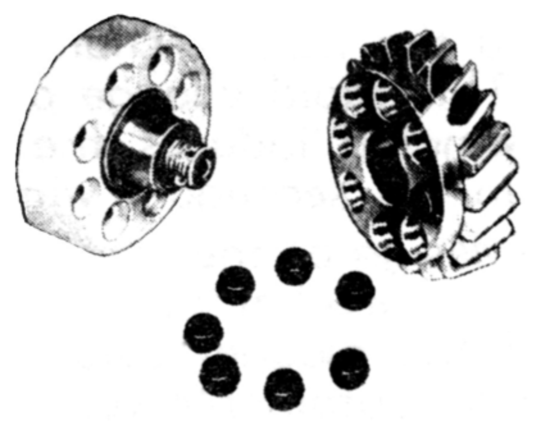
This test also revealed a 3.5X linear vibration for the first time. This 3.5X vibration would prove elusive and troublesome, coming and going from test to test, distracting the team from the more urgent 2X problem. Sprenkle, experienced with similar vibration difficulties in the R-1830-C engine, assigned this problem to valve inertia[6]. This indeed turned out to be the case. The 3.5X linear vibration problem was cured by redesigning the cam profile to provide for more gradual opening and closing of the valves as well as stiffening the valve push rods. Nevertheless, it would be early December of 1938 before these valve gear problems and their associated maverick 3.5X linear vibration was laid to rest.[7]

Experience with the R-1830 also suggested that a 180-degree master rod placement might improve the 2X vibration. This concept was explored during the first week of July, but produced no change in 2X linear vibration. Instead, the change worsened second-order torsional vibration.[8] During this test, the role of the propeller or test club was also investigated to determine whether interference between the propeller blades and engine parts contributed to vibration. It was decided that propeller contribution to 2X linear vibration was insignificant.[9]

By the end of July, experiments with the S.A.E # 50 propeller shaft and metal props indicated a worsening linear vibration picture. The S.A.E # 50 shaft, installed as a weight-saving feature, had reduced the resonant speed to about 2550 RPM, which is below takeoff RPM.[10]

To further investigate secondary counterbalances as a possible solution to the vibration problems, a new counterbalance drive was designed using neoprene rubber buttons in the drive couplings.

It was hoped that these rubber buttons would dampen the torsional vibration that had so rapidly destroyed the earlier counterbalance drives. First, a drive with six buttons was tried and failed due to shearing of the buttons. Next, a drive coupling using fifteen buttons was tested, but it also failed. In both cases, the counterbalance bearings showed galling. This testing was completed August 16, 1938.



Rubber Button Magneto Drive Representative of Those Used to Isolate Crankshaft Torsional Vibration from Secondary Counterbalances

In spite of these problems with the drive couplings and counterbalance bearings, the concept of the secondary counterbalance continued to show promise, producing over seventy-five percent reduction in vibration with the metal flight propeller.

It was suggested that a new counterbalance drive coupling using leaf springs be developed, and that lead plating of the counterbalance bearing would eliminate the bearing distress.[11]

By August 31, 1938, counterbalance drives incorporating leaf springs to isolate the counterbalance system from crankshaft torsional vibration were ready to be tested. Reduction in vibration using these drives was about the same as that using rubber buttons, and the durability of the drive system was improved[12]. Some of these tests had shown that the actual 2X linear vibration reduction was not as good as theoretically predicted. Several explanations were put forward for this, including the idea that the 4-blade test club in combination with the 2:1 reduction was causing a prop interference and producing additional 2X excitation. [13]

While the leaf-spring drive secondary counterbalances were more durable than the previous ones using rubber drive couplings, they were still not as reliable as they needed to be. In a test on October 28, 1938, the 2X vibration had returned. Upon teardown, it was discovered that the leaf springs in the counterbalance drives had broken, rendering the drives inoperative.[14]

Extensive testing was done between November 23 and December 5, 1938 to compare the vibration-attenuating characteristics of light, medium, and heavy secondary counterbalances with both wooden test clubs and metal flight propellers. Earlier testing had been done with counterbalances having an unbalance mass-radius product of 2.0 lb/in, theoretically producing a 68 percent reduction in unbalanced secondary forces. This test series explored the behavior of counterbalances having 2.41 (84 percent reduction) and 2.82 lb/in (100 percent reduction) of unbalance.

On runs with the wooden test club, all three counterbalance designs produced 2X linear vibration measurements that were similar. Gorton points out that since there was no control over the relative position of the crankshaft and propeller shaft during assembly of the reduction gear, it was possible to assemble the engine so that a 4-blade prop running with the 2:1 reduction ratio always wound up in the same spot, with the propeller interference producing 2X excitation. By indexing the test club about all possible positions, it was possible to prove that 2X prop interference was indistinguishable from 2X linear vibration, and could contribute as much as 50 percent of the total vibration. Future tests considered this and avoided assembly combinations that lead to prop interference,

On runs with the 3-blade metal flight propeller, the 2.82 lb/in counterbalances were successful in reducing 2X linear vibration to levels below that of the 1.5X propeller interference vibration.[15] In early November of 1939, when testing first began in the new horizontal-intake test house, this 1.5X vibration was reduced 40-60 per cent. Gorton recommended that all future tests using flight propellers be conducted in the new test house to allow accurate measurement of other vibration components.[16]

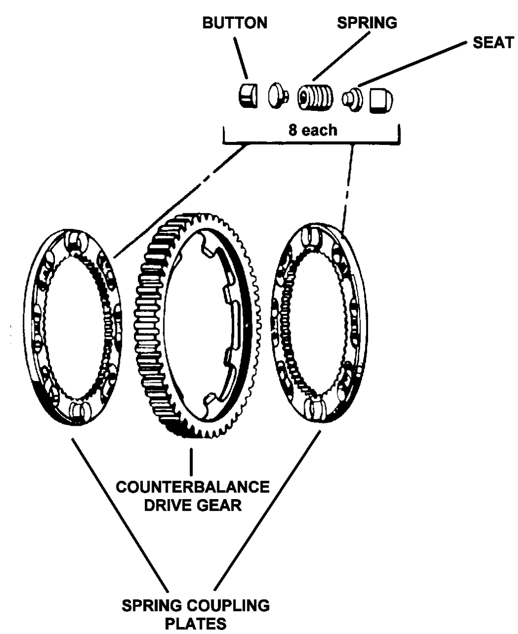


Figure 4.5 Counterbalance Spring Drive (Pratt & Whitney)

The leaf spring secondary counterbalance drive design was discarded in favor of a design using coil springs and leveling buttons in the drive. See Figure 4.5. This was first incorporated in engine X-79 on October 10, 1938. This design continued to be effective in 2X linear vibration reduction and was more durable than previous designs. However, the engine could not be operated for extended periods due to interference between the countershaft and its bushing. Once the shafts and bushings were modified with more generous radii, the interference problems went away.[17]With slight modifications, this counterbalance drive design was used in the Army No. 1 Type Test engine.

While this secondary counterbalance design was successful in eliminating objectionable 2X linear vibration, numerous changes were made to the counterbalances and drives as a result of R-2800 service experience, power increases, and engine design evolution. No fewer than six counterbalance revisions had been made to "A" and "B" series engines by July of 1943.

The original Type I counterbalance (Figure 4.6), used in early "A" and "B" engines, had a light bob-weight and small reinforcing ribs around the outside rim of the counterbalance.[18] Two minor changes were made to the Type I design during its service life. Copper plating was added to the inside diameter of the counterbalance bearing, and silver plating of the spring drive plates replaced the lead flashing that had been originally been used.[19]

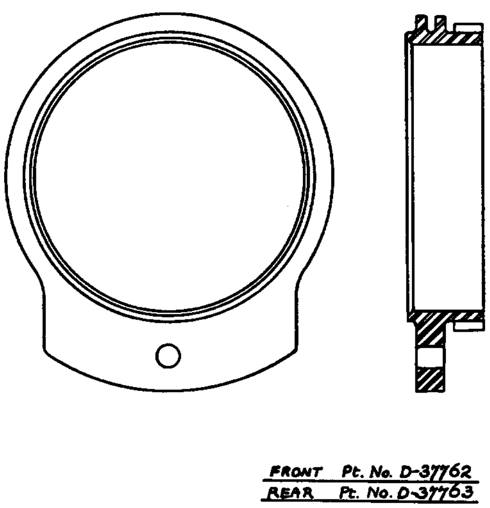


Figure 4.6 Type I Counterbalance `(Pratt & Whitney)`

Bearing failures caused by deflection of Type I reinforcing rim resulted in a change to Type II (Figure 4.7). Here the reinforcing rim was enlarged and the bob-weight was made heavier.

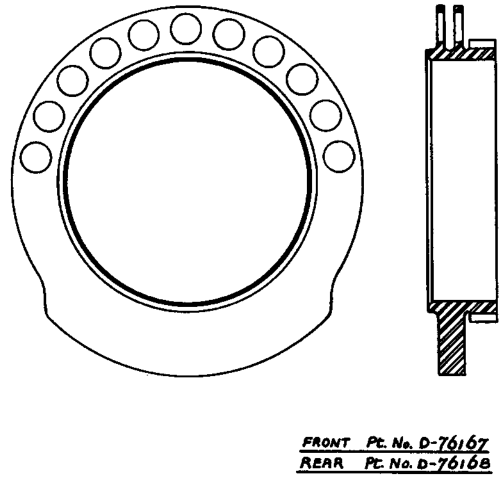


Figure 4.7 Type II Counterbalance `(Pratt & Whitney)`

Nevertheless, it was unsuccessful in service, and a campaign was necessary to alleviate the trouble by replacing the Type II with Type III. (Figure 4.8). A Service Bulletin was issued which detailed the process of reworking Type II counterbalances by removing material from the bob-weight and improving lubrication to the counterbalance bearing.[20]

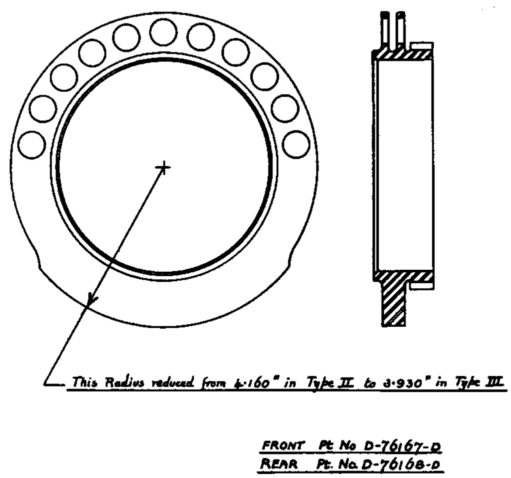


Figure 4.8 Type III Counterbalance `(Pratt & Whitney)`

When a rash of "B" series engine failures resulting from seized counterbalance bearings grounded the entire European P-47 fleet, a crash program was instituted to find the source of trouble. The problem was traced to engines built at the Ford Rouge River Plant. Failure to properly clean the crankcase castings was allowing core sand[21] from the manufacturing process to contaminate the lubricating oil, causing bearing distress.[22]The fix consisted of a hat-shaped hood over the lubricating oil jets and extension of the standpipe inside the propeller shaft that delivers oil to crankshaft. By forcing oil to follow a path against the centrifugal force gradient inside the shaft, particles of sand were prevented from reaching the bearing. This fix was referred to as Type III(A)[23]. See Figure 4.9.

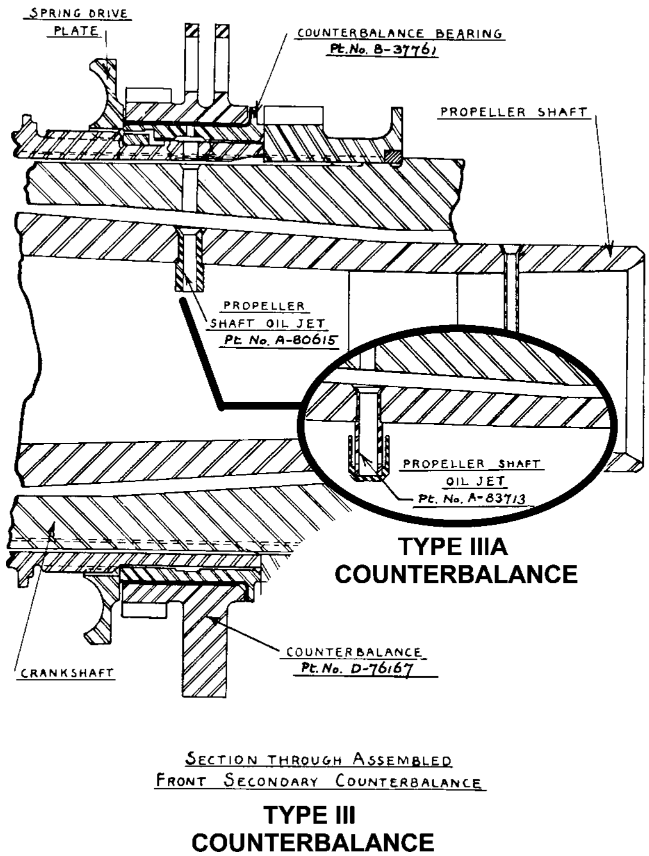


Figure 4.9 Type IIIA Counterbalance (Adapted from Pratt & Whitney)

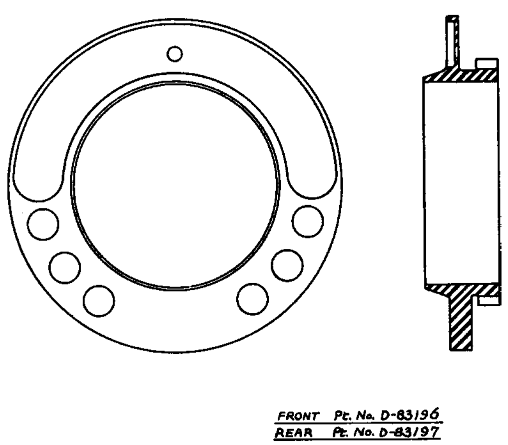


Figure 4.10 Type IV Counterbalance (Pratt & Whitney)

The Type IV secondary counterbalance, shown in Figure 4.10, was a completely new design and featured a wider bearing which was splined to the crankshaft in front and an integral part of the rear crankshaft gear in the rear. The counterbalances themselves were wider with a single strap all around.[24]

A Type V secondary counterbalance included all the design changes of the Type IV, but was intended to make maximum use of existing "B" engine parts and be installed in the field.[25] It is not clear if this change was ever actually implemented.

When the R-2800 "C" series engines were introduced, master rod location was changed to cylinders 8 and 9 (20 degrees apart). This was done to reduce troublesome first order (1X) torsional and linear vibration that had plagued both the "A" and "B" series of engines. The "C" produced a maximum of 2100 HP at 2800 RPM (2400 HP was planned), so the 1X vibrations had to be fixed. Relocation of the master rods solved the 1X problems but worsened both 2X torsional and 2X linear vibration. The 2X torsional vibration was solved by installing 2X bifilar dampers on the front crankshaft counterweight. The 2X linear vibration was compensated for by installing secondary counterbalances with even higher unbalance mass radius products.[26] At least one improvement was made to the "C" secondary counterbalances because of service experience. Figure 4.11 shows a simplified assembly drawing of the final secondary counterbalance design. Both the secondary counterbalance bearing and reduction gear drive coupling are splined to the crankshaft. Power is transmitted from the crankshaft to the reduction gear drive coupling, via the rear spline to the spring coupling plates. The counterbalance drive gear is driven via the eight spring packs. This, in turn, drives the intermediate drive gear which drives the counterbalance at twice crankshaft speed.

After WWII, Pratt and Whitney was anxious to bring the R-2800 to the commercial sector, but wanted a major leap forward in power, smoothness, reliability, and longevity. Thus, the R-2800 "CA" series was born. The "CA" initially used the same secondary counterbalances as the "C" series, but improvements resulting from service experience produced two additional secondary counterbalance designs that were used in the "CA", "CB", and "CE" engines.

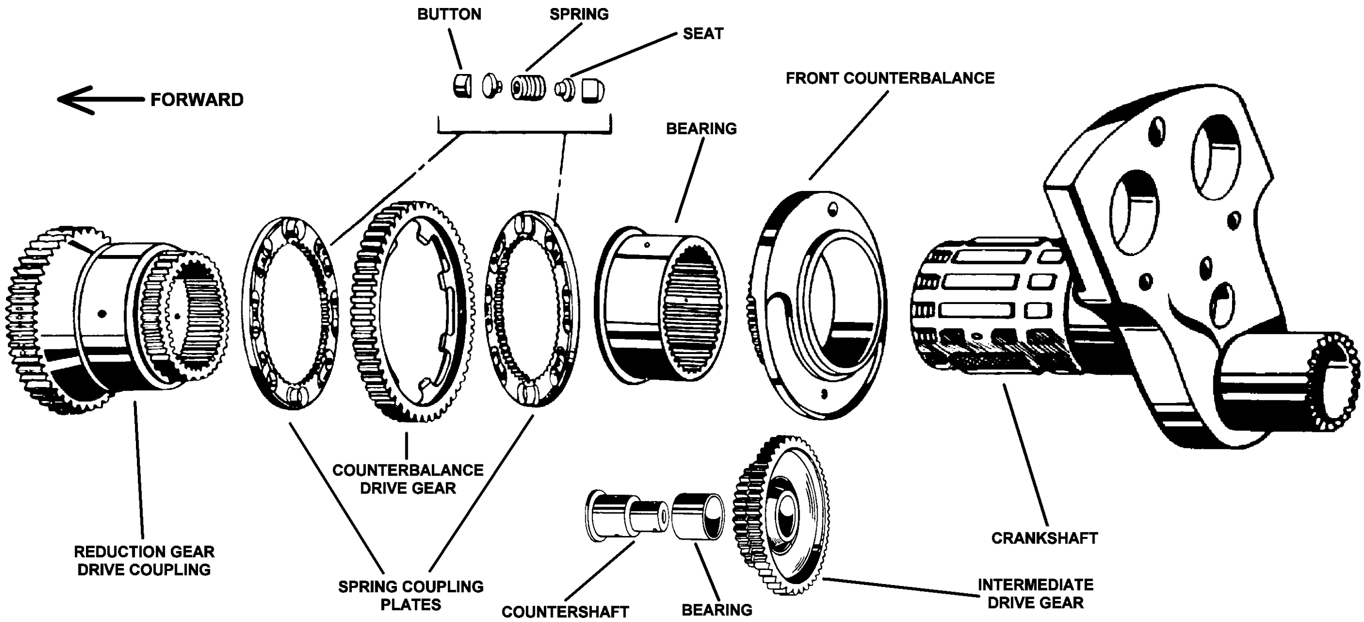


Figure 4.11 Simplified View of Front Secondary Counterbalance and Drive (Pratt & Whitney)

[1] W.H. Sprenkle and R. E. Gorton, "Torque Stand Vibration of R-2800 Engine X-79", *Pratt & Whitney Aircraft Experimental Test Department Short Memorandum Report (SMR) No. 420* (April 29, 1938).
[2] 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.
[3] Ibid.
[4] Pratt & Whitney test reports use the term "Second Order Counterweights". Later Pratt & Whitney overhaul manuals use "Secondary Counterweights" while parts catalogs use "Secondary Counterbalances". The author uses "second-order" when referring to vibration and forces, "counterbalance" when referring to mechanical components that mitigate second-order vibration, and "counterweight" when referring to components that achieve crankshaft primary balance.
[5] W. H. Sprenkle, "Torque Stand Vibration of R-2800 Engine X-78 with Weighted Pistons", *SMR No. 432* (May 18, 1938).
[6] W.H. Sprenkle, and R. E. Gorton, "Vibration Test on R-2800 Engine X-78 with Second Order Counterweights" ,*SMR No. 442* (July 1, 1938).
[7] George E. Meloy, "Report on History of R-2800 Engine Development", (PWA Report No. PWA-192, May 30, 1939), 9.
[8] SMR No. 449 explores torsional vibration issues associated with this approach.
[9] W.H. Sprenkle and R. E. Gorton, "Linear Vibration of R-2800 Engine X-78 with 180° Master Rods", *SMR No. 450 (*July 15, 1938).
[10] W.H. Sprenkle and R. E. Gorton, "Linear Vibration of R-2800 Engine X-78 with 50 Spline Propeller Shaft", *SMR No. 454* (August 5, 1938).
[11] 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).
[12] W.H. Sprenkle and R. E. Gorton, "Linear and Torsional Vibration Tests on R-2800 Engine X-78 with Spring Drive Second Order Counterweights and Wooden Test Club", *SMR No. 475* (September 28, 1938).
[13] 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).
[14] 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).
[15] R. E. Gorton, "Comparison of Linear Vibration Characteristics of the R-2800 Engine with Light, Medium, and Heavy Second Order Counterweights, both with Wood Test Club and Hydromatic 6159-0 Propeller", *SMR No. 515* (February 1, 1939).
[16] R. E. Gorton and A. R. Crocker, "Linear and Torsional Vibration or R-2800 Engine X-83 with Loose Crankshaft Counterweight Plugs Operating in Horizontal Intake 18' Test House", *SMR No. 619* (November 22, 1939).
[17] George E. Meloy, "Report on History of R-2800 Engine Development", (PWA Report No. PWA-192, May 30, 1939), 7.
[18] E. M Speer, "Resume of Secondary Counterbalance Equipment O. H. 2800 Engines", *Internal P&W Memorandum to T. Gurney* (July 16, 1943), 1.
[19] Technical Information Letter No. T-5, (Pratt & Whitney Aircraft, March 9, 1944), 1.
[20] Speer, 1.
[21] Core sand is a material used to fabricate the molds in which complex engine parts are cast.
[22] Larry Carlson, meeting of P&W R-2800 developers, November 5, 1998.
[23] Speer, 1.
[24] "Technical Information Letter No. T-5", 2 – 3.
[25] Speer, 2.
[26] B. E. Miller, "Second Order Linear Vibration Characteristics of R-2800 Engines", *SMR No. 879* (January 18, 1943).

5 Crankshaft Development

One of the things that made the original Pratt & Whitney "Wasp" so successful in 1926 when it first passed its type test was the ability to make its power at a higher RPM and a lighter weight than its competition. Key to this accomplishment was the use of a one-piece master rod and two-piece crankshaft. Though two-piece crankshafts had been built before, George Mead and Andy Willgoos chose a new construction consisting of a split crankpin splined to its mating crankpin, the whole assembly being held together with a bolt through the center of the crankpin. See Figure 5.1.

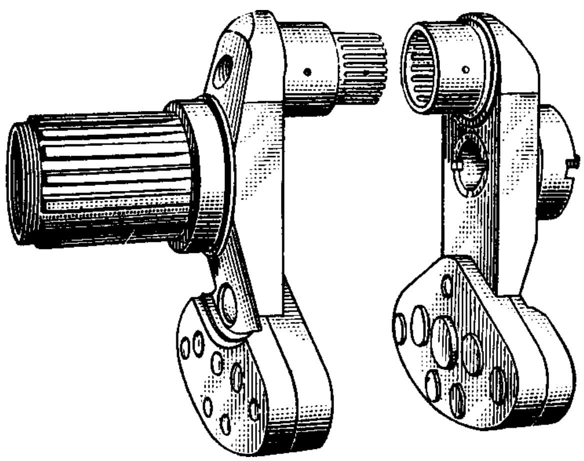


Figure 5.1 "Wasp" Crankshaft (Pratt & Whitney)

This construction was used in many, but not all, Pratt & Whitney designs preceding the R-2800. It is therefore no surprise that the designers chose this same type of construction for two-throw R-2800 crankshaft. The original R-2800 crankshaft compensated for the weight of the master rod and link rods in the usual fashion, by providing a counterweight that balanced all of the rotating mass and one-half of the reciprocating mass. Initially, no vibration dampers of any kind were provided. It is unclear whether this was wistful thinking on the part of the designers, or merely acknowledgement that no one could predict the vibration behavior anyway, so they may as well start testing to uncover the problems as early as possible. One thing the designers did consider was placement of the master rods as close at possible to 90 degrees to one another so that second-order inertia torques could cancel as nearly as possible, reducing 2X torsional excitation of the crankshaft.

George E. Meloy was heavily involved in R-2800 crankshaft development almost from the start. One of his first jobs at Pratt & Whitney was to write a report on the history of R-2800 development, which included many details on the successes and failures of the crankshaft. Meloy was later responsible for sorting out problems with the "C" engine crankshaft and getting it into successful production in the Kansas City, Missouri plant. Some of the people who worked for Meloy remember him for being the only person they know who could walk into a test cell and not get oil on his clean white shirt.

Meloy was born in Chicago in 1916, but at the age of four moved east to New York. He eventually settled in Teaneck, New Jersey where he graduated from Teaneck High School. Meloy received a Bachelor of Aeronautical Engineering from New York University. Despite the scarcity of jobs brought about by the Depression, Meloy started work at Pratt & Whitney one week after graduation in 1938. Initially a test engineer, Meloy advanced rapidly through project engineering and finally into management. While his real love was in development, like many capable technical people, he had the management role forced upon him. However, he did not despair. Says Meloy, "Every moment spent at Pratt, to me, was worth while. I didn't watch the clock, didn't have to. During the war years, we worked 54-hour weeks. There were no perks back in that time, understandably. We were just happy to do it. It gave us a feeling we were doing something worthwhile for the defense of the nation."

Connecting Rod Evolution

The first one-piece master rod assembly featured a locked silver-plated bearing and locked knuckle pins. A silver-plated flange on the forward face of the master rod bearing carried thrust loads on the master rod. This design was discarded because of weaknesses that became apparent during testing. By strengthening portions of the master rod and link rods that were highly stressed, as well as increasing the fillets and radii at stress concentration points, master and link rod structural failures were eliminated. Aiding this process was moving knuckle pin oil delivery passages to the knuckle pin retaining plates.

Much of the master rod development was done using brittle lacquers. These coatings were the only instrumentation available at that time for internal engine parts. Brittle lacquers have the characteristic of cracking when the material to which they have been applied flexes. By analyzing the concentration and orientation of cracks in the lacquers, highly stressed engine components could be improved by adding metal in the right places

Master rod bearing failures prompted a series of experiments into bearing construction and materials. The original copper-bronze and bronze bearings were replaced with silver-lead bearings in April of 1938, eliminating the material problems. The question of how to retain the bearings got more attention. These were originally a press-fit. Use of set screws to lock the bearings was tried but not successful.

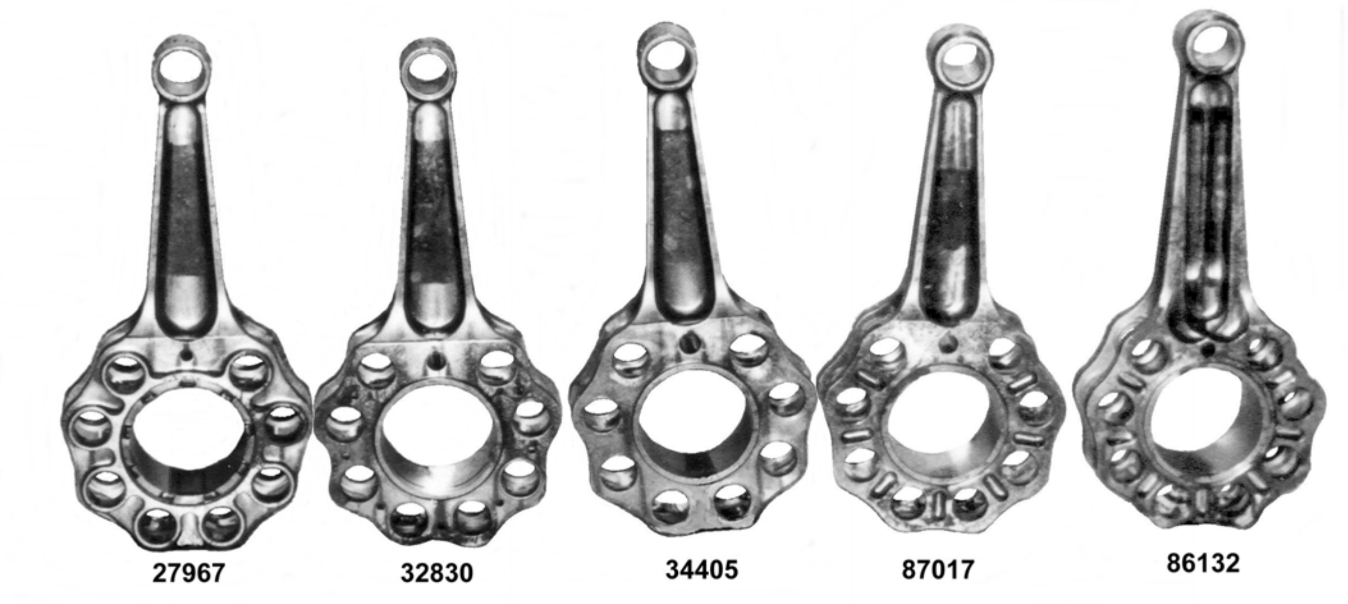


Figure 5.2 Master Rod Evolution **(Author)**

Neither was a floating bearing with silver-lead both inside and outside and a floating bronze thrust collar. Another floating bearing design with large aluminum plates fastened to the sides of the master rod was rejected because of metal transfer on the mating faces. Finally, a successful locked-bearing design with floating knuckle pins was tested in October of 1938. In order to reduce oil flow to the power section, master rod bearing clearances were reduced by 0.004”.[1] As engine power and maximum RPM continued to increase, connecting rod design evolved to meet the new challenge.

Figure 5.2 shows the evolution of R-2800 master rods. The two left-most rods, P/N 27967 and P/N 32830 are early experimental designs that never saw production. The center rod, P/N 34405 was used in the “A” and “B” series of engines. The fourth one, P/N 87017, was used in the “C” series of engines. The one on the right, P/N 86132, was used in early “E”, “CA”, “CB”, and “CE” series engines. Compare the sharp edges and tight radii on the early rods with the generous fillets and large radii of the later ones. Note the progressively larger cross section of the rods, and the center rib in the web of the later design. Extremely high quality of fit and finish is evident in all the examples.

Crankshaft Evolution

Early experience with the initial crankshaft design was problematical. Almost immediately, spiral fractures on the front crankpin began causing crankshaft failures. This was first blamed on master rod bearing seizures, but crankshaft failures continued to occur even after the bearing problems were solved. On August 8, 1938, a failure on engine X-79 after just 41 hours of operation forced design changes. These included revisions in the oil distribution and changes to the rear crankshaft gear locking provisions. It was during this same period that torsional vibration testing had indicated the need for 4.5X torsional vibration dampers which were then included in the rear counterweight.

Continuing problems with the spline that joined the R-2800 crankpins had resulted in several redesigns. This included moving the joint to the crankpin center from its previous off-center position, replacement of the machined spline with a splined plug, and hardening of the mating surfaces. In all cases, the changes failed to eliminate galling of the crankpin mating surfaces and spline faces.

These efforts were further hampered by occasional crankshaft failures resulting from the fact they were hand-forged. Whereas later production crankshafts would be die forged, the crankshaft design was not yet finalized, and the price of forging dies prohibited their use for experimental crankshafts. Problems with hand forging due to inclusions and poor grain structure were well documented, and led to many crankshaft failures.[2]

Dana Waring, one of the test engineers who made a career at Pratt & Whitney, remembers a spectacular crankshaft failure. Waring was observing an engine running at full power in the test cell. It was outfitted with a metal flight propeller that, in conjunction with the short exhaust stacks, was making a huge amount of noise. In the blink of an eye, and with a loud bang, the engine rotated 180 degrees in its test stand fixture, tore loose from its mounts and came to rest on the test cell floor, leaking oil and smoking. In the mean time, the propeller had sheared off and flown forward to the front of the test cell, knocking a dent in the concrete wall. The propeller hovered there for a few revolutions until it lost some momentum, and then slid to the floor, still rotating. When the propeller blades began hitting the floor, the entire propeller began walking around the forward end of the test cell until it used up its remaining momentum and came to rest. Dana Waring was thereafter very reluctant to enter the test cell while an engine was running.

Despite difficulties with crankshaft development, it was this crankshaft design that was used in the R-2800 "A" and "B" series engines that saw the majority of the action and contributed so much to the winning of World War II. See Figure 5.3.

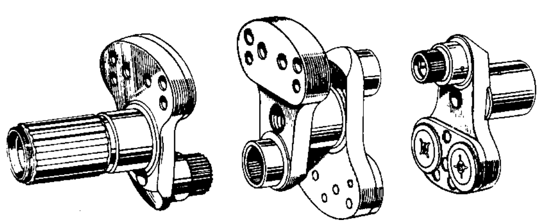


Figure 5.3 "A/B" series Crankshaft **(Pratt & Whitney)**

The higher horsepower and redline RPM of the "C" engine required major changes in crankshaft design. The engineers followed two different threads of crankshaft development. The first continued to refine the splined crankpin connection while the second pursued a clamp-type crankshaft.

In late February and early March of 1939, a new crankshaft design with two counterweights instead of four was tested. This design offered a considerable weight savings of over 32 pounds, and also facilitated elimination of the two-piece crankcase center section that had been used on the "A" and "B" models[3]. The initial two-counterweight crankshaft was made from an old four-counterweight crankshaft, and did not have 4.5X torsional vibration dampers.[4] This crankshaft, an old design that was hand-forged, failed through the rear crankpin after it had accumulated a total time of 453.2 hours, and 151 hours after rework to the two-counterweight configuration. Metallurgical examination revealed poor grain flow and structure and recommended strategies to prevent such failures in the future.[5]

In addition to problems with material properties, failure of the two-counterweight splined crankshafts, was attributed to the bending vibration in the crankshaft. This led to a design in which the effective mass of the rear counterweight was reduced in the fore-aft direction by installation of two cylindrical plugs in the counterweight that were free to slide fore-aft along their axes. Torsional and linear vibration were not measurably different from the earlier two-counterweight spline-joined crankshafts without the loose plugs.[6]

Frequencies of resonance in bending were measured using some clever instrumentation produced by Gorton and Crocker. This consisted of a horizontal linear vibration pickup mounted on the crankshaft axis. An adapter tube screwed to the rear crankshaft journal extended through the accessory drive shaft to the exterior of the engine. Rotation between the adapter shaft and vibration pickup was via a preloaded double-row ball bearing. A second horizontal vibration pickup mounted on the vacuum pump adapter pad external to the engine sensed overall engine vibration. Comparison of signals from the two pickups allowed measurement of fore-aft motion of the crankshaft. This motion could then be related to the bending vibration of the crankshaft. These bending vibration tests indicated that the loose plugs in the rear counterweight were effective in eliminating 4.5X bending vibration that was believed to have contributed to the breakage of the earlier two-counterweight crankshaft design.[7]

Clamp-type Crankshaft

One solution to the weakness of the splined crankshaft was a clamp-type crankshaft. This took the form of a two-counterweight crankshaft without 4.5X torsional vibration dampers that received considerable attention and testing from May through October of 1939. This crankshaft design had slightly better 4.5X propeller blade tip stress characteristics than the four-counterweight crankshaft, but otherwise had identical vibration characteristics with the two-counterweight splined-crankpin crankshaft.[8] But it was also harder to assemble, requiring special alignment fixtures and assembly techniques, and prone to slippage. Considerable experimentation went into finding the correct amount of clamp bolt stretch. Each experiment involved engine teardown, inspection, and reassembly. The frequent tightening of the clamp bolt caused galling of the clamp surfaces and necessitated re-drilling of the cotter pin hole in the clamp bolt with each assembly.[9]

Refinement of the clamp-type crankshaft continued. Dynamic counterweights were added, along with other improvements. Planners intended this type of crankshaft for the production "C" engine to be built in Kansas City, Missouri. Much of the experimental development of the "C" engine, which began on September 1, 1940, was done with the clamp-type crankshaft.[10] But this crankshaft design never saw production. See Figure 5.4.

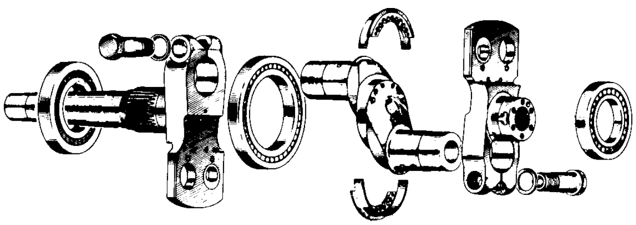


Figure 5.4 Clamp-type Crankshaft Representative of Those Tested By Pratt & Whitney (Navy)

Face-splined Crankshaft

Instead, a face-splined crankshaft construction was developed and used in the "C" and all subsequent R-2800 engines. See Figure 5.5.

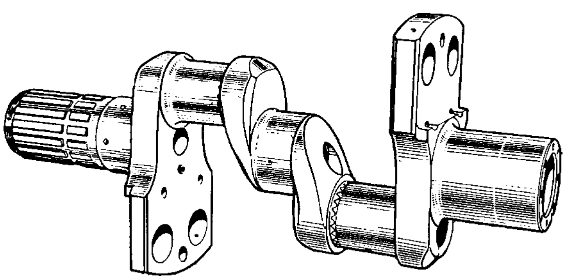


Figure 5.5 "C" series Crankshaft **(Pratt & Whitney)**

It is the opinion of the author, and this opinion is shared by retired Pratt & Whitney engineers Elton Sceggel[11] and Gordon Beckwith[12], that improvements in gear-cutting technology at the Gleason Works of Rochester, N.Y. made possible the machining of complex involute splines necessary for this new joint. See Figure 5.6.

The face-splined crankshaft is first mentioned in a report on the bending behavior of various crankshaft joints. In this report, six joint designs were tested: the traditional internal spline; the clamp-type; the face splined with an internal tension bolt torqued to a stretch of 0.0018"; a hollow one-piece pin (to simulate a one-piece crankshaft; a face-splined with plug; and a face-splined with an internal tension bolt stretched to 0.0068".

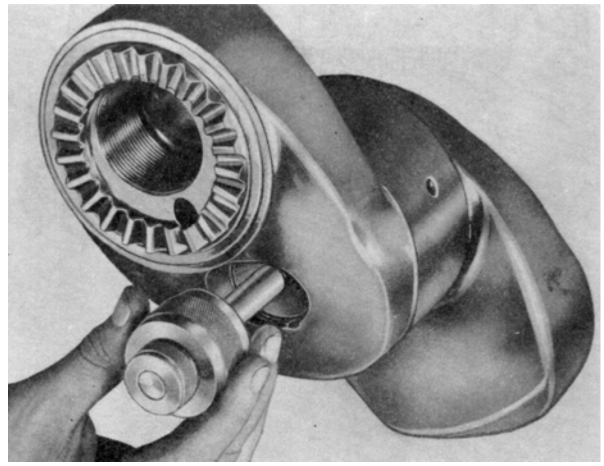


Figure 5.6 Detail of Face Splines **(Pratt & Whitney)**

The results are presented in Figure 5.7, which strongly supports the argument that the face-splined construction with proper tension bolt torque is far superior to other designs.[13]

The face-splined crankshaft construction was not without its development troubles. A large bolt centered in each crankpin held the face splines in close contact. It took considerable experimentation and cost George Meloy a lot of sleep before suitable locking pins for this bolt were produced.[14]

By October 29, 1942, the first examples of the face-splined two-counterweight cranks with 4.5X bifilar dampers on the rear counterweight were undergoing torsional and linear vibration testing. It is noteworthy that in this test, master rods were installed twenty degrees apart in cylinders 8 and 9. This arrangement was ideal for eliminating 1X torsional vibration at the expense of 2X torsional vibration.[15] Later addition of a 2X bifilar torsional vibration damper to the front counterweight eliminated the 2X torsional vibration problem inherent to this master rod orientation.

While the crankshaft would undergo continued improvement during its service life, these changes were minor, consisting of things like silver-plating the face spline mating surfaces and use of lighter weight bifilar damper construction. The face-splined joint concept proved itself in service and remains in use in R-2800 "C" and later engines in use today.

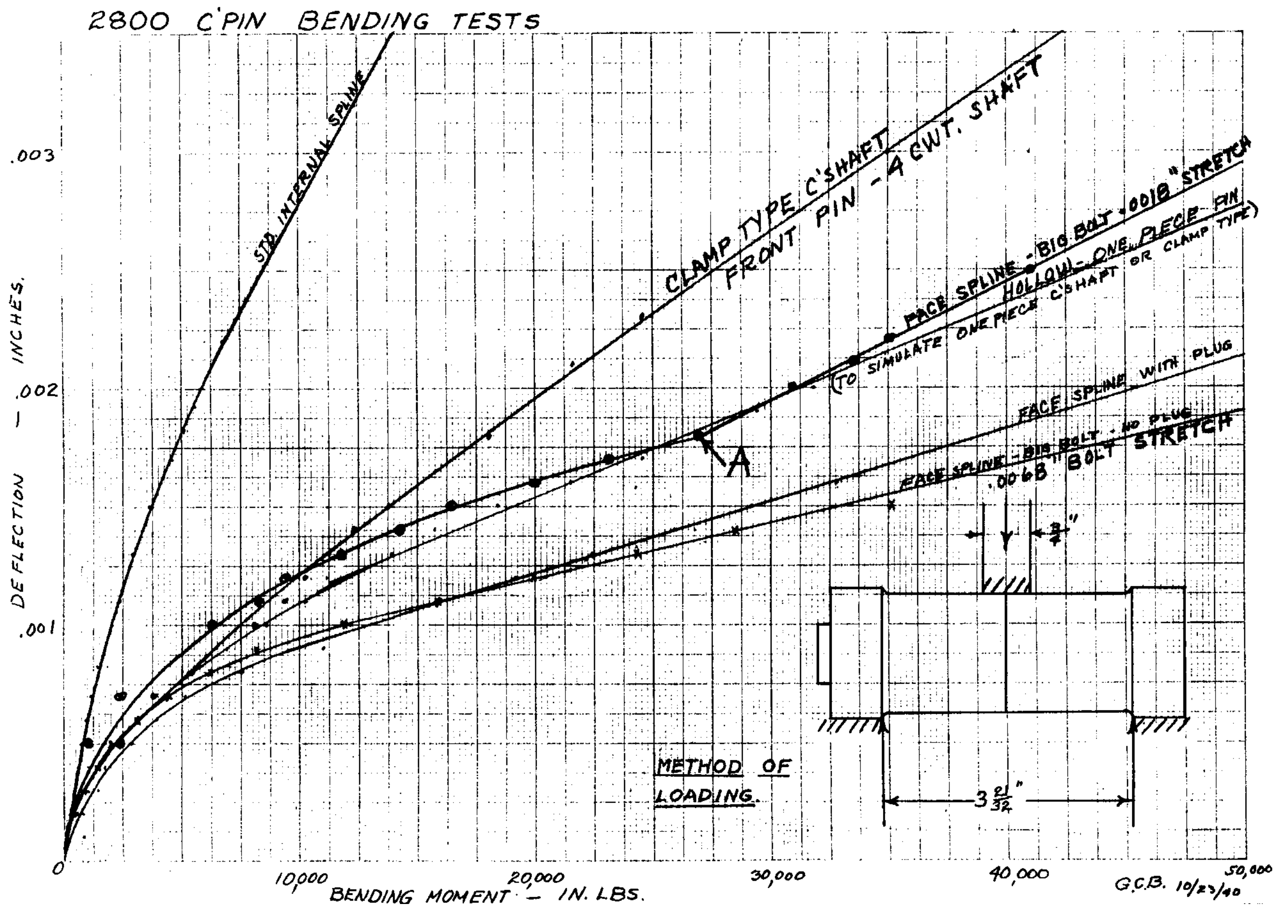


Figure 5.7 Crankshaft Bending Studies **(Pratt & Whitney)**

[1] George E. Meloy, "Report on History of R-2800 Engine Development", (PWA Report No. PWA-192, May 30, 1939), 8.

[2] Ibid., 9.

[3] The two-piece center crankcase had always been problematical. It required additional machining operations in production, and was subject to fretting between the case halves. A one-piece casting would eliminate these difficulties.

[4] R. E. Gorton and A. R. Crocker, "Torsional Vibration of the R-2800 Engine with Two-Counterweight Crankshaft using 6159-0 Hydromatic Propeller With and Without Paddle Dampers", *SMR No. 547* (March 31, 1939).

[5] W. J. Closs, "First R-2800 Two Counterweight Crankshaft", *SMR No. 617* (November 21, 1939).

[6] R. E. Gorton and A. R. Crocker, "Linear and Torsional Vibration of R-2800 Engine X-83 with Loose Crankshaft Counterweight Plugs Operating in the Horizontal Intake 18' Test House", *SMR No. 619* (November 22, 1939).

[7] R. E. Gorton and A. R. Crocker, "Vibration in Bending of the Two-Counterweight R-2800 Crankshaft with Loose Plugs in Rear Counterweight", *SMR No. 622* (December 5, 1939).

[8] R. E. Gorton and A. R. Crocker, "Crankshaft Torsional Vibration and Linear Vibration of the R-2800 Engine with Clamp-Type Crankshaft and Hydromatic 6159-0 Propeller", *SMR No. 569* (June 27, 1939).

[9] W. J. Closs, "R-2800, Two Counterweight, Clamp Type Crankshaft", *SMR No. 609* (October 31, 1939).

[10] "R-2800 Development", (Internal P&W working paper, author unknown, some pages marked "F.W.P. 6-22-45).

[11] Elton Sceggel, telephone interview by the author, (Huntsville, AL, March 22, 1999).

[12] Gordon Beckwith, telephone interview by the author, (Huntsville, AL, March 22, 1999).

[13] G. C. Barnes, "2800 Crankpin Bending Tests", *SMR No. 686* (October 29, 1940).

[14] Beckwith.

[15] R. W Pratt, "Crankshaft Torsional and Engine Linear Vibration of the R-2800-37 Engine X-88 with Two-Counterweight, Face-Splined Crankshaft, a 4 1/2X Bifilar Damper on Each Counterweight, and Master Rods in Cylinders #8 and #9", *SMR No. 871* (November 27, 1942).

6 Conclusion

Despite the problematical development of the R-2800, it became a fine engine. In World War II, it powered numerous fighters and medium bombers, and secured a reputation for ruggedness that was unsurpassed.

Howard Camp, a fighter pilot friend, flew both P-51s and P-47s in World War II. I once asked him which airplane he preferred. “It depends”, he replied without hesitation, “on whether you are shooting or being shot at. You want the Mustang if you are shooting and the Thunderbolt if you are being shot at!”

The R-2800 also had a reputation for being robust. While the Wright R-3350 was a great engine, it required considerable care from its operators. On the other hand, the Pratt & Whitney R-2800 could take a lot of abuse and keep right on going. Just prior to World War II, Frank Walker was responsible for the development of anti-detonation injection (ADI) for the R-2800. ADI forces a water-alcohol mix into the induction system to cool the supercharged fuel-air mixture, thereby allowing a much higher manifold pressures and power outputs. Using ADI, Walker was able to coax 3800 HP from an experimental “C” engine at manifold pressures up to 150 in Hg![1] This is nearly twice the power the engine was designed to produce.

In addition to its reputation for ruggedness in aircraft like the P-47, the R-2800 developed a reputation for reliability in airline service after World War II. It had a recommended time between overhauls of 2000 hours on twin-engine aircraft, and 3000 hours on 4-engine aircraft.[2] The Douglas DC-6 was powered by four R-2800s. When Douglas designed the newer, larger DC-7, it chose the more powerful R-3350, and instructed pilots to run them at high power settings in order to achieve promised performance. There is more than a grain of truth in the old joke “What’s the difference between a DC-6 and a DC-7? The DC-6 is a four-engine airplane with three-bladed props; the DC-7 is a three-engine airplane with four-bladed props.”

The fact that many R-2800s are still in use today nearly sixty years after they were built is testimony to the quality of the vibration solution and crankshaft construction. It is also testimony to the dedication of the engine designers and test engineers. It is no doubt satisfying to Gordon Beckwith, as well as the other test engineers who did not know when to go home, that all of that time spent after hours in the test house was worthwhile.

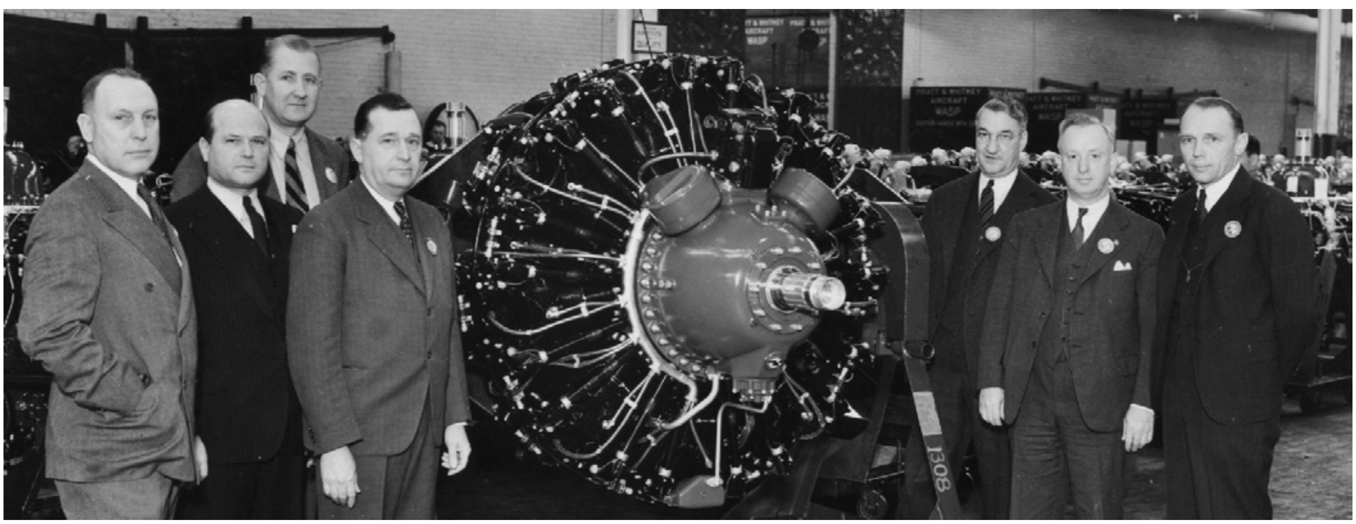


Figure 1. First production R-2800. Pictured left to right are E. Wilson, A. Willgoos, W. Parkins, W. Levack, B. Miller, D. Jack, and L. Hobbs (Pratt & Whitney)

[1] Frank Walker, interview by author (Huntsville, AL, March 29, 1999).
[2] *R-2800 Engine Reliability,* internal memorandum to Pratt & Whitney Field Service Personnel (December 21, 1967).

R-2800 Crankshaft Evolution

MILITARY DESIGNA-TION	ENG SERIES	CRANK MACHINING ASSEMBLY	CRANK BALANCE ASSEMBLY	FRONT COUNTER BALANCE	REAR COUNTER BALANCE	MASTER ROD	PARTS MANUAL DATE	CRANKSHAFT CONSTRUCTION
8, 10, 65	B	96894 > 53509, 76526, 83202	96896 > 53510, 83358	83196 > 37762, 76167	83197 > 37763, 76168	34405	12-1-53	4-counterweight,spline in center of crankpin, two 4 ½ X Pratt & Whitney spool-type dampers in rear counterweight
14W, 22, 22W, 28, 34, 34W, 36, 57, 73, 77, 81, 83, 83WA, 85	C	102193 > 90977, 94710	94710 > 90978	96778 > 84313	96779 > 84314	87017	4-1-55	2-counterweight, face spline, 2X heavy bifilar damper front, 4 ½ X heavy bifilar damper rear
18	C	100874	94711	96788 > 84313	96779 > 84314	87017	11-1-56	2-counterweight, face spline, 2X heavy bifilar damper front, 4 ½ X heavy bifilar damper rear
27, 31, 43, 51, 59, 71, 75, 79	B	96894 > 38318, 53509, 76526, 83202, 92348	96895 > 53659, 59526, 83201, 92419	88196 > 37762, 76167	83197 > 37763, 76168	34405	2-1-66	4-counterweight,spline in center of crankpin, two 4 ½ X Pratt & Whitney spool-type dampers in rear counterweight
30, 32	E	258213 > 110937, 151398, 168172, 199265	258212 > 110939, 151399, 168173, 199264, 233878	236592 > 223393, 84313	87769	144305 > 86132	3-15-55	2-counterweight, face spline, 2X light bifilar damper front, 4 ½ X light bifilar damper rear
42	CE	257037 > 108276, 156615, 190154	258211 > 107147, 179445, 196266, 233876	236592 > 84313, 223393	236593 > 84314, 223394	144305 > 86132	11-1-56	2-counterweight, face spline, 2X light bifilar damper front, 4 ½ X light bifilar damper rear
44, 48, 97	CA, CB, CE	323.036 > 146697, 156615, 156617, 190154, 257037	398395 > 156616, 189249, 257038, 233877, 273703, 310688, 323040	236592 > 84313, 223393	236593 > 84314, 223394	144305 > 86132	3-1-57	2-counterweight, face spline, 2X light bifilar damper front, 4 ½ X light bifilar damper rear
50, 50A, 52W, 54, 99W, 103W	CB	323036 > 156615, 190154, 257037, 309919	398395 > 156616, 189249, 233877, 257038, 273703, 310688, 323040	236592 > 84313, 223393	236593 > 84314, 223394	144305	11-1-65	2-counterweight, face spline, 2X light bifilar damper front, 4 ½ X light bifilar damper rear
CA3, CA15, CA18, CB3, CB16, CB17	CA, CB	323036 > 108276, 146697, 156615, 156617, 190154, 257037	323040 > 101178, 156616, 156618, 189249, 233877, 257038, 273703	236592 > 84313, 223393	236598 > 84314, 223394	144305	6-66	2-counterweight, face spline, 2X light bifilar damper front, 4 ½ X light bifilar damper rear

10-13-99

NOTES: 1) “>” means “supercedes”.