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.

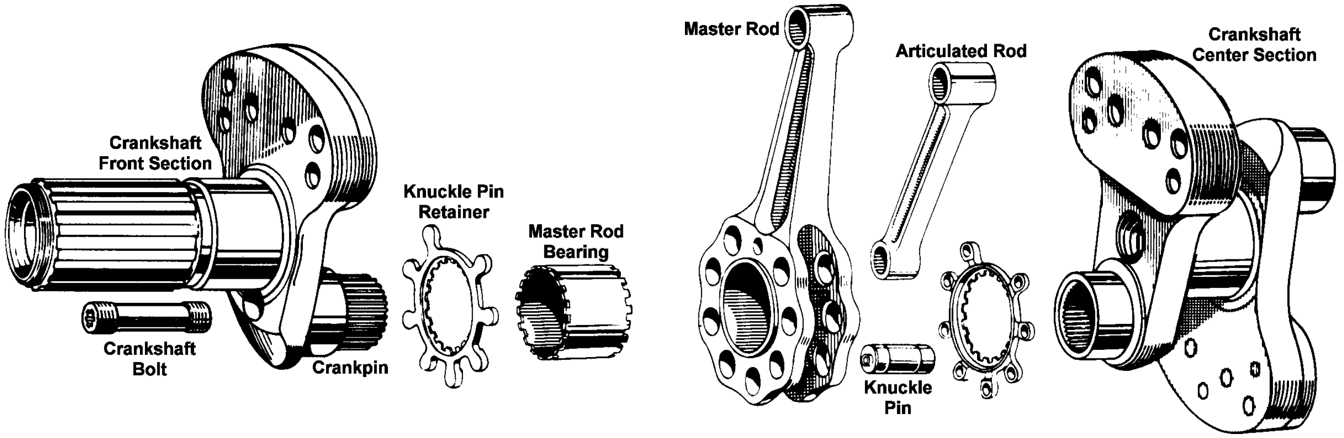

Figure 2.1 Radial Engine Crankshaft Showing Master/Articulated Rod Construction **(Pratt & Whitney)**

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.

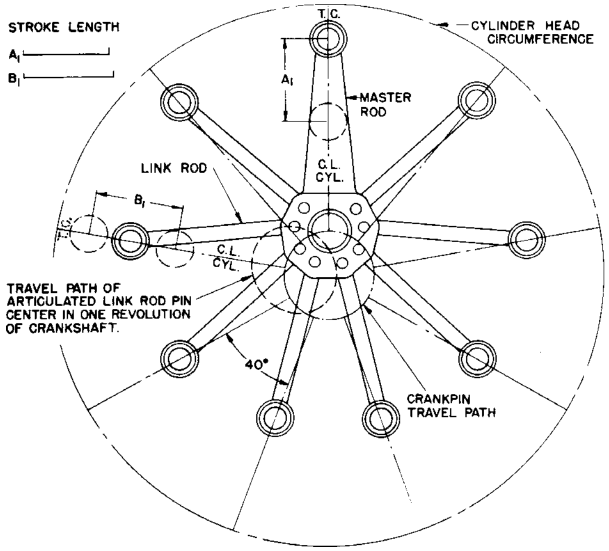

Figure 2.2 Elliptical Knuckle Pin Paths **(Naval Air Training Command)**

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.

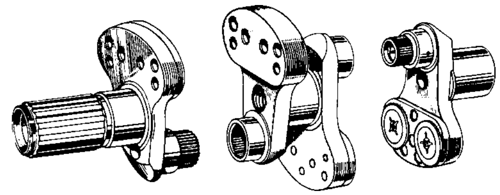

Figure 2.3 Two-row Radial Crankshaft **(Pratt & Whitney)**

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.

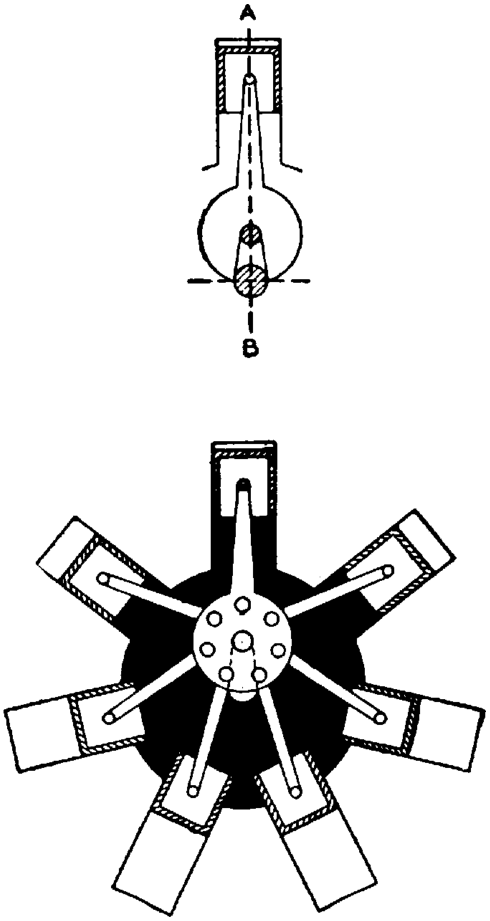

Figure 2.4 Master Rod at Top Dead Center **(adapted from *Aircraft Power Plants*, McGraw-Hill, 1955)**

Figure 2.6 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.

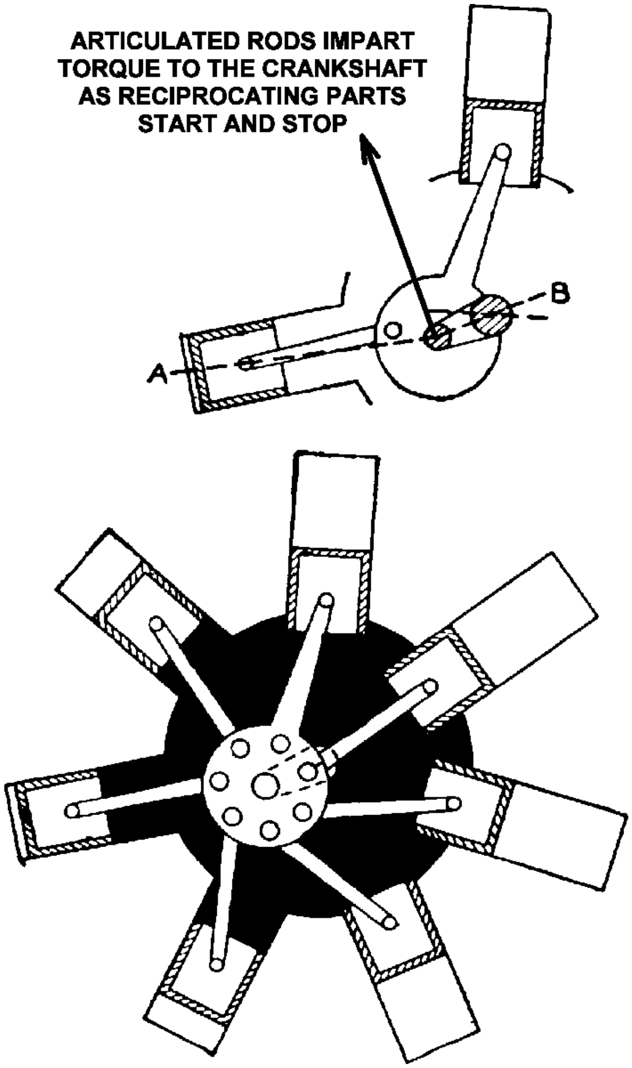

Figure 2.5 Articulated Rod at Top Dead Center **(adapted from *Aircraft Power Plants*, McGraw-Hill, 1955)**

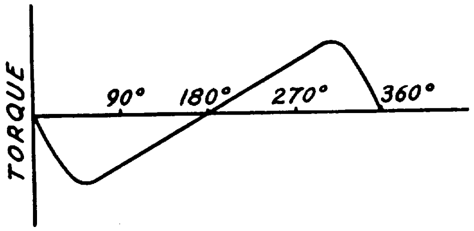

Figure 2.6 Total Inertial Torque Variation **(Pratt & Whitney)**

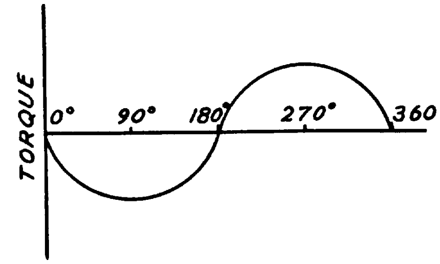

Figure 2.7 First-Order Inertia Torque (Pratt & Whitney)

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.

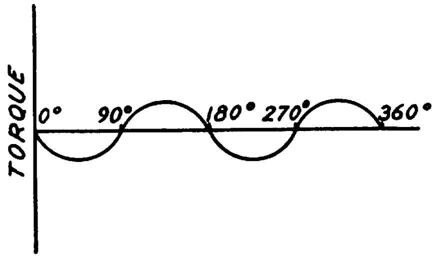

Figure 2.8 Second-Order Inertia Torque (Pratt & Whitney)

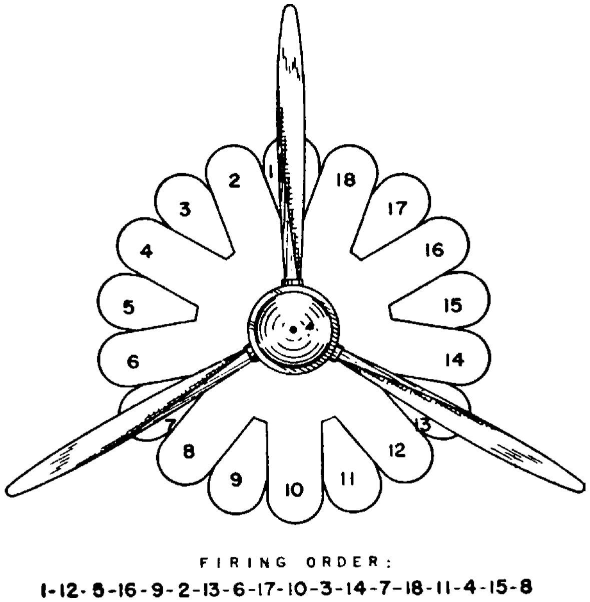

Figure 2.9 Radial Cylinder Indexing

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.

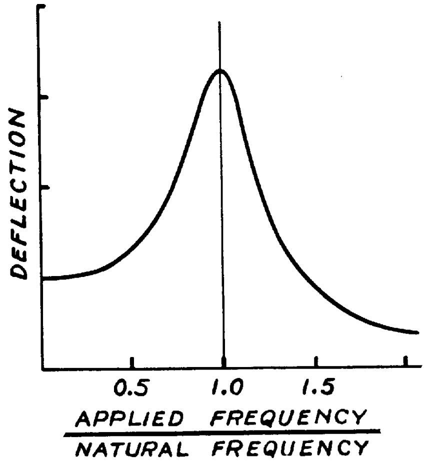

Figure 2.10 Part Behavior in Resonance (Pratt & Whitney)

Engine Anatomy

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

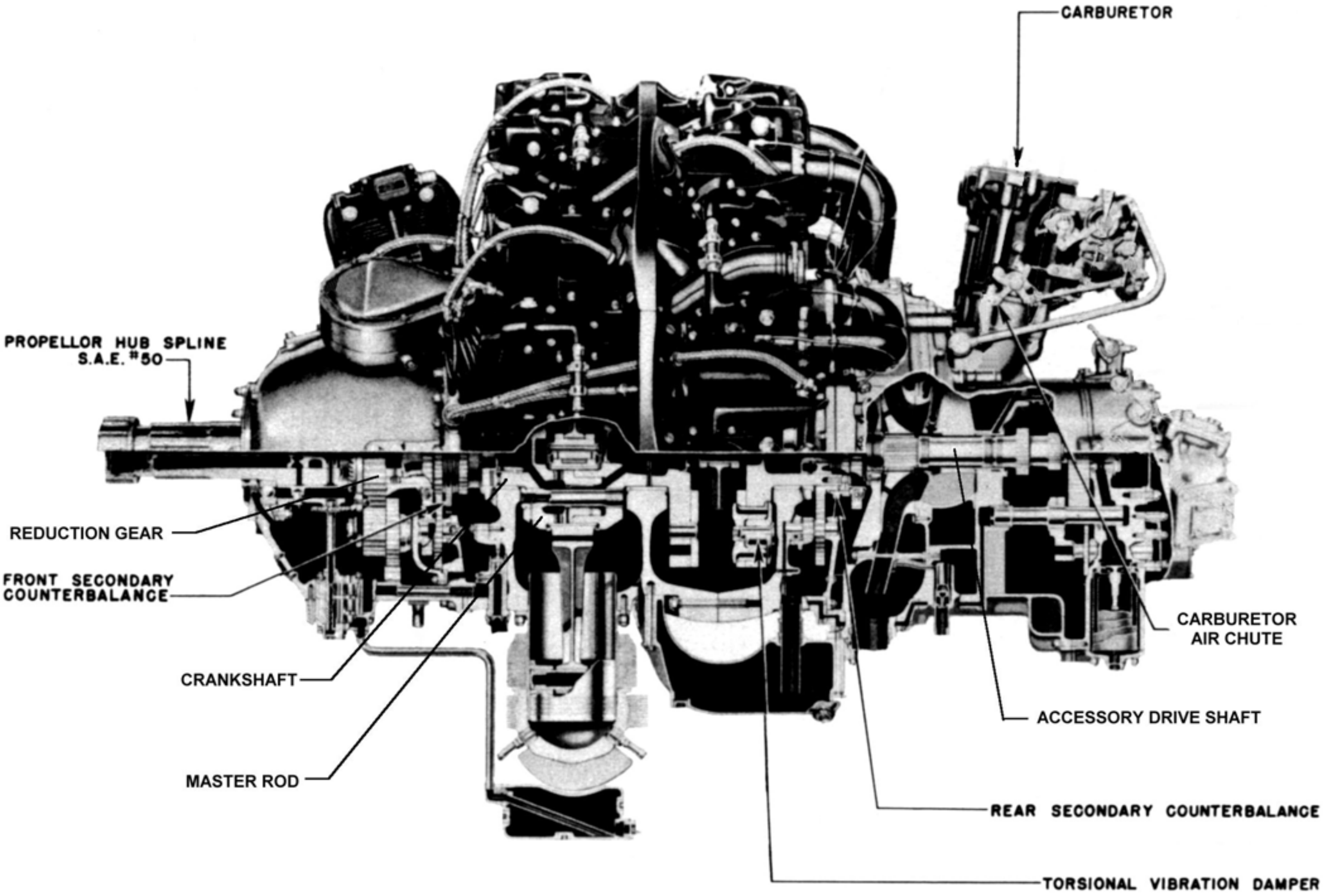

Figure 2.11 R-2800 Left Side View Showing Location of Parts Discussed in Later Chapters (Pratt & Whitney)