
16.1 Definition of Cranktrain Function and Terminology

The Cranktrain is at the heart of the reciprocating piston engine, and its purpose is to translate the linear motion of the pistons into rotary motion for the purpose of extracting useful work. The cranktrain is typically composed of connecting rods, the crankshaft, and a flywheel or power takeoff device.

The crankshaft is usually composed of one or multiple throws, to which the connecting rods are attached with either fluid film journal bearings as shown in Fig. 16.1 or in some cases rolling element bearings. The crankshaft is typically cast or forged of iron or steel. In addition to its main function of translating linear to rotary motion, the crankshaft drives many of the engine accessories including the valvetrain, oil pump, water pump, and charging system. Crankshaft rotation direction can be either clockwise or counterclockwise, depending on driveline packaging and requirements. In most automobile applications the rotation direction is clockwise as viewed from the crankshaft nose and front of the engine. The crankshaft typically incorporates additional balance weights as have been discussed in Chap. 6.

The connecting rod in its simplest form is a beam with a pin joints at each end as shown in Fig. 16.2. The *small end* sees stop-and-start motion as it is connected to the piston, while the *big end*, attached to the crankshaft sees high rotation speeds. The connecting rod supports bearings or bushings at each end. It is one of the most stressed components in the engine and subjected to high tensile, compressive, and bending stress. It can be cast, forged, or sintered from powdered metal.

The flywheel serves several functions. First, it adds additional rotating inertia to the crankshaft to reduce the cyclical speed variation produced by the reciprocating pistons and intermittent combustion events. When attempting to move a vehicle from stationary, the flywheel provides stored energy to overcome the inertia of the vehicle for smooth

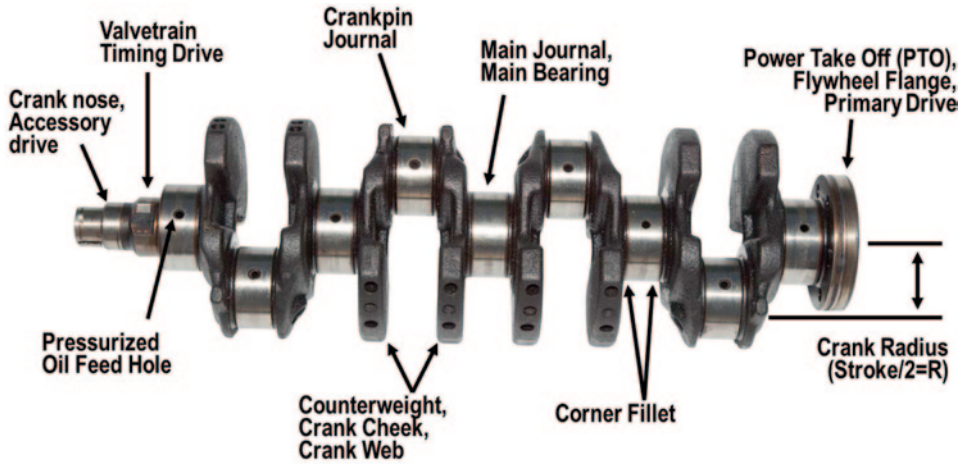


Fig. 16.1 Crankshaft terminology

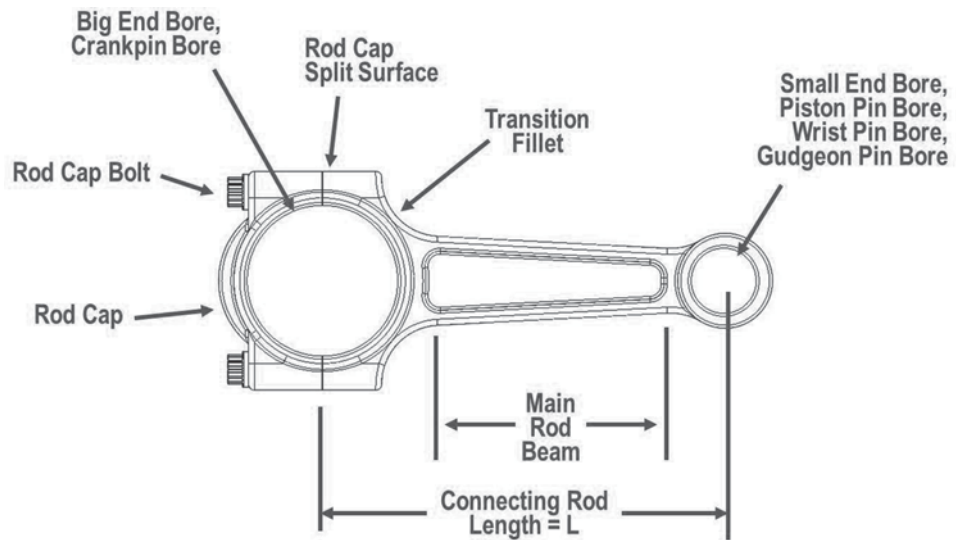


Fig. 16.2 Connecting rod terminology

starts. If the inertia of the cranktrain is too low, the engine may stall or unnecessarily high engine speed will be required to launch the vehicle from rest. A further function of the flywheel is power transfer to the vehicle drivetrain; the clutch assembly or torque converter is mounted to the flywheel, and contributes to the total flywheel mass. A final function of the flywheel is to hold the ring gear against which the starter will engage. An example flywheel bolted to the rear crankshaft flange of a large diesel engine is shown in Fig. 16.3.

Fig. 16.3 Flywheel mounted to crankshaft

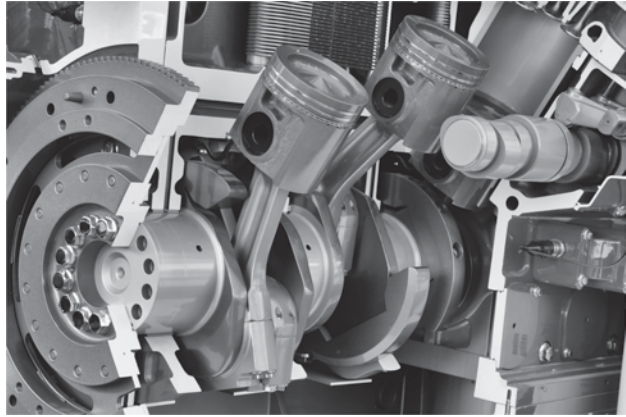
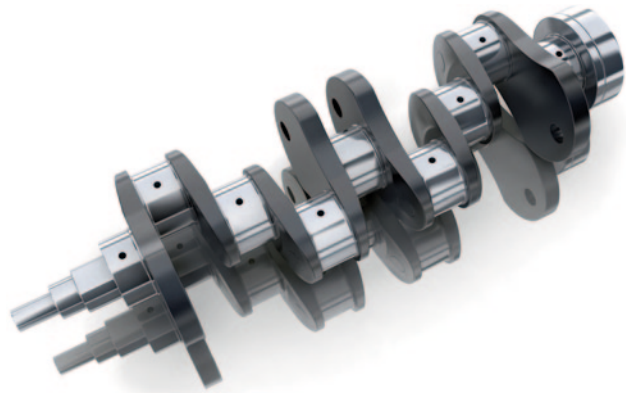


Fig. 16.4 Single-plane crankshaft



16.2 Description of Common Cranktrain Configurations and Architectures

16.2.1 Crankshaft Configurations

Crankshaft throw configurations were previously discussed in Chap. 6, as determined by engine balance and firing order requirements. Multi-throw crankshafts can be arranged in either a single plane or multiple planes. A single-plane or flat-plane crankshaft has all crankpin throws at either 0 or 180°, as shown with the in-line four cylinder example in Fig. 16.4. A multi-plane crankshaft has throws in more than one plane; the cruciform arrangement used in most V-8 engines is given as an example in Fig. 16.5.

The vast majority of engines use a single piece crankshaft that is cast or forged and then machined to achieve its final geometry. Each of the examples shown in Figs. 16.1, 16.4 and 16.5 are single piece crankshafts. In very low volumes a single-piece crankshaft might be machined from billet steel. The single-piece crankshaft requires that both the main and rod bearings are split for assembly. Connecting rod and cap design will be covered later in

Fig. 16.5 Dual-plane crankshaft

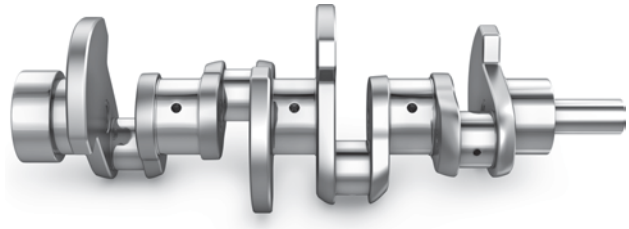


Fig. 16.6 Assembled crankshaft



this chapter. Various approaches for accomplishing the split main bearings were covered in Chap. 8 on cylinder block design.

An alternative to the single-piece crankshaft is the assembled crankshaft such as that shown in Fig. 16.6. The assembled crankshaft is composed of multiple pieces that were manufactured separately, then joined together to form a single crankshaft by means of a press fit, bolted, or welded joint.

An assembled crankshaft is typically used on either very small or very large engines to ease the manufacture. The manufacture of small crankshafts, typically single throw, can be made less expensive since the creation of multiple circular pieces are made on a simple lathe and are then centerless ground. The requirement for a more expensive offset crankshaft grinding machine is eliminated. This also makes manufacture of very large crankshafts for ships and stationary pumping engines easier, as they can be made of more easily managed ‘small’ pieces.

One of the key challenges for the assembled crankshaft is aligning all of the sub-components during assembly to a high tolerance. This usually requires the application of large force to assemble and then straighten the components to achieve the desired straightness

Fig. 16.7 Plain connecting rod**Fig. 16.8** Angled split connecting rod**Fig. 16.9** One piece connecting rod

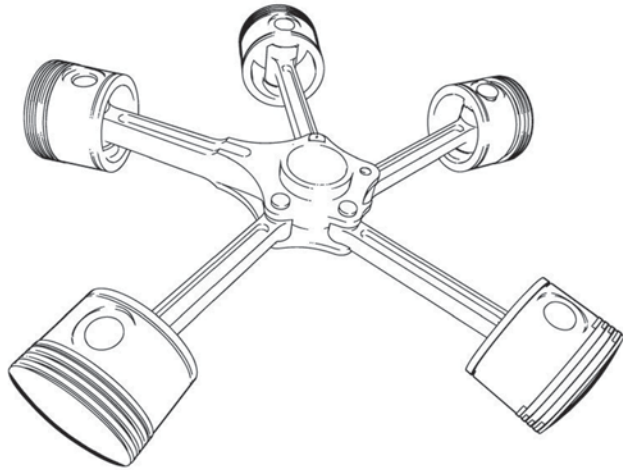
of the crankshaft for the alignment of bearings. The key failure mode of an assembled crankshaft is the misalignment of these components during operation. The torque of the engine may cause the assembled joints to slip with respect to one another, and “scissor”, leading to misalignment of the crankshaft.

16.2.2 Connecting Rod Configurations

There are four basic configurations for the big end or crankpin bore of the connecting rod as shown in Figs. 16.7, 16.8, 16.9, and 16.10.

1. Plain (most common),
2. Angled split (allowing assembly of larger diameter rod bearings),
3. One piece (for rolling element bearings), and
4. Articulated (for radial engines).

Fig. 16.10 Articulated connecting rod



The plain connecting rod is most common in current automotive practice. An angle split connecting rod allows a larger diameter rod bearing while maintaining capability to remove and replace the connecting rod through the top of the engine. This is common in over the highway trucks and locomotives, which have a large crankpin and consequently large bearing. A disadvantage of an angled split rod is non-symmetrical loading of the connecting rod crankpin bore. The rod cap is now at an angle to the loading, and it makes the cap slide with respect to the upper rod, thus requiring additional attention to cap locating. A one-piece connecting rod allows a rolling element bearing without a split cage to be used to reduce friction and lower lubricant flow. However, the one-piece connecting rod is limited to engines with an assembled crankshaft. An articulated rod is used for radial engines where more than one connecting rod must attach to the same crank journal. This arrangement reduces the axial length of the crankpin journal for multiple cylinders, and is most common in air-cooled radial aircraft engines.

Regardless of crankpin bore arrangement one of the requirements of the connecting rod is to minimize crankpin bearing deflection to ensure an even oil film and prevent pinching of the rod bearing relative to the crankshaft. The connecting rod cap may have reinforcing ribs to increase stiffness, as shown in Fig. 16.11. A single rib is easiest to cast or forge, but a double rib provides a more efficient distribution of stiffness.

There are two general configurations of connecting rod beam cross-section. The I-Beam is shown in Figs. 16.7 through 16.10, and again in Fig. 16.13. An H-Beam connecting rod is shown in Fig. 16.12. The beam is the portion of the connecting rod that connects the piston pin and crankpin ends, and when viewed in cross section resembles a capital letter “I” or capital letter “H”.

The I-Beam connecting rod is most common in use because it puts the most material in the areas of maximum bending stress while reducing material where it is not needed. The I-Beam shape also lends itself well to the casting, forging, or Powdered Metal (PM) processes. The pull direction for the beam is in the same direction of the pin bosses, making forming easier and reducing the requirement for material removal.

Fig. 16.11 Rib on connecting rod cap



Fig. 16.12 H-beam



The H-Beam connecting rod is more difficult to manufacture, but this geometry offers a more stable, lighter design. The H-beam rod allows a more gradual transition of stiffness between the main beam and the crankpin end of the connecting rod, reducing stress in this critical radius. It also has greater bending stiffness in the crankpin axial direction than an I-beam rod. This may lead to increased edge loading on the crankpin bearing, when two rods share the same journal. Due to the added expense of manufacture, this is usually only seen in competition engines.

Fig. 16.13 Pressed small end bushing, tapered piston pin end geometry, splash oil hole



If the small-end boss is to be used with a full-floating piston pin a pressed bushing such as is shown in Fig. 16.13 is typical. If a bushing is not used, a coating is required on one of the two mating components to reduce wear. The small end of the rod sees only reversing, stop-and-start motion, resulting in mixed-film lubrication. In some engines an oil feed hole is added supplying the connecting rod piston pin bushing either as a passive splash oiling hole from the top or a pressurized lubricant drilling from the crankpin bearing.

16.2.3 Flywheel Configurations

The most common arrangement of flywheel in automotive applications is a separate disc attached to the end of the crankshaft through a bolted joint. The total cranktrain inertia is composed of the flywheel, the crankshaft, the rotating mass of the connecting rods, the accessory drive pulley and vibration dampener, and the clutch or torque converter. Occasionally the flywheel is incorporated as part of the crankshaft counterweights, the disc varying in thickness to achieve the desired balance counterweight. This is common practice in motorcycle applications and is seen in the assembled crankshaft shown in Fig. 16.6.

16.3 Detailed Design of Crankshaft Geometry

The geometry of a crankshaft is complicated, and the loading varies as a function of its rotational position. Because of the dissimilar cross-sections of the crankshaft, it will be sensitive to discontinuities in stiffness. It is at these stiffness discontinuities that stress will be concentrated. Generally, the crankshaft fillet will be the most highly stressed area and will require the most detailed design attention as the materials used to make crankshafts are typically sensitive to notch factors. Loading at the fillet alternates as shown in

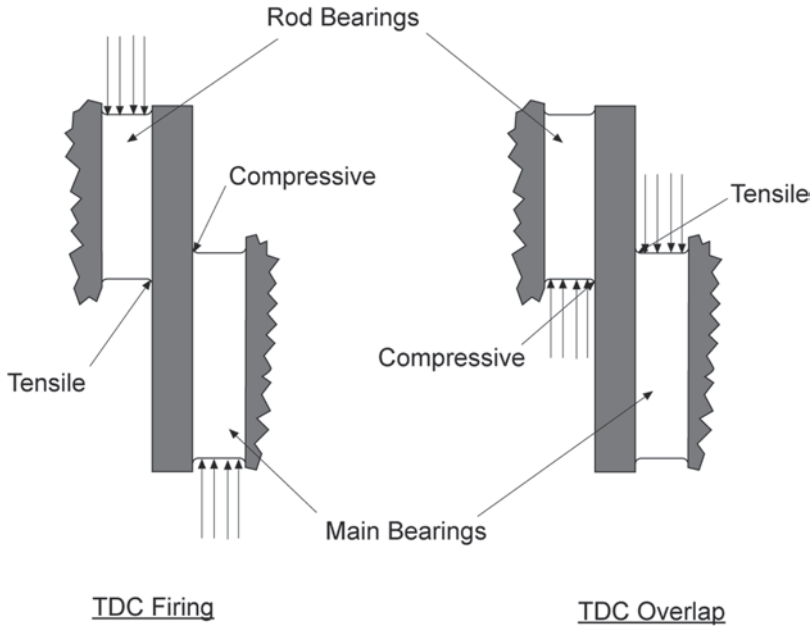


Fig. 16.14 Main and rod bearing loading near TDC firing (a) and TDC valve overlap (b)

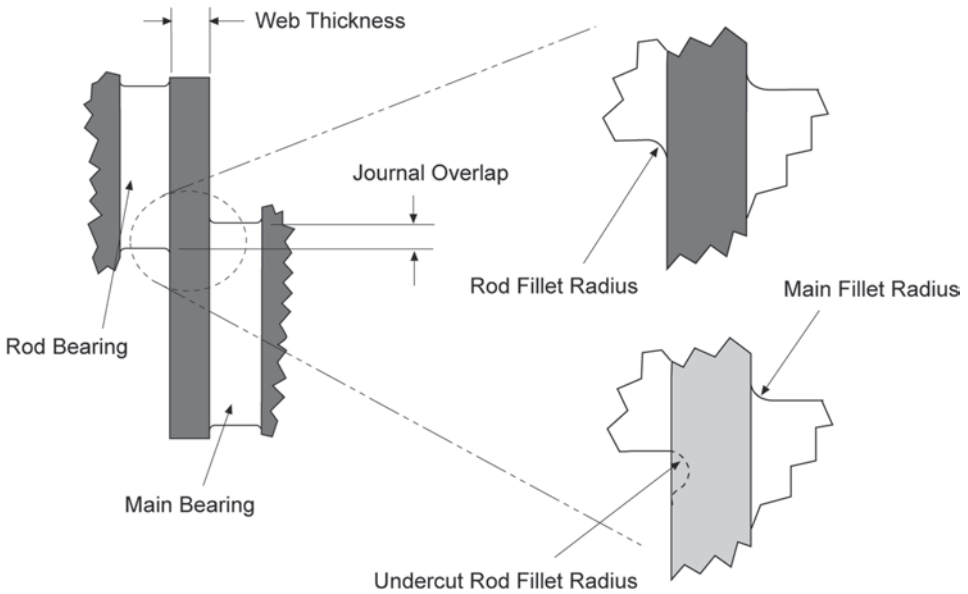
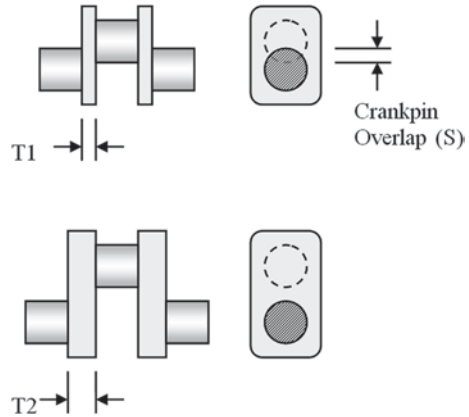


Fig. 16.15 Rod fillet stress versus crank angle for each cylinder of an in-line four-cylinder engine

Fig. 16.16 Critical crankshaft layout dimensions



Figs. 16.14 and 16.15. TDC-Firing puts the crankpin in compression, and TDC-Overlap puts the crankpin in tension. Stress in this area is further magnified by torsional vibration, to be discussed later. Often metal improvement processes, such as shot peening or fillet rolling, are used to introduce residual compressive stress at the fillet radii to improve fatigue life.

Pin overlap, crankshaft web thickness, and journal fillet radius are the key parameters that are varied in crankshaft design. These variables are depicted in the layout in Fig. 16.16. A crankshaft having greater pin overlap (large journal diameters and short stroke) does not require as great a web thickness. A crankshaft that has little or no pin overlap will require a thicker web to gain stiffness.

One way to increase pin overlap, is to increase journal bearing diameter for a given engine stroke, as shown in Eq. 16.1. This will increase crankshaft stiffness, but increasing pin diameter increases fluid film friction at the bearings, reduces net engine power, and increases fuel consumption. It also increases crankshaft weight as the increased mass of the crankpin will need to be offset by increasing mass in the counterweights.

$$S_{Overlap} = \frac{(D_{crankpin} + D_{Main} - Stroke)}{2} \quad (16.1)$$

Where:

- $S_{overlap}$ = Pin overlap
- $D_{crankpin}$ = Diameter of the crankpin journal
- D_{main} = Diameter of the main bearing journal

Fillet radii have enormous effects on the maximum stress in the crankshaft. Once the basic proportions of the crankshaft have been determined, the choice of these radii is critical.

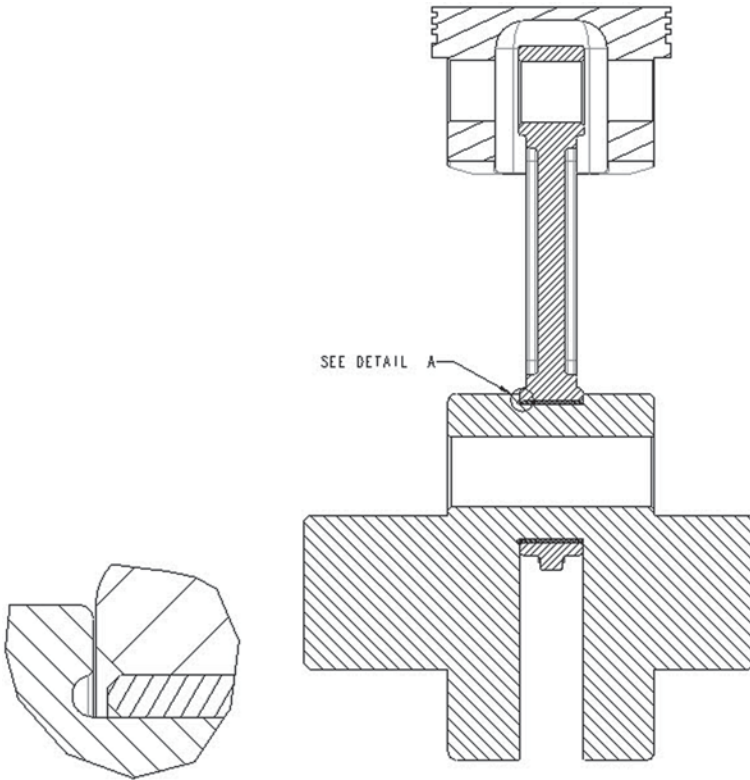


Fig. 16.17 Crank fillet radius, undercut fillet

The larger the fillet that can be used on the crankpin or main journal, the less the stress concentration notch factor will be. However, the larger the journal fillet, the less surface area is available for the connecting rod or main bearing as shown in Fig. 16.17. Several things can be done to minimize the stress at these fillet radii, but at additional cost. The geometry can be improved by using an undercut fillet as shown in the figure to increase the effective radius. The material properties can be improved by introducing residual compressive stresses by fillet rolling or shot peening the fillet radius.

When designing a new crankshaft, it is useful to have a starting point. This can come from benchmarking of successful crankshafts in similar applications, or from initial sizing values. These initial sizing values for the crankshaft were developed in the 1930's and 1940's in an era before computer aided design, and were developed by loading crankshafts and measuring displacement with extensometers or strain gauges. In modern practice, initial values will be assumed and the design will progress to three-dimensional structural analysis. Initial sizing guidelines are listed in Table 16.1. These values are given as a function of cylinder bore diameter.

Table 16.1 Initial sizing values

Feature	Initial sizing value
Cylinder bore diameter	D
Cylinder spacing	$1.20 \times D$
Crankpin diameter	$>0.6 \times D$
Crankpin journal width	$0.35 \times D$, width/dia. >0.3
Main journal diameter	$0.75 \times D$, $>$ pin dia.
Main journal width	$0.40 \times D$, width/dia. >0.3
Web (cheek) thickness	$0.25 \times D$
Crankpin fillet	$0.04 \times D$, $>0.05 \times$ journal dia.
Main fillet	$0.04 \times D$

The majority of external loads are applied to the crankshaft perpendicular to its rotational centerline, and the reaction forces are thus transmitted through the rod and main bearings. However, in addition to these loads the crankshaft is exposed to some thrust loading—loading applied along the axis of crankshaft rotation. Thrust loads occur as the clutch in a manual transmission application is engaged or disengaged. With an automatic transmission, the load transfer through the torque converter includes a thrust component. If the camshaft is driven with a gear train using helical gears a further thrust load is transmitted to the crankshaft. As the engine fires, the crankshaft throw deflects, and the main bearings spread axially. Finally, dimensional stack-up between the crankshaft and the connecting rods and cylinder bore centerlines results in a small thrust load. For all of these reasons the crankshaft must include a thrust bearing surface. This is typically provided in conjunction with one of the main bearings. Because the largest thrust loads are generated at the rear of the crankshaft the thrust bearing is often placed at or near the rear main bearing; for packaging reasons the second-to-rear main bearing is often used since the rear main bearing must also incorporate the rear oil seal. If the crankshaft stiffness is sufficient the middle main bearing may be chosen. This is done because the machined thrust surface provides the fore and aft datum for crankshaft machining. Placing this datum in the middle of the crankshaft allows the fore and aft tolerances to be split equally between the front and rear portions of the crankshaft, making machining process control easier.

16.4 Crankshaft Natural Frequencies and Torsional Vibration

Torsional vibration results whenever an unsteady or cyclical load is applied to a spinning shaft. A number of components in engines, including camshafts, water and oil pump drives, various accessory drives, and the crankshaft meet the criteria for potential torsional vibration problems. Of these the crankshaft receives by far the most attention since the combination of the length of the shaft and the magnitude of the forces result in the most severe conditions. Torsional vibration increases the stresses in the crankshaft webs, and

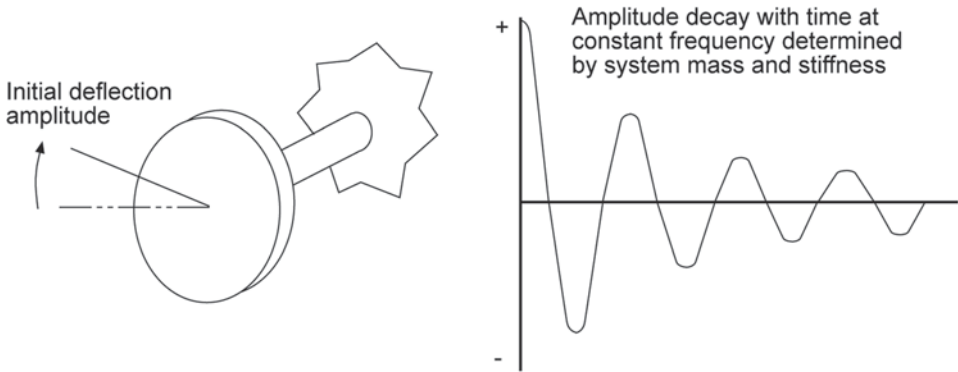


Fig. 16.18 Simple case of Torsional vibration with single degree of freedom

the resulting oscillation of the crankshaft nose loads and unloads the camshaft and accessory drives. The latter significantly increases drive wear and engine noise. In heavy-duty engines camshaft torsional vibration is receiving increasing attention, and examples of vibration problems in accessory drives can be readily found. The focus in this section will be on the crankshaft, but many of the principles discussed here can be applied to other components.

In Fig. 16.18 and Eq. 16.2 the fundamental concepts of torsional vibration are summarized. Looking at this simple case a disk of some appreciable mass is rigidly mounted at the end of a shaft, the other end of which is mounted to prevent it from spinning. A torque is then applied to the disk to rotate it slightly from its initial position, elastically deforming (twisting) the shaft. The torque is then suddenly released, and the shaft unwinds. The shaft seeks to return to its original non-deformed state, but the mass of the disk results in an overshoot and the shaft is twisted in the opposite direction. The disk will oscillate back and forth at the natural frequency of the system, with decaying amplitude in each successive oscillation. The natural frequency is determined by the mass of the disk and the stiffness of the shaft.

$$k_r = \frac{G \cdot J}{L} = \frac{\pi d^4 G}{32L} \quad (16.2)$$

Where:

k_r = Torsional spring constant of the shaft

d = Diameter of shaft

G = Shear Modulus

L = Length of shaft

If the shaft consists of several sections of different diameters, as in a stepped shaft or crankshaft, the equivalent torsional spring constant can be calculated in same way as for springs in series as shown in Eq. 16.3:

$$\frac{1}{k_{eq}} = \frac{1}{k_{r1}} + \frac{1}{k_{r2}} + \dots \quad (16.3)$$

The natural frequency can be modeled using Eq. 16.4:

$$\omega_n = \sqrt{\frac{k_{eq}}{I}} \quad (16.4)$$

Where:

I = Inertia of Disc

The case just described can now be extended in two important ways. First, one can imagine that the shaft and disk are spinning at some constant speed and that the torque is applied to and released from the spinning disk. In this case the resulting oscillation of the disk will be superimposed on the mean speed of the shaft and disk. The next extension is to consider a case where the spinning shaft has not one but several disks, and at various points in time sudden torque impulses are applied in succession to various of the disks. The problem has now become appreciably more complex as different portions of the shaft twist and untwist relative to other portions. Where the first two cases each had one *degree of freedom* and a single *natural frequency* this third case has an additional degree of freedom and natural frequency for each additional disk. Assuming the mass of each disk, and the shaft stiffness between each disk, are known, and assuming that the torque impulses applied to each disk can be characterized as a function of time the torsional vibration can still be calculated. The resulting matrix of equations is difficult to solve by hand, but can readily be addressed with computer calculations.

The third case just described is exactly that of the crankshaft in a multi-cylinder engine. In order to address crankshaft torsional vibration it is necessary to characterize both the crankshaft system and the system excitation—in this case the torque applied to the system at each cylinder.

Turning first to the crankshaft system it is helpful to characterize this system as a series of disks connected by stiff springs as shown in Fig. 16.19. Each disk represents the rotating mass associated with a portion of the crankshaft system. The first disk represents the crankshaft nose, vibration dampener, and accessory drive pulleys. Disks two through seven represent the cylinders of this in-line six-cylinder engine. The eighth disk represents the flywheel or torque converter. It should be noted that the transmission and the remainder of the drivetrain are not represented. This is an accurate approach in automotive engines because transmission of torsional vibration is minimized through the fluid coupling in the torque converter and the clutch pack typically has a torsional compensator built in. If the

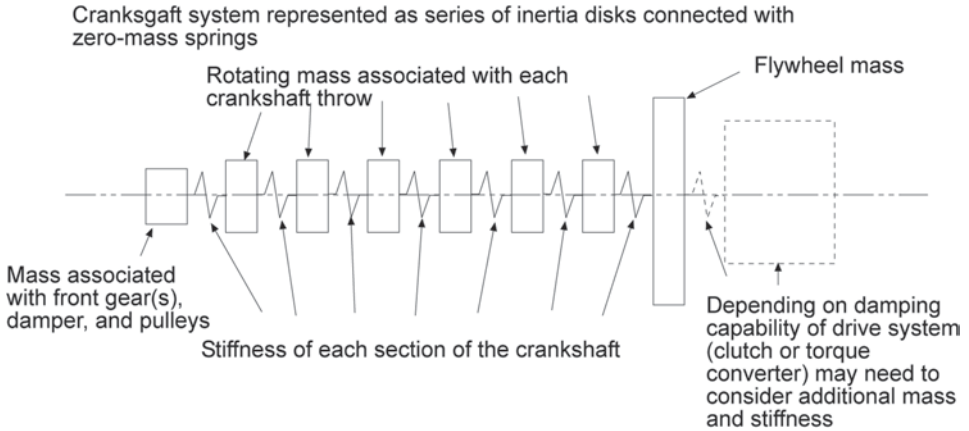


Fig. 16.19 Representation of six-cylinder crankshaft system for torsional vibration analysis

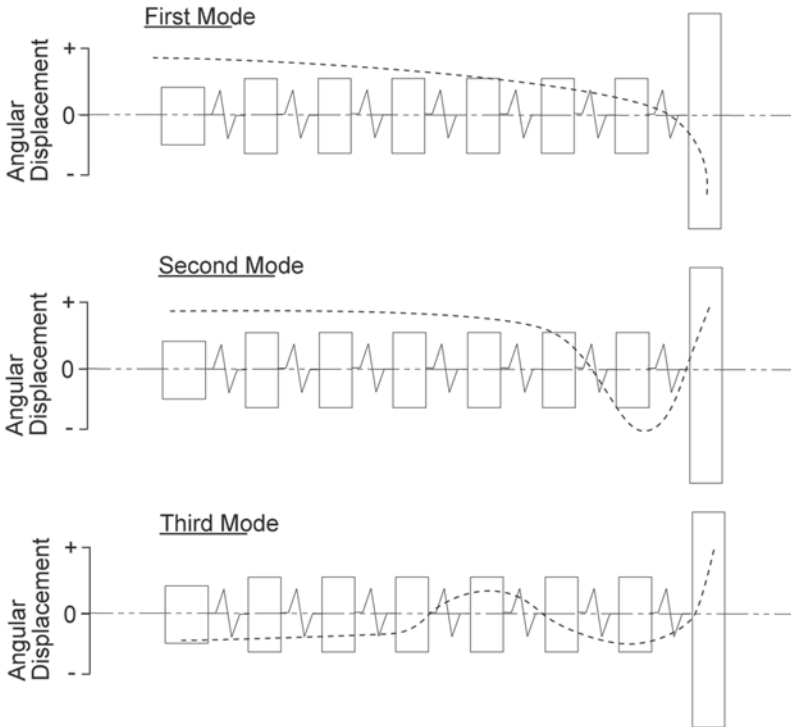


Fig. 16.20 First three modes of a six-cylinder crankshaft system

engine were rigidly mounted to the component being driven (an electric generator, for example) this additional mass would need to be included. The example eight mass system shown here has eight *degrees of freedom* and eight *natural frequencies*. The first three vibration *modes* are depicted in Fig. 16.20. For each vibration mode the crankshaft will have that number of *nodes*—locations along the crankshaft at which angular deflection

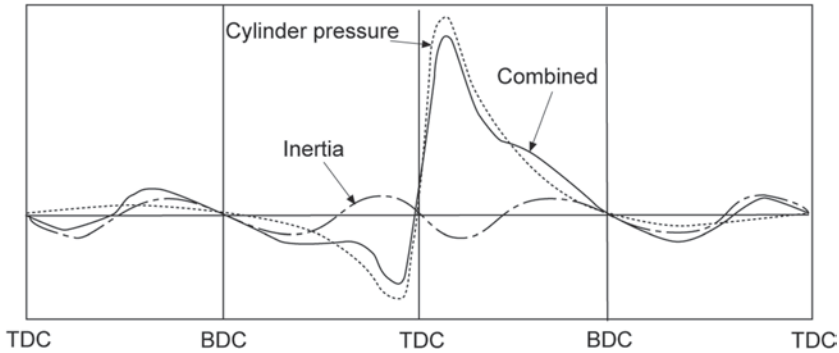


Fig. 16.21 Forces transmitted to crankshaft at each cylinder versus crank angle

relative to the mean crankshaft speed is zero. Because crankshafts are relatively stiff only the first mode or two are generally of interest. As crankshaft stiffness increases the natural frequency of each mode increases; the natural frequencies of the higher modes of a crankshaft system are typically above the frequencies at which significant forces will be seen. It should be noted that for each vibration mode the *anti-node*, or location of maximum angular displacement, is at the crankshaft nose. This results directly from the flywheel location at the opposite end of the crankshaft.

It is now important to look at the exciting forces—the net torque impulse applied at each crankshaft throw, or at each of the disks two through seven in the model shown in Fig. 16.19. The net torque impulse, as a function of crank angle is the net result of the pressure and reciprocating forces, as depicted in Fig. 16.21. Because this combination of forces cannot be directly represented mathematically it is helpful to represent it as a Fourier Series. The torque signal is expressed as the sum of a constant value and an infinite series of harmonics at various amplitudes and frequencies. Because the torque impulse is repeated every second revolution in a four-stroke engine the fundamental frequency is a half-order frequency, repeating every second revolution. The remaining harmonics then represent each half order in an infinite series. The Campbell Diagram shown in Fig. 16.22 shows the frequency of each harmonic, from the half-order through the sixth order, as a function of engine speed.

Torsional vibration requires the application of the torque impulse, now represented as a series of harmonics, to the crankshaft system described earlier. Returning to the Campbell Diagram a given crankshaft will have various natural frequencies—one for each vibration mode described earlier. Each natural frequency can be overlaid as a horizontal line on the Campbell Diagram. It follows that there will be a critical, or resonant speed for every whole and half-order harmonic. Some of these will occur well outside of the operating speed range of the engine.

The torque impulses are being applied at various points along the length of the crankshaft, and in a particular sequence based on the engine's firing order. As a result some harmonic orders assist one another and increase the vibration amplitude. These are referred to as the *major orders*. Others partially cancel one another and are termed *minor orders*. The

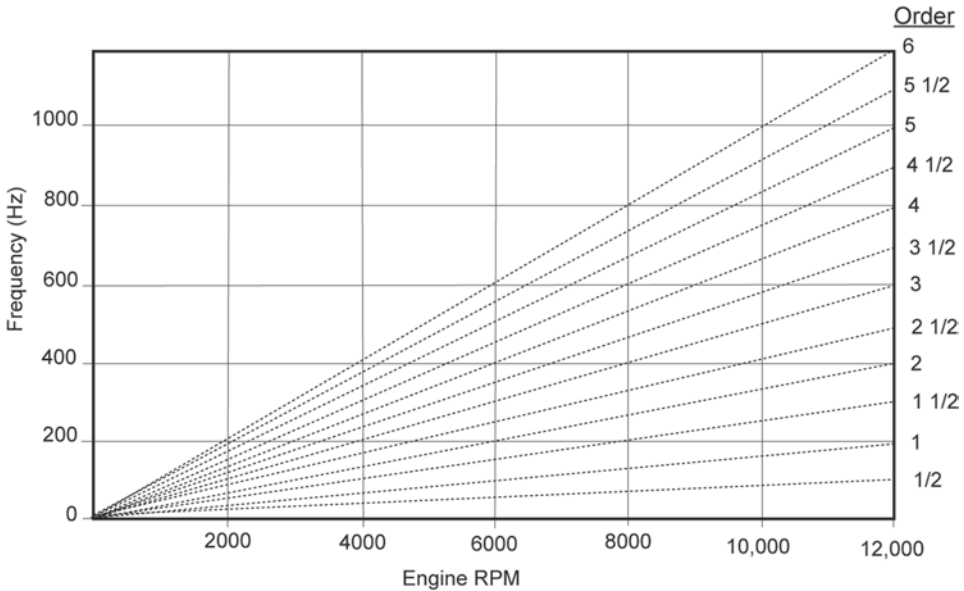


Fig. 16.22 Campbell plot identifying frequency versus engine speed for each vibration order—shown for every half order through sixth order

major orders tend to be those that are direct multiples of the number of torque pulses per revolution. However, other orders may be important depending on the particular crankshaft layout and firing order. For example, on an in-line six-cylinder engine the third and sixth orders might be expected to be important. The ninth order may be important although the magnitude of vibration is typically much lower. The $4\frac{1}{2}$ order will also be found important on most in-line six-cylinder engines. A complete explanation is beyond the scope of this book, but lies in the choice of firing order resulting in relative high amplitude forces acting over the front versus the back halves of the crankshaft at the $4\frac{1}{2}$ order.

As a summary of the concepts just discussed, the third, $4\frac{1}{2}$, sixth, and ninth order frequency versus engine speed of the Campbell Diagram are re-plotted in Fig. 16.23. Overlaid on this diagram are the first- and second-mode natural frequencies of a particular crankshaft system. It can be seen that for this engine resonant speeds for the first vibration mode will occur at approximately 2700 rpm (6th order), 3600 rpm ($4\frac{1}{2}$ order), and 5700 rpm (3rd order). Second mode resonance will occur at 7300 rpm (6th order) and 9600 rpm ($4\frac{1}{2}$ order). Note that unless the engine will be run at very high speed the third crankshaft vibration mode will not be seen.

Some general trends can be observed in crankshaft geometry. Higher stiffness crankshafts have higher natural frequencies which is important for both torsional stresses and lateral (axial) vibrations. Low moment of inertia crankshafts also help raise the natural frequencies of the system. Conversely, adding counterweights adds to the moment of inertia of a crankshaft, lowering the frequency. On multi-throw crankshafts, such as V-6 or V-8 engines, shaping the counterweights is critical to keep the inertia low and the counterweight effectiveness high.

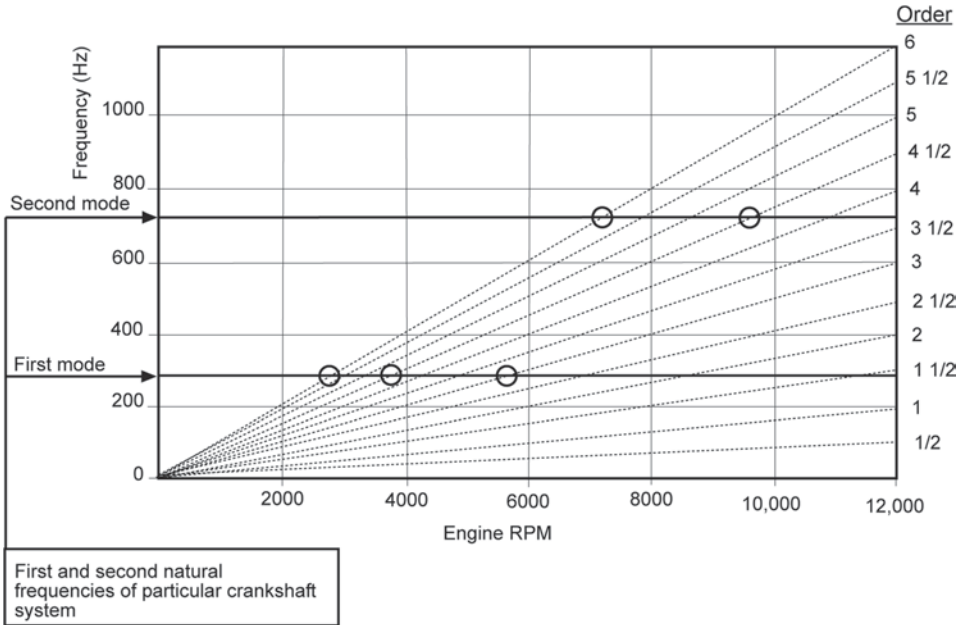


Fig. 16.23 Identification on Campbell plot of resonant speeds for a particular crankshaft system

For the reasons discussed at the beginning of this section damping is applied to reduce the torsional vibration amplitude in most automotive engines. Various damping techniques can be used but the most common are the untuned viscous shear damper and the tuned harmonic damper.

A cross-section of a typical viscous shear damper is depicted in Fig. 16.24 along with its effect on crankshaft deflection. The viscous shear damper consists of a ring of mass encased in a cavity filled with a silicon-based fluid. The casing is rigidly bolted to the crankshaft nose, and the mass is free to float in the viscous fluid. As the crankshaft nose experiences a vibration impulse the mass reacts in the opposite direction, creating a shear force in the fluid that damps the vibration impulse. The vibration energy is dissipated as heat energy from the viscous shear. This type of damper is referred to as untuned because it will damp the vibration pulses regardless of frequency, as can be seen in the plot of Fig. 16.24. The unit must be sized large enough to dissipate the vibration energy without overheating, and adequate air flow around the damper must be ensured. This type of damper is relatively expensive so is generally seen only on larger, higher cost engines. Because the running clearance is quite small it is susceptible to damage.

More commonly used in automobiles is the tuned harmonic damper shown in Fig. 16.25. In this design the damper hub is again rigidly mounted to the crankshaft nose, and a seismic mass is then mounted around the hub through a rubber isolator. The combination of

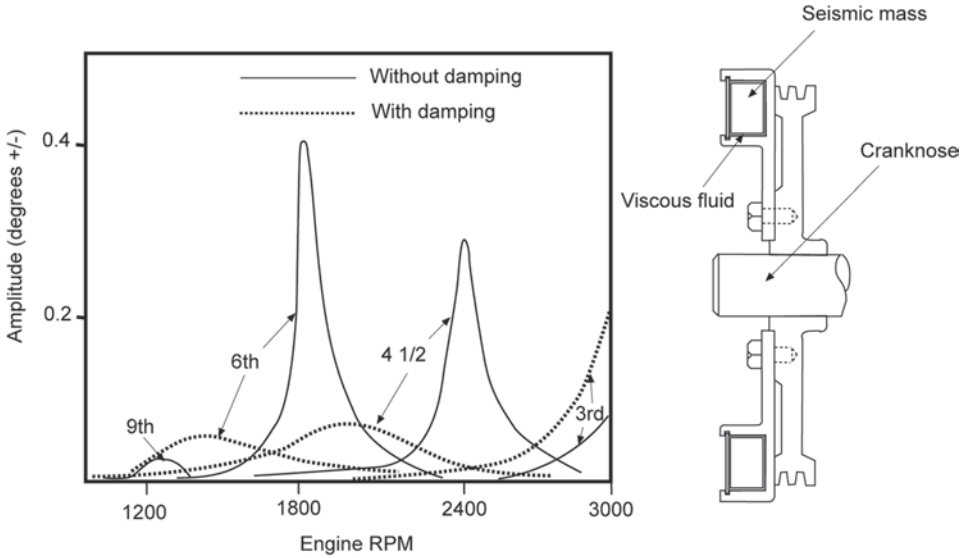


Fig. 16.24 Untuned viscous shear damper operation

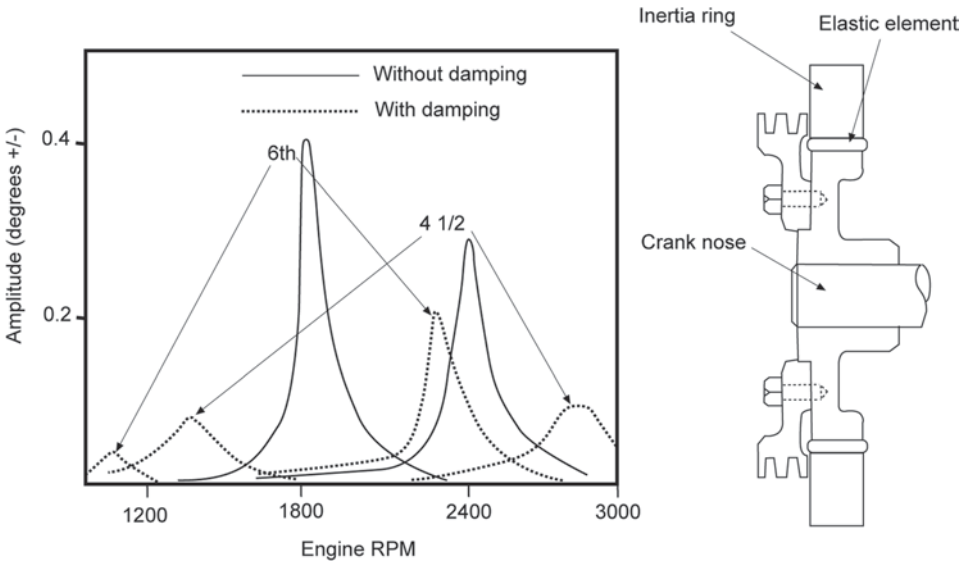


Fig. 16.25 Tuned damper operation

the mass and the stiffness of the rubber isolator is tuned to damp out vibration at a particular frequency, as shown in the figure. It should be noted that each order now has two resonant frequencies because the coupled mass acts as a second torsional system linked through the rubber isolator.

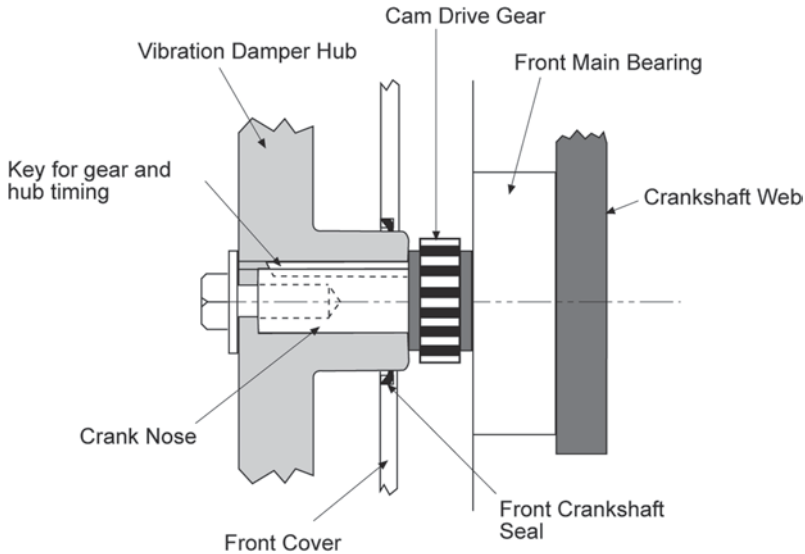


Fig. 16.26 Crankshaft nose design and assembly details

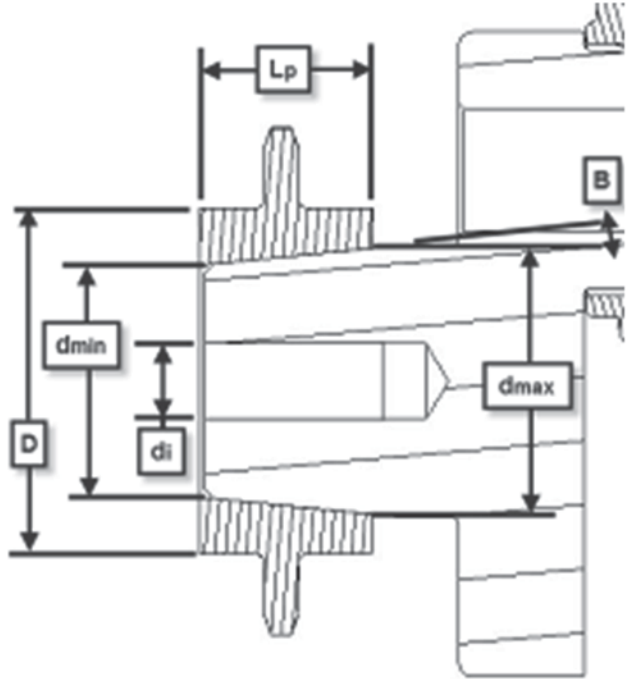
16.5 Crankshaft Nose Development (Straight, Taper, Spline Fit)

A cross-section showing an example of the crankshaft nose and the front details of the engine are shown in Fig. 16.26. A cam drive gear is fit onto the front of the crankshaft and located using a keyway or spline. From this gear the camshaft will be driven using a gear, chain, or belt drive, as discussed further in Chap. 17. This or an adjacent gear may also be used to drive the oil pump and other internal engine drives (balancer shafts or a fuel pump are examples). The crankshaft nose then protrudes through the front cover, and the vibration damper and accessory drive pulleys are mounted to the nose. The front oil seal is mounted to the front cover, and either directly contacts the crankshaft surface inboard of the damper, or contacts the damper hub as shown in the figure.

There are three drive options for accessories at the front of the crankshaft: straight fit, taper fit, or spline. The torque capacity of each these joints needs to be calculated to ensure a successful design. The straight fit and taper fit configurations will need a keyway, not to handle drive torque, but to ensure timing of the accessories to the crankshaft. The torque applied to the clamping fastener(s) provides normal force and friction in the joint, which provides the torque capacity.

The objective of the taper fit joint is again to provide sufficient fastener clamp load (F_{clamp}) to withstand torsional loading on the crankshaft. However, the taper design is benefited by the interference fit generated between shaft and hub. These equations are developed from those used for a thick wall pressure vessel, and the variables are defined in Fig. 16.27. This method assumes that the hub does not bottom against a shoulder, and

Fig. 16.27 Taper fit variables



is only supported by the taper. It also assumes that the clamping fastener does not bottom on the end surface of the shaft, to ensure that all fastener clamp force is going into the hub.

Using the assumed starting value for interference fit the contact pressure is calculated as shown in Eq. 16.5.

$$P = \frac{i}{\frac{d}{E_o} \left(\frac{1 + ce^2}{1 - ce^2} + \nu_o \right) + \frac{d}{E_s} \left(\frac{1 + ci^2}{1 - ci^2} - \nu_s \right)} \tag{16.5}$$

$$d = \frac{d_{\min} + d_{\max}}{2} \quad ce = \frac{d}{D} \quad ci = \frac{d_i}{d}$$

Where:

- P = Contact Pressure
- i = Assumed interference fit
- d = Mean Taper diameter
- E_o = Young's Modulus of outer material (hub)
- E_i = Young's Modulus of inner material (shaft)
- ν_o = Poisson's Ratio for outer material
- ν_i = Poisson's Ratio for inner material

c_e = Major diameter ratio
 c_i = Minor diameter ratio
 D = Outside hub diameter
 d_i = Inner hole diameter

Maximum introduction and extraction forces are calculated using Eqs. 16.6 and 16.7 respectively. Maximum torque retention and required fastener torque are then calculated as indicated.

$$F_i = \pi \cdot d \cdot L_p \cdot P \cdot (f + \tan(\beta)) \quad (16.6)$$

Where:

L_p = Length of taper fit area
 f = Coefficient of friction between components
 β = Half of taper included angle

$$F_e = \pi \cdot d \cdot L_p \cdot P \cdot (f + \tan(\beta)) \quad (16.7)$$

$$T_{\max} = F_i \cdot \frac{d}{2} \quad (16.8)$$

$$T_f = k \cdot F_i \cdot d_f \quad (16.9)$$

Where:

k = Nut factor, typically between 0.18-0.22 for steel
 d_f = Fastener major diameter

A splined joint can transmit more torque for its size than other types of joints as shown in Figs. 16.28 and 16.29, and can be used at the nose of the crankshaft to drive accessory loads, or more commonly used at the power take off end of the crankshaft to drive the vehicle. The objective of a spline is to drive using the teeth on the joint; it does not rely primarily on clampload. Several failure modes must be designed for: base shaft breakage, hub bursting, teeth of spline shearing off at the pitch line, teeth of the internal spline breaking at root due to bending stress, and wear on the drive surface of the flank of the spline. Splines can be either “fixed”, where there is no relative or rocking motion between the internal and external teeth as in a clamped joint, or “flexible” where there is relative rocking or axial motion.

Fig. 16.28 Splined shaft**Fig. 16.29** Splined hub (*gear*)

The crankshaft nose is subject to high-cycle fatigue loading. The combination of the dampener and pulley mass and accessory drive belts create a resultant force of constant magnitude and direction. As the crankshaft spins each location on its nose experiences a complete bending load cycle once every revolution of the shaft. The outside surface of the nose experiences alternating tensile and compressive loads, and the threaded mounting hole(s) for the dampener assembly results in local stress concentrations, and the threads should be counterbored to mitigate. Rig testing can be readily devised to duplicate this load cycle and provide durability validation.

16.6 Crankshaft Flange Development

The same rationale just discussed for crankshaft nose loading and durability applies to the flywheel mounting flange as well. The flywheel or torque converter flex plate are typically mounted to the crankshaft flange using a multiple bolt pattern. The flywheel connection relies on friction to generate the shear torque capacity in the joint, but a frictionless condition must also be designed for. The steps in calculating the frictional torque capacity of the bolted joint are as follows:

1. Approximate nominal clampload per fastener
2. Calculate the shear capacity produced per fastener
3. Determine the total frictional torque capacity in the joint

In order to increase the frictional torque capacity of the joint, it may be difficult to change the fastener pitch circle radius or fastener diameter once a design is in production. If it is desired to increase the fastener pitch circle diameter, the main journal diameter may need to increase. If the main journal diameter increases, it will increase the bearing diameter, seal diameter, cylinder block support web, and oil pan rail width, depending on design of the crankshaft.

It is a frequent occurrence that the marketing organization will request an increase in engine torque or power shortly after a new design is released to production. This is one key area of engine design to package protect for future increases in output.

If a design is complete and in production, and more torque capacity is required of the joint, various treatments can be added to increase the capacity of the joint at an additional cost. The coefficient of friction at the joint surface can be increased by changing the machined surface finish, by adding abrasive coatings, or by adding additional locating dowels. Alternately, the fastener grade may be increased to allow greater clampload.

Two examples of the rear crankshaft oil seal details are shown in Fig. 16.30. The case on the left incorporates a split seal in the rear main bearing. That on the right uses a continuous seal riding on the flywheel flange.

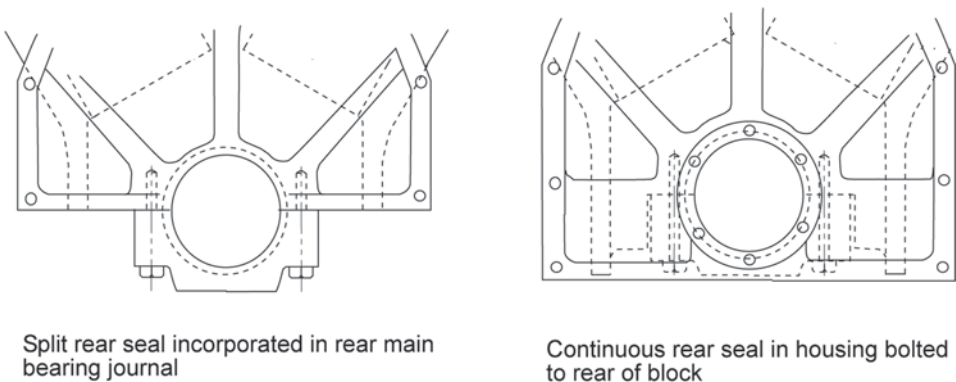


Fig. 16.30 Rear crankshaft seal designs: split seal with short skirt block, and continuous seal with extended skirt block

16.7 Crankshaft Drillings

Crankshafts are drilled for several reasons: material removal for balancing, material removal for weight reduction, and pressurized oil feed. Crankshaft drilling for balance is usually done in the counterweight webs as shown in Fig. 16.31. This is done when dimensions of the forging or casting of the base crankshaft cannot be held close enough to achieve the desired balance, or when a lighter piston is introduced that changes the required balance. This area is usually not significantly stressed, and so drilling does not present a significant stress concentration. These drillings can either be radial, as shown, to remove mass or can occasionally be axial in the same location to allow addition of more dense material such as tungsten alloy, depleted uranium, or lead. The addition of dense balance weights is more expensive, but can make the overall crankshaft lighter through a more efficient use of material, or can be used to reduce crankshaft inertia for racing applications.

Crankshaft drilling for weight reduction is typically done in the crankpin. This drilling of the crankpin reduces the amount of counterweight material required and can make the entire crankshaft lighter. It can also change the local stiffness of the crankpin, to better distribute stress in the crankshaft and reduce it in the critical journal fillet region. The crankpin is typically a highly loaded area, so attention must be paid to detailed design of the cross drilling in this area, and clearance in adjacent counterweights is typically required to allow access for this drilling. Frequently, this cross drilling is not coaxial to the crankpin.

If the crankshaft has plain bearings, it will be necessary to provide pressure lubrication via internal drilling of the crankshaft. These long drillings typically connect the main journal to the crankpin to provide pressurized lubrication and these internal drilled passages come very close to the highly loaded crankpin and main journal fillets as illustrated in Figs. 1.6 and 1.8. If the drilled passage comes close to the surface in a highly loaded area, either at a journal fillet or in the crank web, a fatigue crack may initiate. Additionally, care must be taken at the location where the drilled hole breaks the surface of the journal as this may also create a fatigue crack initiation point. The drilled oil hole is typically chamfered

Fig. 16.31 Crankshaft drilling for balance, oil passages



where it contacts the journal surface to diffuse stress, and also to spread pressurized oil to a larger area under the bearing. One technique to improve fatigue resistance is to use a hardened ball, andpeen the ID of the chamfer where it meets the cross-drilled hole. This will introduce residual compressive stresses in this area and reduce a crack initiation site. Finally, the location where the oil hole breaks the surface of the crankpin is important for the bearings. It should be located somewhere between 90 and 30° before TDC on the compression stroke to provide the best feed of oil to the bearing. This is the point when the load on the bearing is least. The critical pressure region of 0–45° after TDC on the power stroke must be avoided. High rod loading increases oil pressure above the system pressure and blocks the feed to the bearings.

16.8 Connecting Rod Development

16.8.1 Connecting Rod Column Forces

The connecting rod is among the highest stressed components in the engine, and load is applied several times on every revolution of the engine. The connecting rod must be designed for high cycle fatigue to withstand the high number of engine cycles, and also for stiffness in supporting fluid film bearings. The connecting rod transfers the gas and inertia loads from the piston to the crankshaft, and experiences high rates of loading and direction reversal, high temperatures, and varying degrees of lubrication. Axial inertia and connecting rod whip forces (tangential to crank radius) are a function of connecting rod weight, so the higher the weight, the higher the forces. Since inertia can be a dominant load, and the piston, piston pin, and connecting rod all contribute, a more durable connecting rod design might require less material (be lighter) instead of the typical approach of adding more material to increase life. The higher the piston and rod weight, the more crankshaft counterweight is required, which leads to a heavier engine. As a result, rod weight is of primary importance. Value added operations such as shot peening to enable a lighter connecting rod are often justified to reduce forces and weight in the rest of the powertrain. Piston normal forces due to connecting rod angle and connecting rod whip forces are translated to the engine, and require stronger support in the chassis. However, when a connecting rod fails, it typically destroys the entire engine as would a crankshaft failure. The desire for minimum weight must be balanced with rod durability, and the desire for the lowest cost manufacturing methods. Connecting rods must be cost effective, manufacturable, and serviceable.

The engine type has a significant effect on the peak loads seen by the connecting rod. At one extreme is the low engine speed and high BMEP diesel, and at the other end is the high engine speed and lower BMEP high performance gasoline engine. Gas loading and inertial loading at TDC oppose each other, reducing total loading on the connecting rod on the power stroke. At lower engine speeds, the gas forces dominate and put the connecting

rod into compression. At higher engine speeds, inertial loads may dominate and stress the rod in both tension and compression.

Connecting rod loads vary as a function of crankshaft angle, and also as a function of engine cycle. At TDC-exhaust (overlap) on a 4-stroke engine, the connecting rod experiences the highest tension load case because there is very little gas pressure force to resist it. As opposed to TDC-power stroke, when there will be a combination of tensile forces from inertia, and compressive forces from combustion. At BDC-exhaust, the inertia forces put the rod into compression.

The maximum tensile forces on the connecting rod occur at TDC-Exhaust as illustrated in Eq. 16.10. Gas pressure forces can be neglected since they are near zero during valve overlap.

$$F_{recip, total} = -(m_{piston} + m_{conrod, recip}) \cdot r \cdot \omega^2 \cdot (\cos \theta + \lambda \cos 2\theta) \quad \left(\lambda = \frac{r}{l} \right) \quad (16.10)$$

The maximum compressive forces on the connecting rod occur either at TDC-Power stroke for a low speed high BMEP engine, or at BDC-Exhaust for a high speed low BMEP engine as shown in Eq. 16.11.

$$F_{recip, total} = -(m_{piston} + m_{conrod, recip}) \cdot r \cdot \omega^2 \cdot (\cos \theta + \lambda \cos 2\theta) - F_{Gas} \quad (16.11)$$

Notice that these equations are a function of the connecting rod ratio (λ), with typical values between 0.2 and 0.35. As the connecting rod grows compared to the engine stroke, it has the tendency to reduce the peak force on the rod, and also the peak piston thrust force. However, a longer connecting rod has more weight and increases the engine deck height. These conflicting requirements must be balanced.

The main beam of the connecting rod is subject to inertial bending forces (rod whip) as it swings through TDC. For an initial analysis, it is assumed that the connecting rod is a simply supported beam, subject to a linearly varying distributed load as shown in Fig. 16.32. Once the connecting rod geometry is complete, finite element analysis can be performed and the load can be spread to all of the elements of the beam in proportion to their distance from the piston pin end.

The total bending force due to the triangular distribution is calculated in Eq. 16.12, and the bending moment at any distance 'x' from the piston pin centerline is calculated in Eq. 16.13.

$$F_{beam} = \frac{2}{3} \cdot m_{rod, upper} \cdot r \cdot \omega^2 \quad (16.12)$$

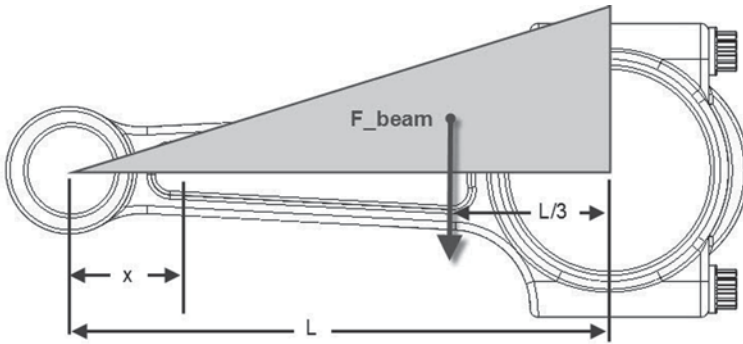


Fig. 16.32 Simply supported beam with uniform increasing load

Where:

$m_{\text{rod, upper}}$ = Mass of connecting rod between two pins

$$M = F_{\text{beam}} \cdot \frac{x}{3} \cdot \left(1 - \frac{x^2}{L^2}\right) \quad (16.13)$$

Since gas pressure loading dominates at low engine speed, the rod must be designed for compression. There is also a high compressive load at BDC due to inertia at high engine speed. In addition to designing for compressive stress, column buckling must also be considered and in high BMEP Diesel engines this is a very important consideration. The rod can buckle in one of two directions, parallel to the plane of connecting rod motion, and perpendicular to it.

The end constraint on the connecting rod significantly influences its buckling resistance. If the rod is pinned at either end the least resistance to buckling is present, and if the rod is fixed in 6° of freedom at either end the most resistance to buckling is present. Depending on the exact end constraint, the column is said to have an equivalent length (L_e) to a column pinned at both ends. Column buckling is also sensitive to the geometry of the cross-section, and typically requires three-dimensional analysis.

16.8.2 Connecting Rod Crankpin Bore Cylindricity

The crankpin end of the connecting rod is primarily designed for stiffness, as the connecting rod needs to support the crankpin journal bearing and maintain roundness. As the engine progresses through its cycle, tensile inertia forces attempt to stretch the rod and deform the crankpin end of the connecting rod. Stretching of the connecting rod along the cylinder bore causes the connecting rod to pinch in along the cap split due to the Poisson effect, which may lead to the oil film breaking down and the bearing contacting the crank-

shaft. If rolling element bearings are used, pinching of the rolling elements may occur leading to skidding or spalling.

Since downward compressive forces load the upper half of the bearing, and not the bottom half, only tensile forces are considered. Now, both the upper half and lower half of the rod mass are counted for inertial calculations. At TDC exhaust, the entire rod/piston system is creating a force away from the crank equal to the reciprocating force and the rotating force from the lower end of the rod as shown in Eq. 16.14. This force is reacted by the lower end of the connecting rod. In high speed engines, this can be the limiting load on the connecting rod because of ovalization of the crankpin bore, bolt limits, or cap strength.

$$F_{TDC-Exh} = -(m_{rot} + m_{recip}) \cdot r \cdot \omega^2 (1 + \lambda) \quad (16.14)$$

Where:

m_{rot} = Rotating mass of connecting rod

m_{recip} = Reciprocating mass of connecting rod and piston assembly

In order to achieve a true deflected shape, the bearing press fit and support from the fluid film or rolling element bearing must be incorporated in this analysis. It is important to work with the bearing manufacturer to establish limits of cylindricity.

The outermost surface of the split line has the largest stress. This is because as the rod cap is loaded in tension, the bolt flange will rotate about this outmost line of contact. As this line of contact is moved further away from the crankpin bore, the lever arm increases on the flange, and the unit load decreases. For this reason, most connecting rods have additional material added outboard of this flange to decrease unit loading and increase flange stiffness. Ideally the connecting rod bolt would be as close to the bore as possible, to reduce this flange rotation due to cantilevering.

16.8.3 Connecting Rod-to-Cap Alignment

Connecting rod-to-cap alignment is critical to a successful rod design, and ensuring crankpin journal bearing life. Angular split lines present more of a challenge because of the direction of the applied forces in relation to the split line, but are sometimes necessary. Some alignment methods are:

- *Increased* fastener shank diameter at split line, to provide alignment
- Ring dowels, coaxial with the fastener in a counter-bore
- Separate dowel pins next to the connecting rod bolt.
- Specially machined serration at the rod-to-cap split surface.
- Stepped surface, specially machined rod-to-cap split surface.
- Cracked, manufactured by cracking the end of the rod off by force at the intended parting plane.

Each method has pros and cons, which is why there are so many methods in use. The increased shank fastener is inexpensive to manufacture and is compact in design, however if the alignment deviates from ideal there may be interference between the bolt and rod at the split line. This may add bending stress to the bolt, or upset material in the rod into the bearing bore and degrade circularity. Ring dowels are very similar in performance, with the added disadvantage of moving the rod bolt further away from the bearing bore. For packaging reasons, this may increase the overall rod size. Separate dowel pins are inexpensive, but may also increase the size of the connecting rod and add weight.

A serrated or stepped surface between the connecting rod and cap is more expensive to manufacture due to the increased sensitivity to manufacturing tolerances, but this type increases the shear resistance in the rod-to-cap joint. Stepped surface rod caps operate in a similar manner, using a lip to limit cap motion. Angle split rods increase the amount of shear force on the joint, and often use one of these types of alignment.

The cracked design is now widely used in high volume production, in both forged and powdered metal rods. This method is cheap, repeatable, and insensitive to tolerances. Alignment and shear resistance are excellent. There is very low distortion of the split surface, since both surfaces are symmetrical. However this method dictates the parent material of the connecting rod, and may limit the choice of manufacturer since high production volumes are typically needed to justify the expense of special equipment to form and crack the rod.

16.8.4 Connecting Rod Bushing Press Fit and Journal Bearing Crush

The small end, or pin end of the connecting rod is almost invariably a continuous bearing surface—no separation plane is needed as the piston pin can be inserted through the connecting rod from either side. Two designs are commonly seen for retaining the piston pin. One is the fixed pin, where the pin is pressed into the connecting rod bore, and rotates freely in the piston pin bore. The other is the floating pin, where the piston pin is free to rotate in both the piston and connecting rod bores, and is held in place with snap rings fitted in grooves on both outer edges of the piston pin bore. The fixed pin reduces cost and weight but has lower load carrying capability and requires special assembly provisions. The degree of required press fit generally requires the connecting rod to be heated. Because the connecting rod does not provide an acceptable bearing surface the floating pin requires the addition of a bushing to the connecting rod. This and the required snap rings and machined grooves significantly increases cost relative to the fixed pin. Traditionally the floating pin was seen only in high performance and heavy-duty engines, but due to the demand for increased loads it is now being adopted in many engines.

Interference fits are used to retain the piston pin bushing, as well as the crankpin journal bearing. Both are based on the same fundamentals of thick walled pressure vessels, but their application and assembly method are different. This requires different approaches to design. The thick wall pressure vessel equations are valid as long as the wall thickness-to-

Table 16.2 Press conditions to evaluate

		Temperature		
		Maximum temperature	Operating temperature	Minimum temperature
Tolerance (fit)	Maximum interference	X		Y
	Nominal interference		X-Y	
	Minimum interference	Y		X

radius ratio is greater than 0.1. Bearing press fits must be designed for operating loads, but also extremes of temperature. The goal is to have sufficient press fit to retain the bearing in its bore under all operating conditions as presented in Table 16.2, while not exceeding the stress limits for the given materials.

Depending on the material used for the housing and the material used for the bushing, differential thermal expansion may help or hurt the design. For a low thermal expansion coefficient housing (steel) and a high thermal expansion coefficient bushing (aluminum), the “X’s” represent the extreme conditions to evaluate. For a high thermal expansion coefficient housing (aluminum) and a low thermal expansion coefficient bushing (steel), the “Y’s” represent the extreme conditions to evaluate.

The final hoop stress on the crankpin end of the rod is developed in much the same way as the press fit bushing example above, except the bearing is split into two different pieces for installation in the upper rod and cap. The diameter of the journal bearing is slightly larger than the bore in the connecting rod (or main bearing), and is compressed during installation. Since the bearing is neither uniform in thickness, or constant in radius, a direct measurement of the diameter is misleading. A different method of measurement is required to describe the press fit. This is called crush height and is measured by the overstand test, the amount of which affects the press fit. The overstand test consists of placing the bearing shell in a half circle gauge, flush at one end and extending above the other. The amount the shell protrudes from the gauge is the amount of circumferential crush the bearing will be under when assembled.

The crush height is measured under a gauge force. This is effectively measuring the circumference or length of the journal bearing shell, rather than diameter. A bearing with sufficient crush height ensures that the bearing is preloaded enough to have uniform contact with the housing. This ensures good heat transfer from the bearing to the housing, and prevents fretting. Typical values of the crush height of automotive bearings are 0.05–0.10 mm.

16.8.5 Connecting Rod Computational Stress Analysis

Typically, the entire connecting rod is not analyzed. A half model with symmetry, or even a quarter model, is used. The mesh is typically refined around the bolted joint area, and around critical fillets and transitions. The connecting rod is analyzed in the following steps.

The assembly load cases are determined by applying the bolt preload to the connecting rod cap. Deformations are incorporated into the initial geometry of the model to simulate machining of the crankpin bore when clamped together. Bushing and bearing assembly load cases are applied by simulating the interference due to press fit of the piston pin bushing and crankpin journal bearings.

The dynamic load cases produced by the engine are analyzed by applying maximum tensile load at TDC-Exhaust stroke and maximum compressive load at BDC-Power stroke. The piston pin and crankpin are modeled as separate cylinders and used as boundary conditions since they contribute to the overall system stiffness. Finally, the loading due to rod whip is analyzed.

The final step of the analysis is to evaluate the connecting rod for critical column buckling in the plane of connecting rod motion, and perpendicular to it. This enables a more refined solution to the analytical method, since detailed geometry is represented.

16.9 Flywheel Design Considerations

The purpose of the engine flywheel is to smooth the cyclical speed variation of the engine within a given engine revolution. It absorbs energy during the power stroke, and distributes it during the exhaust, intake, and compression strokes. The transmission of instantaneous torque spikes developed by the engine will be reduced to the driveline.

The flywheel or torque converter mass is important in controlling idle speed fluctuation within limits acceptable to the driver. As speed increases fluctuations become much less apparent and the need for this mass decreases. An acceptable idle speed fluctuation limit is identified and flywheel mass is chosen to be sufficient to reduce fluctuation within that limit. Additional mass beyond the minimum required will penalize the engine's response to transient requirements.

This storage and release of energy has other uses in the engine and the vehicle, when the engine average speed changes. Large flywheel inertia will enable a low engine idle speed. A low engine idle speed, by reducing the total number of revolutions the engine makes at idle, will reduce fuel consumption and improve emissions during an engine operating regime that does little useful work. The lower the speed variation, the easier it is to calibrate the fuel injection at low engine speeds as the piston approach to TDC is more consistent.

Like in heavy machinery, the flywheel will allow sudden loads to be placed on the engine without stalling it. This is helpful when starting a vehicle from rest, or during sudden load changes. The inertia of the flywheel can be balanced against the inertia of the vehicle. A large engine flywheel will also make the engine less responsive during transient drive modes, as the engine will not accelerate quickly with the application of throttle and will not decelerate quickly during coasting/sail or removal of the throttle. This may have a beneficial effect in a work truck or for cruising on the highway at steady speeds, but may not be desirable in a sports car where quick engine response is desired.

In many cases the flywheel and crankshaft are balanced as a unit, and a non-symmetric bolt circle or locating dowel is used to ensure that if the flywheel is removed it is again mounted in the same position. The clutch disc and pressure plate are then mounted to the flywheel with a series of cap screws around the perimeter of the pressure plate. A pilot bearing or bushing mounted in the rear of the crankshaft at the shaft centerline supports the nose of the transmission input shaft. In transverse installations this shaft may not feed directly into the transmission, but drives a chain that then transfers load to the transmission.

In the case of an automatic transmission a flex plate, bolted to the same bolt circle on the crankshaft flange, replaces the flywheel. The torque converter is then bolted to the flex plate with three or four bolts near its perimeter.

Equation 16.15 for kinetic energy stored in a flywheel is shown below, and is equivalent to the amount of energy released if the flywheel speed is changed from its current rotational speed to a full stop:

$$E = \frac{I\omega^2}{2} \quad \text{or} \quad \frac{Wv^2}{2g} \quad (16.15)$$

Where:

E = Energy stored in the flywheel

I = Polar moment of inertia

ω = Rotational velocity

W = Weight of flywheel rim

v = Linear velocity at mean radius

A flywheel is designed to reduce the speed fluctuation to a desired amount. This is measured in percent change in speed called the Coefficient of Fluctuation (C_f) as represented in Eq. 16.16, or in absolute speed change. If a lower speed fluctuation is desired, a larger flywheel will be required. However, the larger the flywheel, the more difficult to package and the heavier the engine will be. Since the flywheel is usually directly coupled to the crankshaft, it is not possible to spin the flywheel faster than engine speed, removing this as a design option. To gain smoothness, the flywheel diameter or thickness will need to

be increased. A thin disc of large diameter has the most efficient use of material, and a hub with most material concentrated near the rim would be even better. Typically dense materials such as steel or cast iron are used for flywheels.

$$C_f = \left| \frac{\dot{u}_2 - \dot{u}_1}{\dot{u}_{mean}} \right| \quad (16.16)$$

If the velocity of a flywheel changes, the energy it absorbs or discharges will be proportional to the difference of initial and final speeds per Eq. 16.17:

$$E = \frac{I(\omega_2^2 - \omega_1^2)}{2} \quad \text{or} \quad \frac{W(v_2^2 - v_1^2)}{2g} \quad (16.17)$$

The traditional method for calculating the required amount of energy in the flywheel is to determine the peak energy needed by the work operation and subtract the average energy provided by the engine or motor providing energy input. The difference is the flywheel energy needed to maintain speed as represented in Eq. 16.18. This method is relevant to the vehicle side, as a load is applied to the vehicle, the required flywheel inertia to maintain engine speed can be calculated.

$$E_{work} - E_{motor} = E_{Flywheel} \quad (16.18)$$

Once the desired flywheel energy storage is determined, and the desired speed fluctuation is set, the equation can be rearranged to determine inertia required utilizing Eq. 16.19:

$$I = \frac{2E}{(\omega_2^2 - \omega_1^2)} \quad (16.19)$$

16.10 Crankshaft and Connecting Rod Construction

Crankshaft steels are typically medium carbon (0.3–0.4%) and heat treated to increase the tensile strength. Forged crankshafts have inherently higher material strength, especially in critical regions through close control of grain flow. Cast iron alloys vary over a wide range in tensile strength depending on material and heat treat; their values can be found in many references. Casting a crankshaft can enable other desirable geometry for little cost, such as hollow crankpins and detailed shaping of the counterweight webs. The development of the metallurgy and casting methods has allowed the use of cast crankshafts in more and more applications. Where packaging dimensions are not strictly constrained, cast crankshafts can be a reasonable alternative to forged steel shafts.

As discussed when covering bearing design, an important variable in determining the minimum oil film thickness for hydrodynamic operation of plain bearings is that of shaft surface finish. As the finish is improved unit loading can be increased, and lower viscosity lubricants can be used to improve fuel efficiency. In order to achieve the required main and rod bearing surface finishes the machining is done in three steps. Lathe cutting is followed by grinding, and finally by lapping or polishing.

In high output applications steps may be taken to increase crankshaft strength in the critical fillet transitions. These value-added operations are ion nitriding, shot peening, induction hardening, and fillet rolling or roll hardening. These operations locally increase the strength of the crankshaft by modifying the material properties or introducing compressive residual stresses. These additional operations cost additional money, and careful design can often avoid their requirement. However, the designer must compare the added manufacturing costs to the larger implications to the engine and vehicle downstream. In order to increase the power output of the engine, these processes can be added without having to redesign the engine. Enabling smaller journal diameters can reduce engine friction. These operations can be beneficial when considering the system as a whole.

Historically connecting rods have been cast, forged, or machined from billet. Recently, connecting rods have been manufactured from forged powdered metal for high volume automotive applications. This process can be lower cost, enabling more consistent parts and a beneficial rod cap alignment surface. Heavy-duty engines still primarily use forgings.

The materials used are cast iron, steel, aluminum, and titanium. Cast iron is typically used in cost sensitive applications, with forged steel being used in high load applications. Aluminum connecting rods are being used in light-duty engine applications, and recent improvements in metallurgy have increased the suitability of this material. Recall from earlier discussion that the connecting rods own weight acts against itself in high speed applications, so lighter materials enable higher engine speeds. Titanium is typically reserved for racing applications due to its high cost.

Connecting rods proceed from casting or forging, to rough machining of the upper rod and rod cap separately—or as a single piece if the cracked rod cap is used. The two components are then bolted together for the finish machining of the assembly and precision honing of the bores. It is important to perform this precision machining in the assembled state, as bolt clamp loads will distort the big end bearing shape, and cylindricity is key to fluid film bearing life. Occasionally, the piston pin bushing and crankpin journal bearing are assembled to the rod, and an additional boring and honing operation is performed if a higher class of tolerance is to be maintained.

Frequently, value added operations are applied to the connecting rod to increase component life, or reduce component weight. Trimming of a casting or forged connecting rod blank will often leave score marks in the finished part. These score marks can act as stress risers in the key loaded areas of the connecting rod. They may be ground or polished smooth to reduce the stress concentrations. Additionally, shot peening of the connecting rod will introduce compressive residual stresses in the component. While these operations add cost, the designer must balance this against the potential added benefit to the engine and vehicle.

16.11 Analysis and Test

The typical analysis path is to evaluate each component individually, and then once a level of maturity is reached, combine the individual components into a system model. The crankshaft analysis typically starts with analysis of a simply supported, single throw. Once that is complete, a full crankshaft model is analyzed. Often a coarse mesh model of the crankshaft is analyzed to determine system natural frequencies for torsional calculations. Then the crankshaft, connecting rod, and cylinder block are combined in a system analysis. This is typically more complex, as the fluid film must be modeled as gap elements. Some system stiffness is contributed by the crankshaft, and some by the cylinder block.

There are two opposing design philosophies for crankshaft-cylinder block design, one of cylinder block guided stiffness, and another of crankshaft guided stiffness. Each of the different methods has one component significantly stiffer than the other. Regardless of the method chosen, the goal is to have sympathetic deflection in the same direction between the housing and the shaft, to prevent edge loading of the bearings.

If a crankshaft or connecting rod fails during engine testing, the results usually include extensive progressive damage. If either component fails many other engine components are damaged beyond repair. For this reason, crankshafts and connecting rods are typically subjected to component tests to prove their reliability, prior to exposing an expensive prototype engine to destruction.

Various servo-hydraulic rigs are used for component level testing. The crankshaft is usually evaluated for torsional strength and bending resistance at the crankpin. For torsional testing, it is usually supported at the main bearings, fixed at the power take off, and a firing pressure load applied at the crankpin. The bending load at the crankpin can either be simulated by fixing the crank near TDC and loading via a driven connecting rod, or can be subjected to a moment by bending the crankshaft from one main bearing to another as represented in Fig. 16.34. A single throw section is rigidly mounted in a heavy fixture that is then suspended as shown. By making the fixture mass large it will resonate at a very high natural frequency. Once this frequency is identified high fillet stress can be achieved with a very low driving force. Strain gauges are added in the fillet region to drive the testing apparatus, or to assess the impact and sensitivity of different fillet radii under a given load. As shown in Fig. 16.35 the location of maximum alternating stress along the fillet is identified. By varying the magnitude of loading at the resonant frequency an ‘S-N’ diagram can be generated specific to the crankshaft section. The cycle accumulation rate is so high that the number of cycles the crankshaft would accumulate over its entire life in an engine can be accumulated in a matter of hours. The ‘S-N’ diagram results can be compared with the actual stresses seen by the crankshaft in the engine to determine the actual fatigue life. The rig test method described here is so easily done that it can be used for production quality checking as well. Production samples can be tested and overlaid on the ‘S-N’ diagram to quickly identify shifts that might occur due to casting or forging problems, material alloy changes, or grinding, hardening, shot peening, or cold rolling process control problems.

Fig. 16.33 Connecting rod column bending

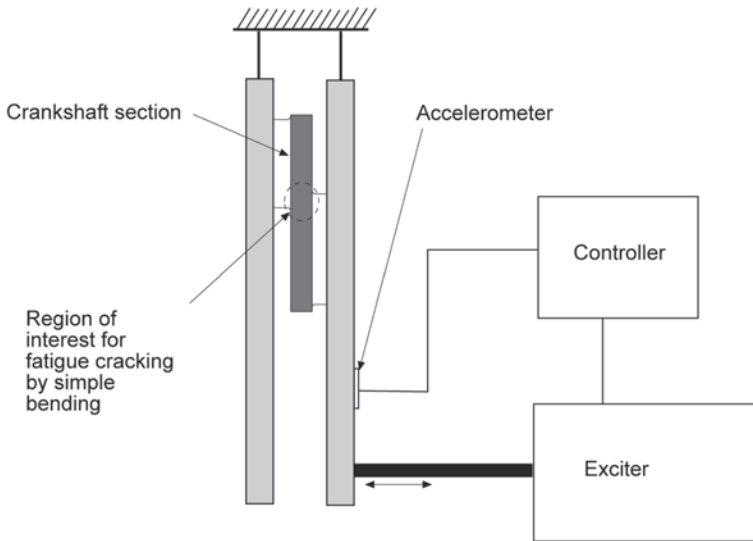
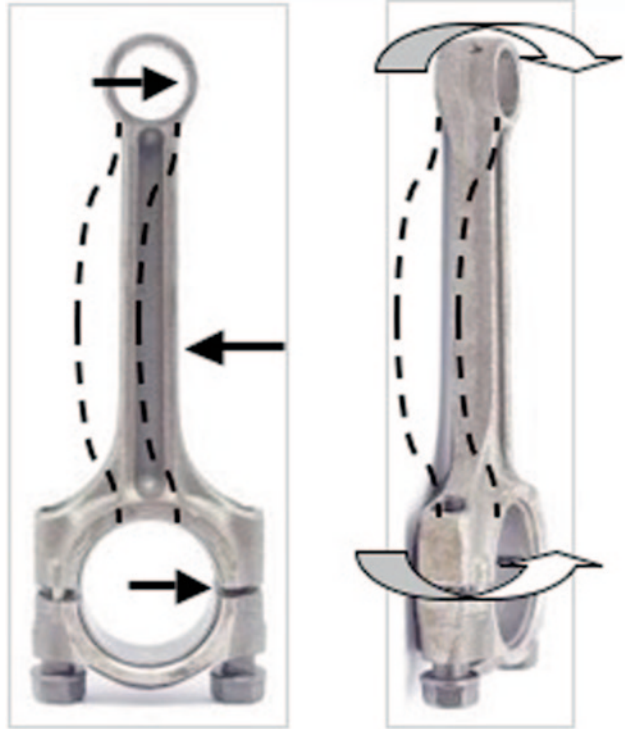


Fig. 16.34 Rig test for simple bending across a crankshaft web

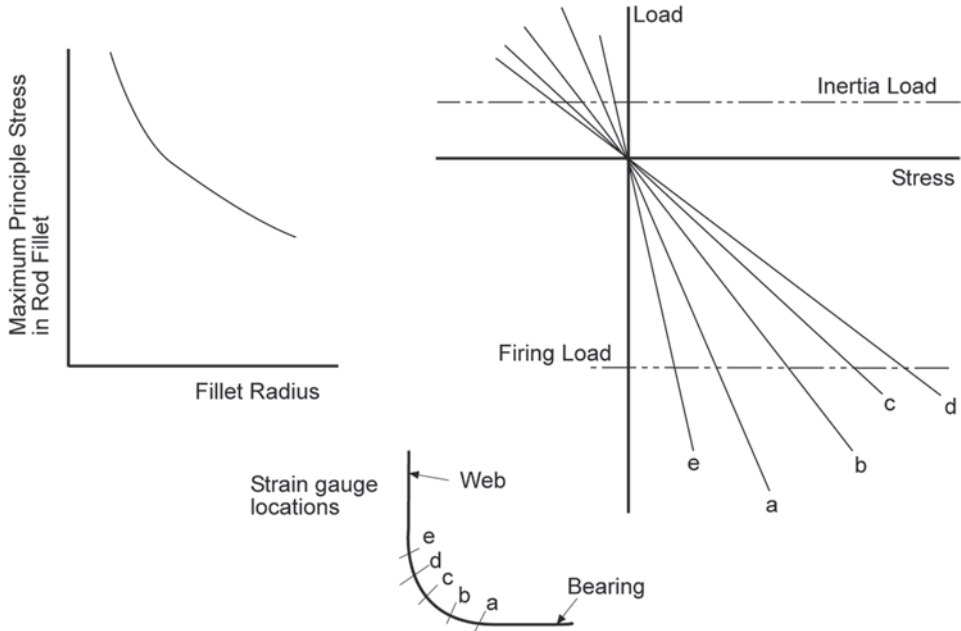


Fig. 16.35 Role of fillet radius in control of principle stress at bearing journal; identification of maximum stress location

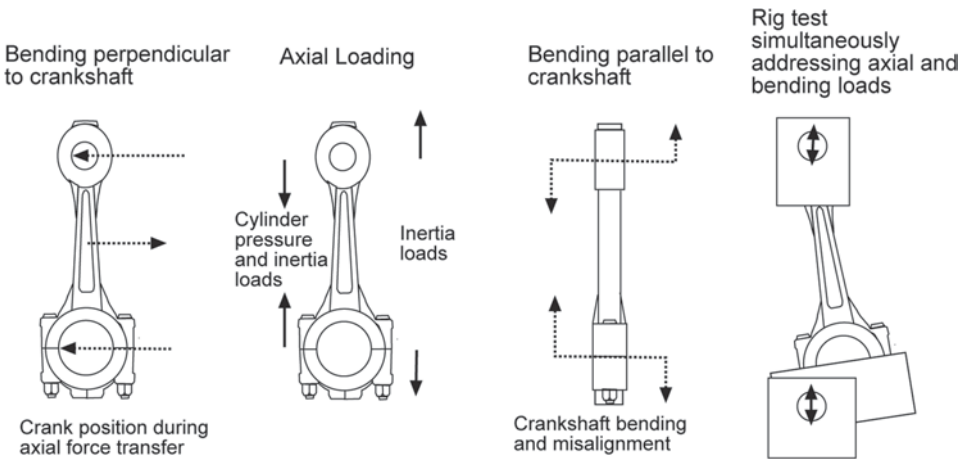


Fig. 16.36 Connecting rod loads and example rig test fixture

In connecting rod rig testing a load is applied at the piston pin bushing, axial to the main beam of the rod. The crankpin end of the rod is fixed by a shaft of the same diameter of the crankpin to limit bore distortion. These tests are sometimes accelerated by inputting a force at the natural frequency of the component. An enhancement of this test is to mount the crankpin end of the rod off center from the piston pin end, and this can simulate axial as well as bending load due to rod whip as presented in Fig. 16.36.

Once individual component testing is passed, system level tests can begin. These can again be servo-hydraulic, where a hydraulic cylinder applies load to simulate combustion to a short block assembly of crankshaft, connecting rods, and cylinder block. The short block can also be rotated on a motoring rig, where the engine is spun by an electric motor without combustion taking place. This also allows the verification of the lubrication and breathing systems, and the measurement of engine friction. A disadvantage of a motoring rig test is the lack of combustion forces. This will under-represent the force due to combustion, but will amplify the loading due to inertia since gas forces will not counteract inertia during TDC power stroke.

These system level tests allow easier access to components for strain measurement. The use of strain gages and telemetry or slip rings are also methods to obtain information. Finally, a specialty device known as a “grasshopper linkage” can be used to take direct measurement of strain on a moving connecting rod. This is a two link assembly with one end attached to the connecting rod, and one end attached to a fixed point in the engine. This removes the need to have strain gauge wires rotate around the crankshaft, and once assembled and moving resembles the rear leg of a grasshopper.

Once these components have completed testing, they are cleared for use on the engine dynamometer. Typically, low speeds are evaluated first, and engine load and speed are gradually increased as confidence grows.

16.12 Recommendations for Further Reading

The following paper provides a design and development flowchart specifically addressing crankshafts. The level of detail provided will be especially appreciated by a first-time crankshaft designer (See Dubensky 2002).

Although this is an older paper it provides a good summary of the dimensional details and design variables that determine crankshaft fatigue life (See Shaw and Richter 1979).

The following three papers summarize crankshaft development for new automobile engines. The first is for a V6 engine and the second an in-line four. The third paper, again for a V-6 engine, emphasizes NVH considerations (See Paek 1999; Fujimoto et al. 2001; Druschitz et al. 1999).

The following papers discuss crankshaft durability. The first discusses resonant bending fatigue testing, and the second discusses crankshaft forging and grain flow (See Yu et al. 2004; Shamasundar 2004).

Shot peening is an important process for applying compressive residual stress to crankshaft fillets, thus improving their fatigue life. This paper presents a detailed look at the process, the resulting surface characteristics, and the effect on fatigue life (See Wandell 1997).

This presentation, available through the Gas Machinery Institute provides an extremely detailed look at Torsional vibration in engines, and the effects of various vibration dampers (see Feese and Hill 2002).

The following paper presents a recent study of connecting rod fatigue in powdered metal and forged designs. The next paper discusses fracture-split connecting rod design. The text then listed devotes a chapter to connecting rod fatigue, and rig testing for rod development (See Afzal and Fatemi 2004; Park et al. 2003; Wright 1995).

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