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.

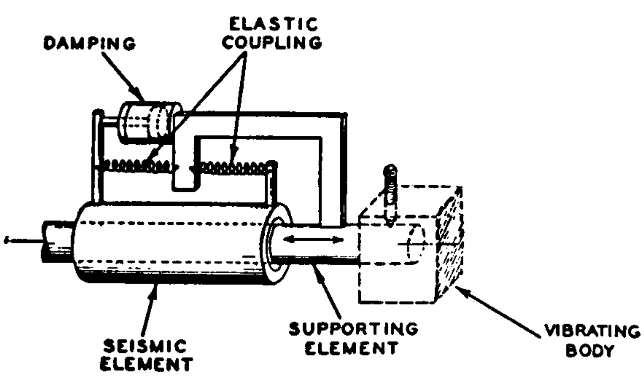


Figure 4.1 Linear Vibration Pickup Mechanical Components (Draper[2])

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.

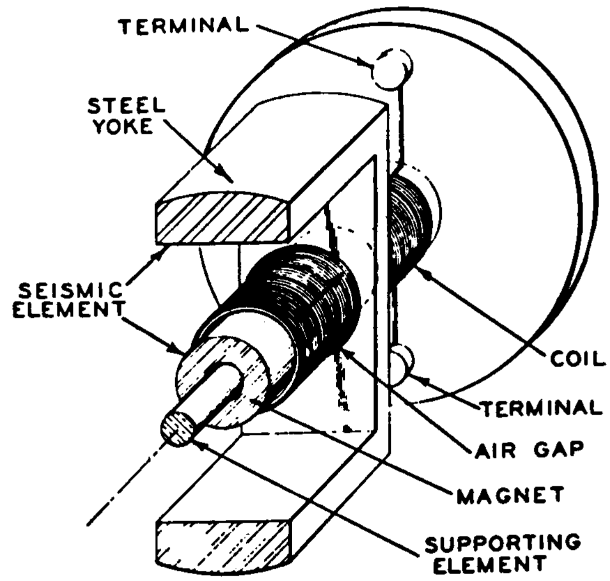


Figure 4.2 Linear Vibration Pickup Electrical Components (Draper[3])

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.

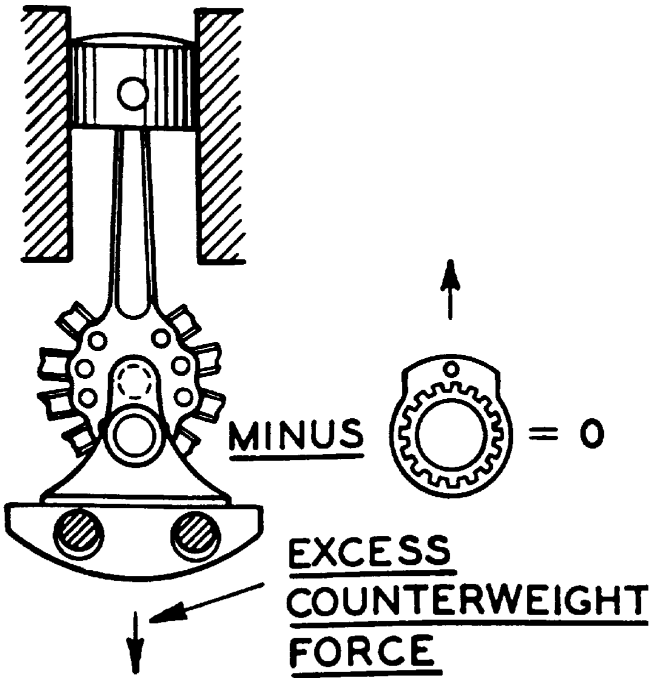


Figure 4.3 Counterbalance Action at TDC.
(Pratt & Whitney)

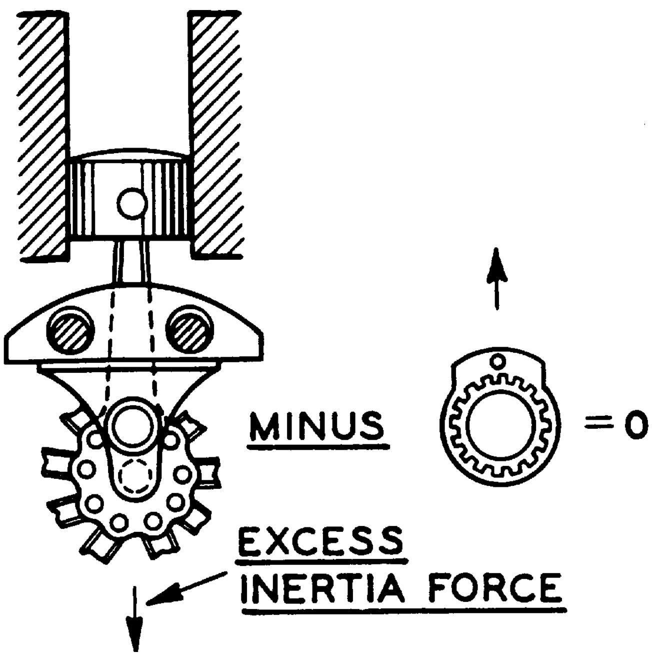


Figure 4.4 Counterbalance Action at BDC
(Pratt & Whitney)

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.

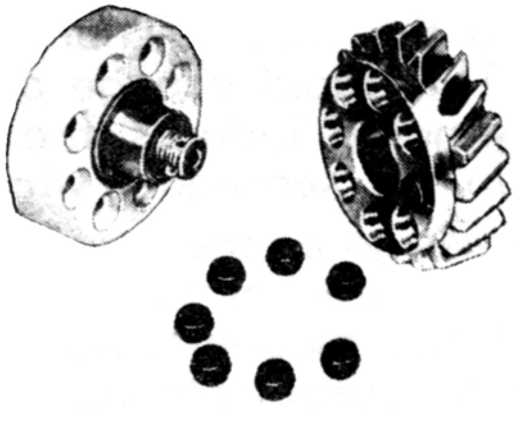
This test also revealed a 3.5X linear vibration for the first time. This 3.5X vibration would prove elusive and troublesome, coming and going from test to test, distracting the team from the more urgent 2X problem. Sprenkle, experienced with similar vibration difficulties in the R-1830-C engine, assigned 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]

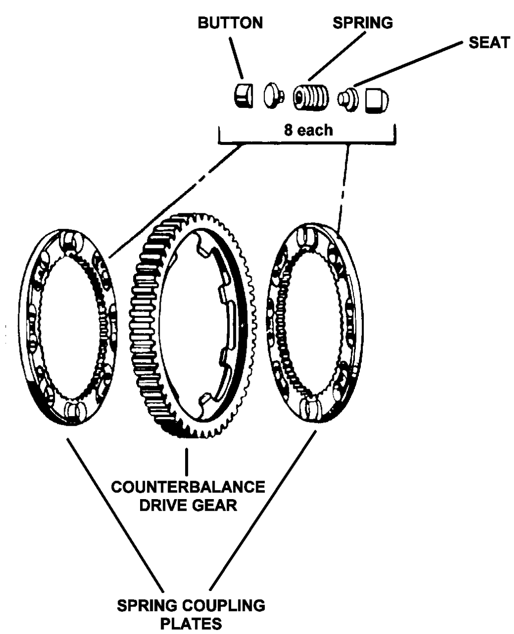
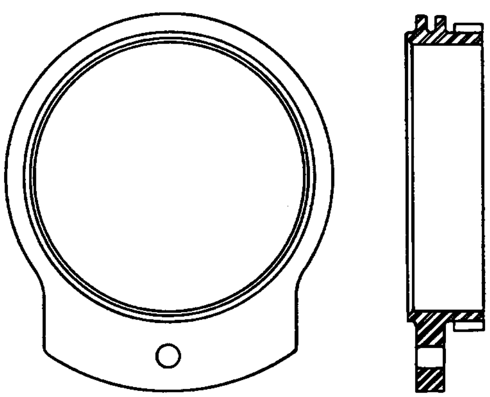


Figure 4.5 Counterbalance Spring Drive (Pratt & Whitney)

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]



FRONT Pt. No. D-37762
REAR Pt. No. D-37763

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.

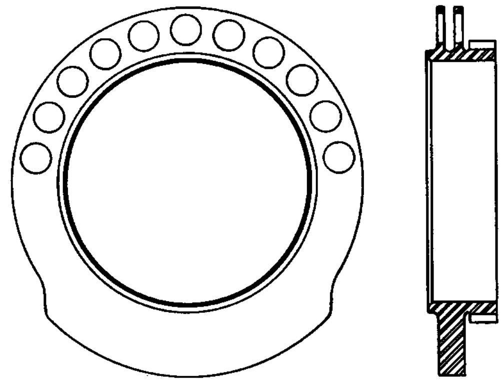


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Figure 4.7 Type II Counterbalance (Pratt & Whitney)

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]

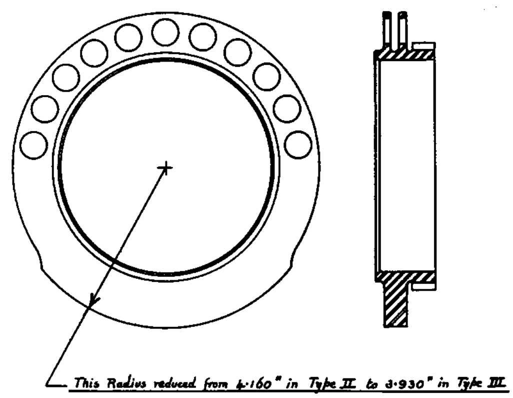

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Figure 4.8 Type III Counterbalance (Pratt & Whitney)

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.

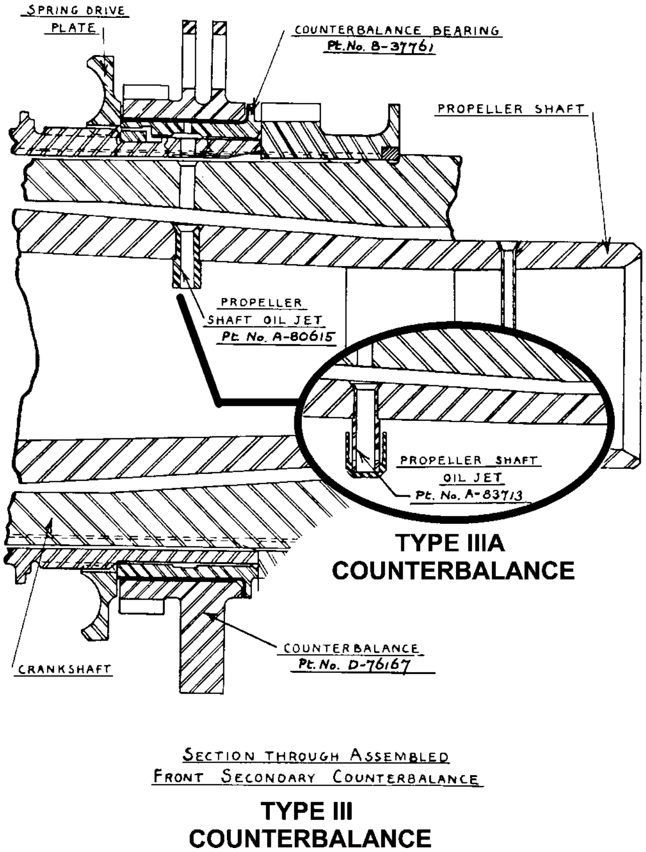

Figure 4.9 Type IIIA Counterbalance (Adapted from Pratt & Whitney)

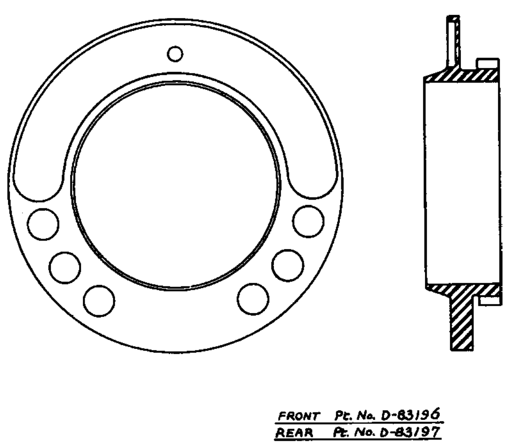

Figure 4.10 Type IV Counterbalance (Pratt & Whitney)

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.

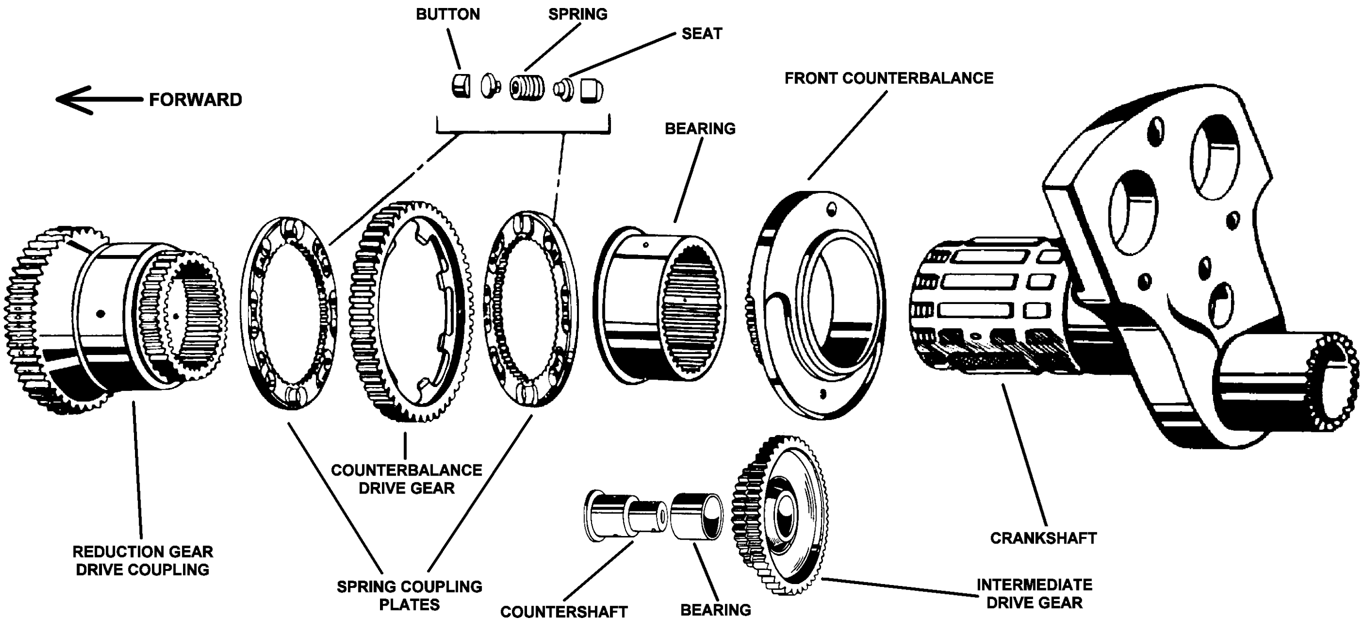

Figure 4.11 Simplified View of Front Secondary Counterbalance and Drive (Pratt & Whitney)

[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).