Engine Configuration and Balance

6

6.1 Determining the Number and Layout of Cylinders

Once the fuel type, engine operating cycle, total displacement, and supercharging decisions have been made for a new engine, the next tasks will be to decide upon the number of cylinders over which the displacement will be divided, and the orientation of the cylinders. The factors that must be considered include cost and complexity, reciprocating mass and required engine speed, surface-to-volume ratio, pumping losses, packaging, and the balancing of mechanical forces. The majority of this chapter will address the mechanical forces and engine balancing as this is a key factor explaining why particular numbers and orientations of cylinders are repeatedly chosen.

6.2 Determining the Number of Cylinders

Having determined the total engine displacement, the number of cylinders into which that displacement is divided must be selected. Table 6.1 shows common cylinder volumes for different applications. These volumes vary depending on engine speed, duty cycle, and fuel choice.

Cost and complexity certainly lead the engineer to the lowest practical number of cylinders for any given engine. The quantities of many parts in the engine are directly multiplied by the number of cylinders, so the fewer the cylinders, the lower the cost of materials and assembly. On a cost-per-unit power basis, supercharging an engine will require less displacement and fewer cylinders, which reduces base engine cost. However, adding a compressor, intercooler, and additional intake plumbing to enable supercharging increases the cost of the total engine assembly. For a given cylinder size and power output, it is less expensive to continue to add more cylinders, until it starts to become cost neutral to add

K. Hoag, B. Dondlinger, *Vehicular Engine Design,* Powertrain,
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Table 6.1 Typical ranges for cylinder volume

Engine type	Cylinder volume (cc)
Small utility	140–500
Motorcycle (sport)	100–350
Motorcycle (cruiser)	375–1000
Automotive gasoline	350–850
Automotive turbo-diesel	475–840
Over-the-highway truck turbo-diesel	840–2660

supercharging above approximately seven cylinders. In other words, it may cost less to build a four-cylinder, super-charged engine of a given power output than a comparable eight-cylinder engine of the same power output.

Conflicting with the requirement to reduce cost and complexity, ***reciprocating mass*** is reduced by increasing the number of cylinders for a given engine displacement. As cylinder diameter is reduced the mass of the piston, rings, piston pin, and connecting rod diminish as well. At any instant in time the force transmitted through a connecting rod, and seen by the rod and main bearings, crankshaft, cylinder block and main bearing caps is the product of the mass of the reciprocating components and the instantaneous rate of acceleration or deceleration. Higher engine speed requirements result in greater acceleration rates, and in order to keep the reciprocating forces manageable the reciprocating mass must be reduced. It follows directly that increasing the number of cylinders is attractive for high speed engines, as it allows the mass to be distributed, reducing the force transmitted at each cylinder.

Next, the engine designer must choose the ***cylinder arrangement***: in-line, vee, W, horizontally opposed, radial, or Wankel, as shown in Figs. 6.1, 6.2, 6.3, 6.4 and 6.5. Packaging

Fig. 6.1 Inline configuration

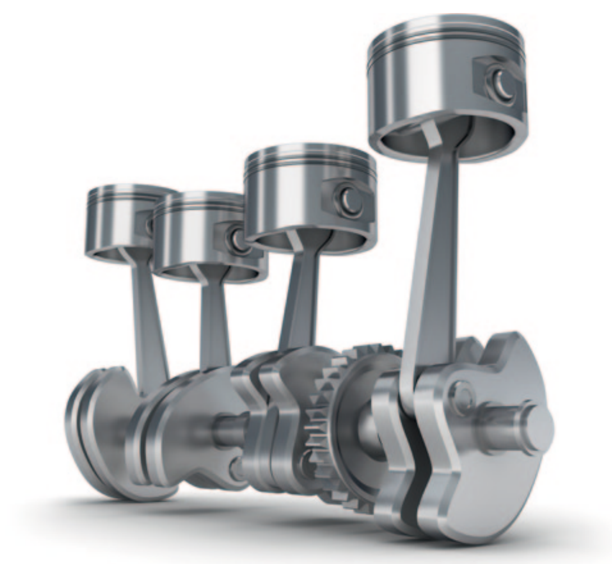

Fig. 6.2 Vee configuration

Fig. 6.3 Horizontally opposed configuration

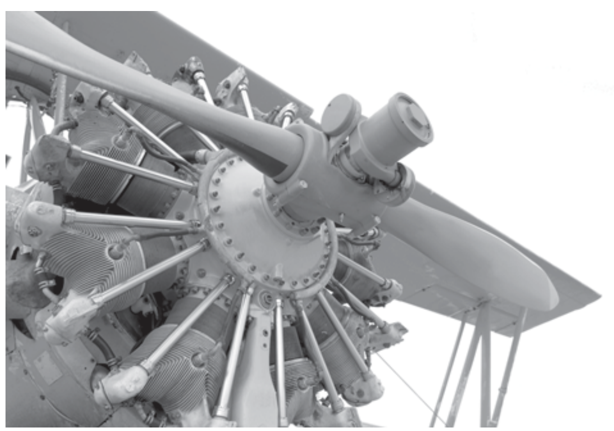

Fig. 6.4 Radial configuration

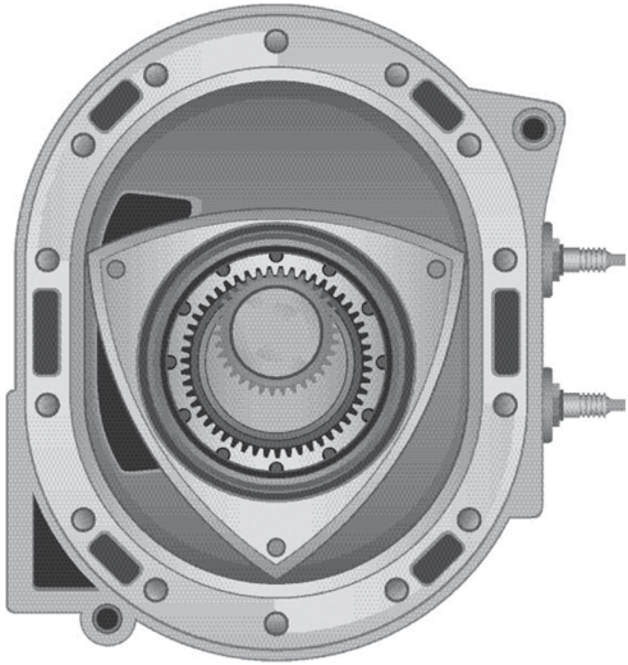

Fig. 6.5 The Wankel rotary engine

considerations include not only the length, width, and height of the engine, but installation requirements such as those pertaining to intake and exhaust fittings and locations, and cooling system connections. The need for duplication of some components with certain engine configurations—cylinder heads or exhaust piping in vee engines for example—must also be considered. The in-line engine, with a single bank of cylinders provides the simplest configuration. Because only one bank of cylinders must be served, the intake, exhaust, and cooling systems are all easily configured. However, the engine height creates packaging challenges in some applications, and as the number of cylinders increases, length may become prohibitive. The height is sometimes addressed by tipping the engine in a "slant" mounting, or by mounting it horizontally.

The reduced height of alternative configurations comes at the price of increased width. The vee configuration allows a reduction in both height and length at the expense of width, but now requires two banks of cylinders to be supported with cooling, intake, and exhaust systems. A further potential disadvantage is that of reduced crankshaft and connecting rod bearing area. Whereas the in-line engine allowed a main bearing to be placed between each cylinder, the vee configuration results in a pair of cylinders between each pair of main bearings. With only slightly increased spacing between the main bearings sufficient bearing area must be supplied for two rod bearings. These challenges are met without difficulty in most automobile engines, but are very difficult to meet in heavy-duty applications. The compact 'W' configuration can be thought of as two vee engines placed side-by-side with a single crankshaft. The 'W' engine further demonstrates the trade-off between engine width and length. While there are now four banks of cylinders their close proximity reduces (but does not eliminate) the challenges of intake, exhaust, and cooling system packaging.

The horizontally opposed engine provides minimum height, but a very wide package. The length of the engine is similar to that of the vee engine, and it faces similar bearing area challenges. The support system challenges are made greater by the greater physical distance between cylinder banks.

Beginning from an in-line engine changing the configuration to a vee significantly reduces the length of the engine along the crankshaft. To exaggerate that line of thought, a radial engine can be though of as only one cylinder long, with many cylinders arranged around the crankshaft. This dramatically reduces the length of the engine, but significantly complicates the connecting rod to crankshaft packaging. A special type of articulated connecting rod is required. This arrangement was typically used in air-cooled aircraft engines, where each exhaust port needed to share the same amount of cooling airflow for maximum power density.

A different type of internal combustion engine is the rotary piston engine, or Wankel engine named after its creator. In this arrangement, a triangular rotor moves around the crankshaft, creating increasing and decreasing volumes within its housing. Several combustion events per rotation per working cylinder are possible, which enables a high power density for this engine. The rotary motion of the working piston produces much less vibration than a conventional reciprocating piston engine. Unfortunately, an inherently poor combustion chamber shape and difficulty sealing the combustion chamber lead to fuel consumption and exhaust emission challenges that have limited the Wankel engine's use.

6.3 Determining the Cylinder Bore-to-Stroke Ratio

Once the number and configuration of cylinders has been determined the final remaining basic layout question is that of bore-to-stroke ratio. For any given cylinder displacement volume a theoretically infinite range of cylinder bore-to-stroke ratios could be chosen. Choosing the bore-to-stroke ratio will have a fundamental impact on the overall engine size. As the bore increases the block length will grow; as the stroke increases the deck height will raise. Choosing the bore-to-stroke ratio is also one of the key drivers to how the engine will perform and has many effects on engine power output, valvetrain architecture, thermal efficiency, combustion efficiency, pumping losses, and mechanical friction. The bore-to-stroke ratio is calculated in Eq. 6.1, and comparison data is presented in Fig. 6.6.

$$\text{Bore-to-Stroke Ratio} = \frac{\text{Cylinder Bore Diameter or Piston Diameter}}{\text{Piston Stroke Length}} \tag{6.1}$$

The first thing to consider when choosing bore-to-stroke ratio is the affect the ratio will have on rated or peak power speed. Valve diameter and timing events are set to optimize

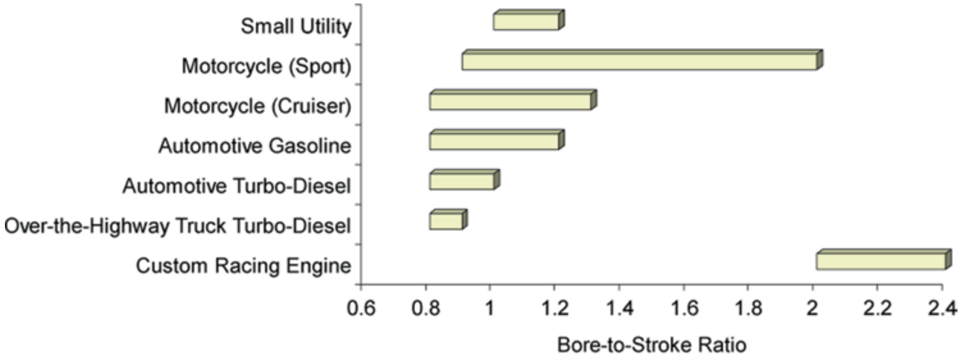

Fig. 6.6 Bore-to-stroke ratio comparison

intake air velocity at the desired peak power speed. The bore-to-stroke ratio and valvetrain must be matched to maximize the output of a given engine. A large bore-to-stroke ratio will enable higher engine speeds, and due to the larger packaging size will enable larger valves, which will allow the intake air velocity to be optimized at high engine speeds, resulting in an engine with high power density. If a small bore-to-stroke ratio is chosen, the long stroke enables the peak power speed to occur at lower engine speed. Comparison data is shown in Fig. 6.7.

The reciprocating piston has continually changing speed and acceleration rates throughout its stroke. At TDC, the piston speed is zero, and it must accelerate to maximum piston speed near mid-stroke, and then decelerate to a full stop at BDC for each stroke. The maximum piston speed and acceleration are related to the piston stroke and the engine speed. The higher the acceleration, the higher the mechanical stress on reciprocating en-

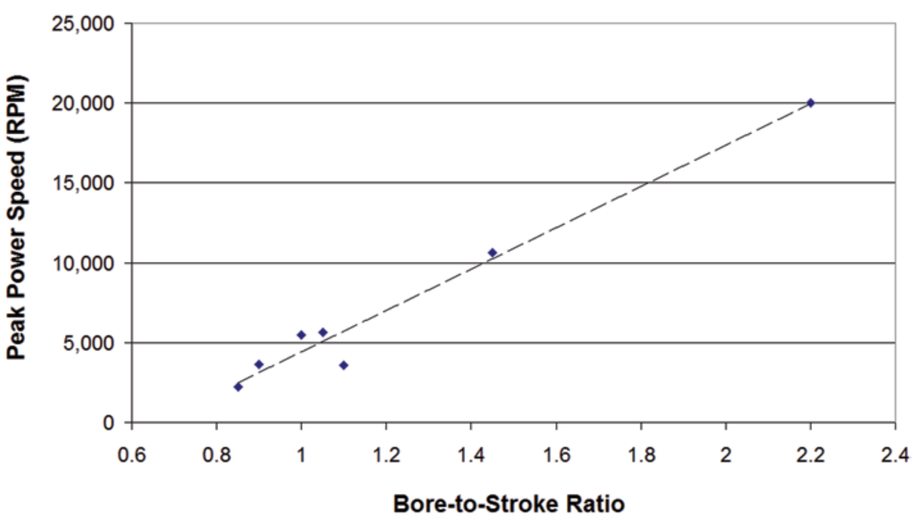

Fig. 6.7 Peak power speed as a function of bore-to-stroke ratio

gine parts. A useful way to describe the design limit is with the ***mean piston speed*** shown in Eq. 6.2. Note the tradeoff that increasing the engine stroke has on engine speed for a given mean piston speed. A high mean piston speed is one of the structural limits imposed on any engine design.

$$\text{Mean Piston Speed, } V_p = 2 \cdot S \cdot N \tag{6.2}$$

Where:

S = Piston stroke
N = Engine Speed

The role of the valvetrain is to allow sufficient air exchange capacity (breathing) and to control the timing of the intake and exhaust events, while not exceeding the limits of contact stresses on the camshaft or the stress limits of other valvetrain components. An approximation of valve inertia forces is illustrated in Eq. 6.3. As the engine's bore is increased, the diameter of the intake valve must also increase to allow sufficient breathing capacity at higher engine speeds. As the valve diameter increases, so does its mass. At low engine speeds, a pushrod or overhead valve (OHV) arrangement may be sufficient to maintain valve timing and meet component stress requirements. As engine speed increases, a stiffer valve spring is required to keep the valve following the cam profile. As engine speed increases beyond a certain point, valve accelerations increase to the point where the valve spring will no longer be able to control the valvetrain due to the large mass, and an overhead cam (OHC) arrangement may be required to reduce valvetrain mass and maintain valve timing. An alternate approach to achieve these same combinations of breathing, timing and stress requirements is to split the single valve mass into two or more valves. This will enable sufficient valve area for breathing while reducing the individual valve mass and inertia.

$$F_v \approx m \cdot h \cdot N^2 \tag{6.3}$$

Where:

F_v = inertia force on valve
m = mass of the valve
h = max lift of the valve

General industry values are shown in Table 6.2**.** As regulatory pressure increases to improve fuel consumption and minimize exhaust emissions, the industry will continue to respond with engines of higher specific output (down-sizing and down-speeding). These general industry values will continue to rise with further development.

Table 6.2 Typical performance ranges for engine types

Engine type	Bore-to-stroke ratio (B/S)	Peak power speed (RPM)	Mean piston speed (m/s)	BMEP at peak torque (MPa)
Small utility	1.0–1.2	3600	8–10	0.8–1.0
Motorcycle (Sport)	0.9–2.0	6800–14,500	16–24	1.0–1.5
Motorcycle (Cruiser)	0.8–1.3	4250–7000	13–23	0.9–1.2
Automotive gasoline	0.8–1.2	4000–7000	13–24	1.1–1.3
Automotive turbo-diesel	0.8–1.0	3300–4000	9–13	1.3–2.1
Over-the-highway truck turbo-diesel	0.8–0.9	1800–2600	10–14	1.9–2.3
Custom racing engine	2.0–2.4	20,000	22–33	1.6

Surface-to-Volume ratio refers to the ratio of combustion chamber surface area to combustion chamber volume, and has a significant effect on heat transfer and combustion efficiency. The actual value of surface-to-volume ratio changes as cylinder volume changes due to piston movement throughout the operating cycle. While surface-to-volume ratio is nearly independent of bore-to-stroke ratio when considering the BDC volume, it is strongly dependent on bore-to-stroke ratio at or near the TDC cylinder volume. For the same cylinder displacement, a larger bore and shorter stroke engine has a higher TDC surface-to-volume ratio and thus greater heat rejection from the cylinder. The surface-to-volume ratio is of greatest importance near TDC, when energy is rapidly released during combustion, and heat rejection to the combustion chamber walls is to be minimized.

The surface-to-volume ratio is impacted by both the number of cylinders and the bore-to-stroke ratio of each cylinder. As can be seen from Eqs. 6.4 to 6.6 below, the larger the cylinder, the lower the surface-to-volume ratio. Increasing or decreasing the number of cylinders for a given engine displacement correspondingly increases or decreases the surface-to-volume ratio.

$$\text{Surface Area of a Sphere} = A_S = 4\pi r^2 \tag{6.4}$$

$$\text{Voume of a Sphere} = V_S = \frac{4}{3}\pi r^3 \tag{6.5}$$

$$\text{Surface-to-Volume Ratio (S/V)} = \frac{A_S}{V_S} = \frac{4\pi r^2}{\frac{4}{3}\pi r^3} = \frac{3}{r} \approx \frac{1}{r} \tag{6.6}$$

Because of the effect bore-to-stroke ratio has on heat transfer, it will also have a significant effect on ***combustion efficiency***. For different reasons the combustion chambers of both diesel and spark-ignition engines become more difficult to optimize as bore diameter increases. For a given cylinder size, the bore-to-stroke ratio will have a significant effect on the shape of the combustion chamber. As the bore diameter increases, the aspect ratio of the combustion chamber will flatten out to a shallow disc.

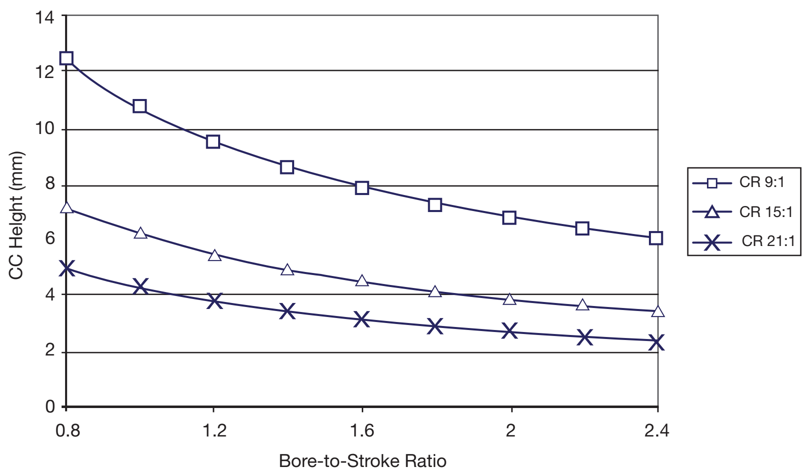

Fig. 6.8 Effect of bore-to-stroke ratio on combustion chamber height

In spark-ignition engines increasing the bore-to-stroke ratio increases the flame travel distance, slowing the energy release rate and increasing the potential for early flame quench and cycle-to-cycle variability. As bore diameter is increased above approximately 105 mm a second sparkplug will be required to reduce flame travel distance and time. A larger bore also increases the top piston ring crevice volume and correspondingly increases un-burnt hydrocarbons. In the diesel engine the larger bore results in a shallower combustion chamber to maintain compression ratio, and leads to a greater propensity for fuel injection spray impingement on either the cylinder head or piston surface.

The bore-to-stroke ratio will also affect the maximum ***compression ratio*** (CR) possible, as defined in Eq. 6.7. A high compression ratio is desirable because it improves the thermal efficiency and generally the fuel efficiency of the engine. For a fixed cylinder volume, as the bore increases and the stroke decreases, it is difficult to achieve a high compression ratio as the valves begin to interfere with the piston crown at TDC-overlap as shown in Fig. 6.8. The plots were generated assuming the combustion chamber is a cylindrical disc of equal diameter to the piston bore.

$$\text{Compression Ratio} = \frac{\text{Volume max}}{\text{Volume min}} = \frac{V_{cc} + V_{cyl}}{V_{cc}} \tag{6.7}$$

Where:

V_{cc} = Combustion Chamber Volume
V_{cyl} = Swept Volume of Cylinder

Pumping losses increase as the length of intake and exhaust ducts increase, and as their diameters decrease. As the number of cylinders increase additional duct length is often required. The adverse effects can often be countered through tuning—selecting duct lengths

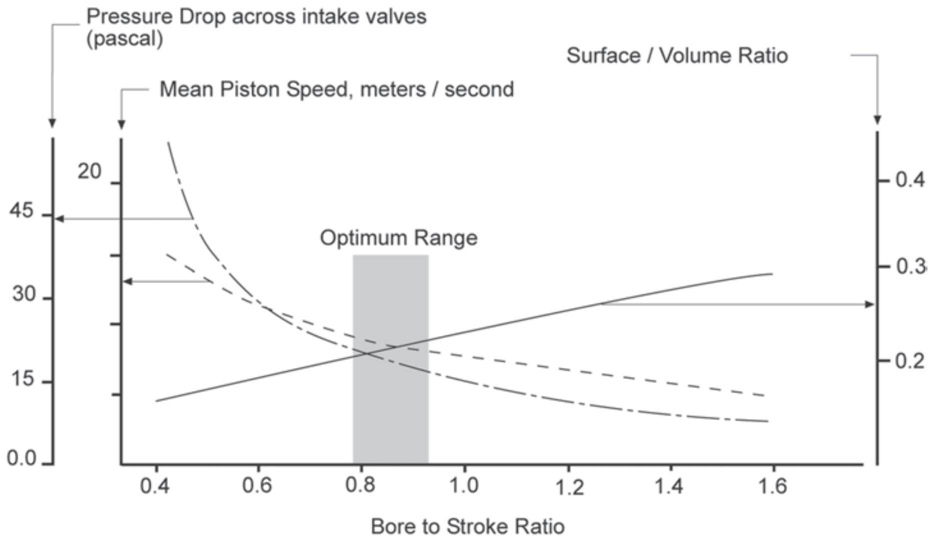

Fig. 6.9 Effects of variables determining the optimum bore-to-stroke ratio

that result in dynamic pressure pulses aiding the cylinder filling and emptying processes at chosen engine speeds. An additional challenge is that of duct diameter, and specifically valve diameters as the number of cylinders is increased, and the resulting bore diameter of each cylinder becomes smaller.

Frictional losses are also modestly affected by bore-to-stroke ratio. For large bore-to-stroke ratios the same combustion pressure is applied over a greater piston area, leading to higher unit loads on bearings. The higher unit loading requires larger bearings and increases hydrodynamic losses. Piston weight increases as a function of diameter and the resulting higher inertial forces will also increase the unit loads on the bearings. At the other extreme, the long stroke engine will see higher mean piston speeds which will increase rubbing friction between the piston and rings and the cylinder bore.

In summary, trade-offs associated with the parameters just described are presented for a given engine in Fig. 6.9**.** In one sense this plot is misleading, as each of the parameters is plotted on a different dependent axis. However it is important to consider the combined effect of each parameter. The bore must be made sufficiently large to keep the mean piston speed below its design target and to minimize pressure drop across the valves (since the pressure drop increases sharply as the valve area is reduced). As a general rule the bore is made no larger than is necessary to fulfill these two requirements—this minimizes heat rejection to the coolant. The resulting bore-to-stroke ratio will differ depending on the design criteria for the specific engine. In automotive applications, fuel efficiency and emission requirements dominate, so the bore-to-stroke ratio is typically just below unity to give the best balance of performance. In high performance applications where the engine displacement is typically limited by racing rules, and a high specific power is required, the bore-to-stroke ratio tends to be larger in order to maximize breathing and provide acceptable piston speeds at high engine speed.

6.4 Vibration Fundamentals Reviewed

An important factor that must be considered in determining the number and configuration of cylinders is that of controlling, and as much as possible balancing, the mechanical forces generated within the engine. In this section the basic concepts of mechanical vibration are reviewed, and the forces generated within the engine are identified. Rotating and reciprocating forces are then separately discussed in the next two sections.

Vibration is defined as the response resulting from any force repeatedly applied to a body. The repeated force may be random, or a force of a given magnitude may be applied at some constant frequency. In the case of forces generated within an engine the magnitudes will be constant at any given speed and load, and will change with either speed (rotating and reciprocating forces) or speed and load (gas pressure forces). Since the frequency of the vibration forces is determined by engine speed it is convenient to define the vibration ***order*** as the vibration frequency relative to shaft speed. A ***first-order*** vibration is generated by forces that are applied over a cycle that occurs once every crankshaft revolution. A ***half-order*** vibration occurs once every second crankshaft revolution. Half-orders may be important in engines operating on a four-stroke cycle—for example, the gas pressure force applied at one cylinder is repeated over a cycle of two crankshaft revolutions. A ***second-order*** vibration occurs twice every crankshaft revolution, and so on.

In assessing engine vibration it is convenient to use a Cartesian coordinate system. The forces acting on the engine can be thought of as attempting to translate the engine along, or rotate it around, each of the three axes. The engine assembly (typically engine and transmission or transaxle) is constrained by the engine mounts, and any forces or moments not counteracted within the engine are transmitted through the mounts to the vehicle. The forces acting along each axis must sum to zero, and the moments acting around each axis must also sum to zero or a mechanical vibration is transmitted to the vehicle.

The engine simultaneously experiences a variety of internal forces. First, at each cylinder some portion of the crankshaft and connecting rod must be centered at some distance away from the crankshaft centerline as determined by the engine's stroke. As the crankshaft spins this mass gives rise to a centrifugal force of a magnitude dependent on engine speed, and acting outward from the crankshaft at each instant in time along the centerline of the particular crankshaft throw. The reaction force is transmitted to the block through the main bearings—it is split between the main bearings surrounding the particular cylinder, and further distributed along the crankshaft to the remaining main bearings in quantities dependent on the length and stiffness of the crankshaft, and any block deflection that may be occurring. In a multi-cylinder engine such forces are simultaneously applied at each crankshaft throw. This results in various moments whose magnitude and direction are determined by the orientation of the throws along the crankshaft. Further mechanical forces are transmitted to the crankshaft at each cylinder due to the continuous acceleration and deceleration of the pistons. These reciprocating forces vary in magnitude with crank angle, and act along the axis of each cylinder centerline. Finally, gas pressure forces, varying continuously throughout the engine operating cycle, are transmitted to the crankshaft at each cylinder.

Two further observations must be made before proceeding into a detailed look at the calculation and management of these forces. First, although what the engine "sees" is the instantaneous summation of each of the forces just described, calculations will be facilitated by recognizing the powerful tool of superposition. If the forces are broken down as described in the previous paragraph each effect can be separately calculated. The resultant seen by the engine is simply the sum of each of the contributions along any given axis. Second, the reciprocating piston engine operates on the kinematic principles of converting reciprocating motion at the piston to rotating motion at the crankshaft. The conversion occurs across the connecting rod, with everything connected at its small end experiencing purely reciprocating motion, and everything at its large end experiencing purely rotational motion. The connecting rod itself experiences complex motion that would be extremely difficult to calculate. Fortunately such calculations are not necessary. A very close approximation can be achieved simply by weighing each end of the connecting rod. The mass is split about its center of gravity, between that in reciprocating and that in rotating motion.

6.5 Rotating Forces and Dynamic Couples

The centrifugal force generated at each crankshaft throw is calculated as depicted in Fig. 6.10. As the crankshaft spins each element of mass located at a distance away from the crankshaft centerline generates an outward centrifugal force calculated as

$$F_{Rotational} = \frac{M \cdot r \cdot \omega^2}{g_c} \tag{6.8}$$

Where:

M = mass
r = radial distance from shaft centerline (stroke/2)
ω = angular velocity, in radians per unit time
g_c = gravitational constant

The mass that must be considered in this calculation includes that of the crankshaft itself as well as that of the portion of the connecting rod affixed to the crankshaft at the given location. The resulting mass and its center of gravity are used to determine the values to be used in Eq. 6.8. Such calculations have traditionally been made using graphical methods

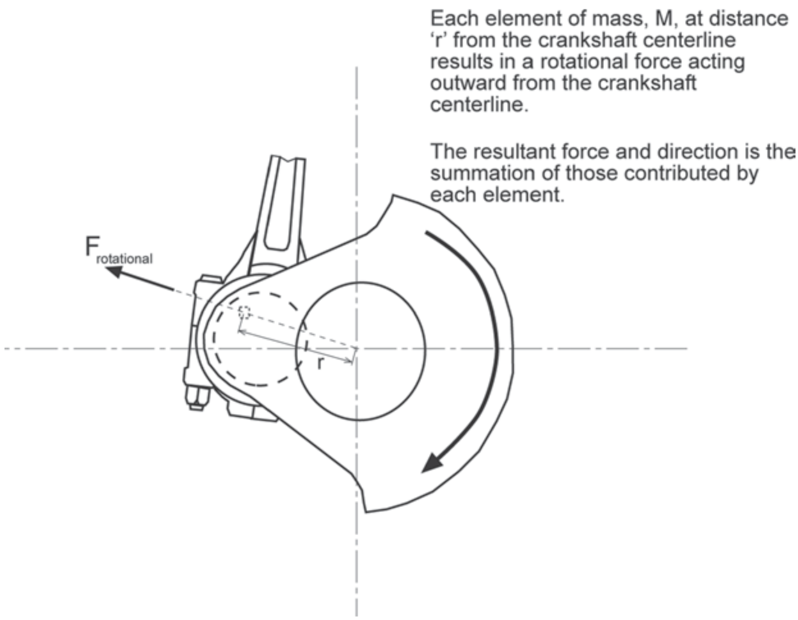

Fig. 6.10 Summary of centrifugal force calculation, showing contributing element and variables pertaining to that element. Each element of mass *m* at distance *r* from the crankshaft centerline results in a rotational force acting outward from the crankshaft centerline. The resulting force and direction is the summation of those contributed by each element

in which the crankshaft section was laid out on a fine grid, and the contribution of each region was calculated to determine the resultant mass and location of the center of gravity. Today such calculations are included in the computer aided design (CAD) software used in most engine design work.

If the centrifugal force generated at any given cylinder or crankshaft throw is considered it should be apparent that mass can be added to the crankshaft opposite of the throw in order to completely balance the force. The added masses are referred to as the crankshaft ***counterweights***. However, in a multi-cylinder engine it may be possible to balance the centrifugal force generated at one crankshaft throw with that generated at another throw. Referring to Fig. 6.11a a simplified crankshaft section is represented in which two adjacent throws are configured 180° apart. The centrifugal force generated at one throw is exactly counteracted by that generated at the adjacent throw, and the sum of forces along the plane of the crankshaft is zero. However, because the throws are located at different positions along the length of the crankshaft the moment generated by this pair of throws is not zero. As shown in the figure, a moment will be centered at the midpoint between the throws. This is referred to as a ***dynamic couple***, and its magnitude is calculated as

$$\sum F_{Rotational} \cdot L \tag{6.9}$$

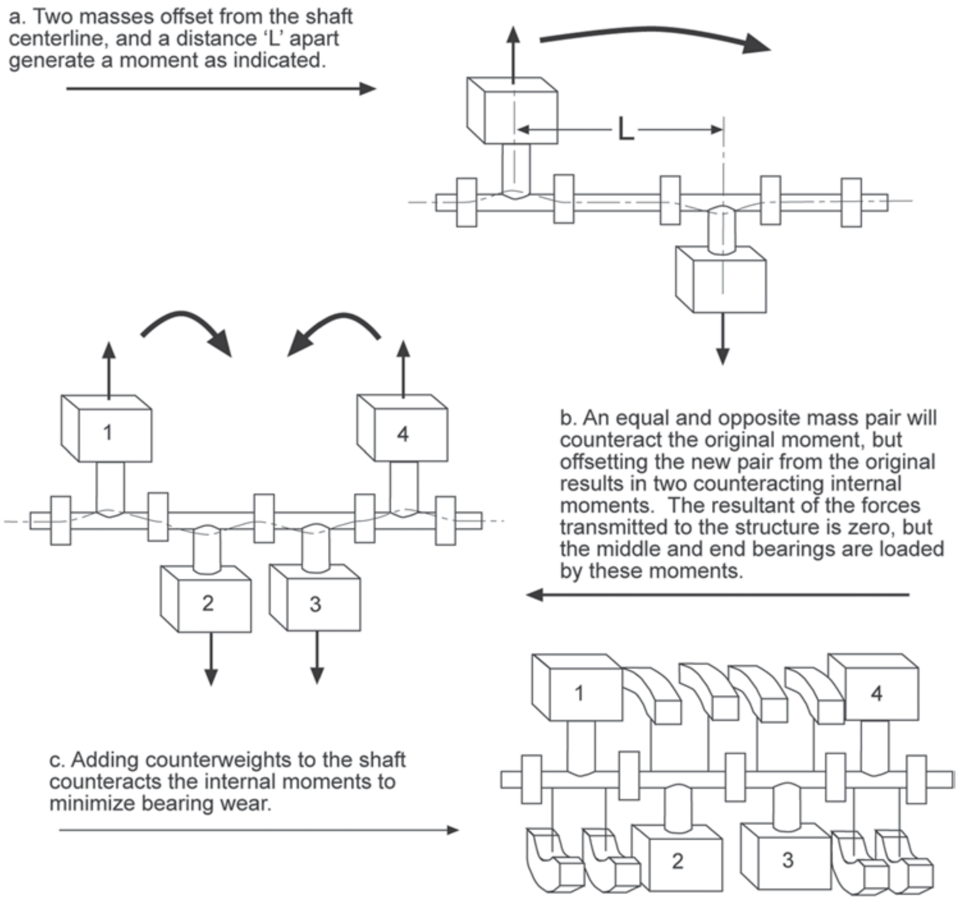

Fig. 6.11 Dynamic couples and their balance, leading to the example of a four-cylinder crankshaft

Where:

L = distance along shaft axis where force is applied, to the neutral axis about which the moment occurs

In Fig. 6.11b the crankshaft representation is extended to that for a four-cylinder engine having the typical firing order of 1-3-4-2. The throws for cylinders two and three are 180° apart from those for cylinders one and four. The centrifugal forces all act in the same plane and again balance one another out. If the moments are now considered across the entire length of the crankshaft it is found that these too are completely balanced. The dynamic couple identified earlier (cylinders one and three) has been balanced by an equal and opposite couple (cylinders two and four) as depicted in the figure. However, because the centrifugal forces are applied at various points over the length of the crankshaft it is also

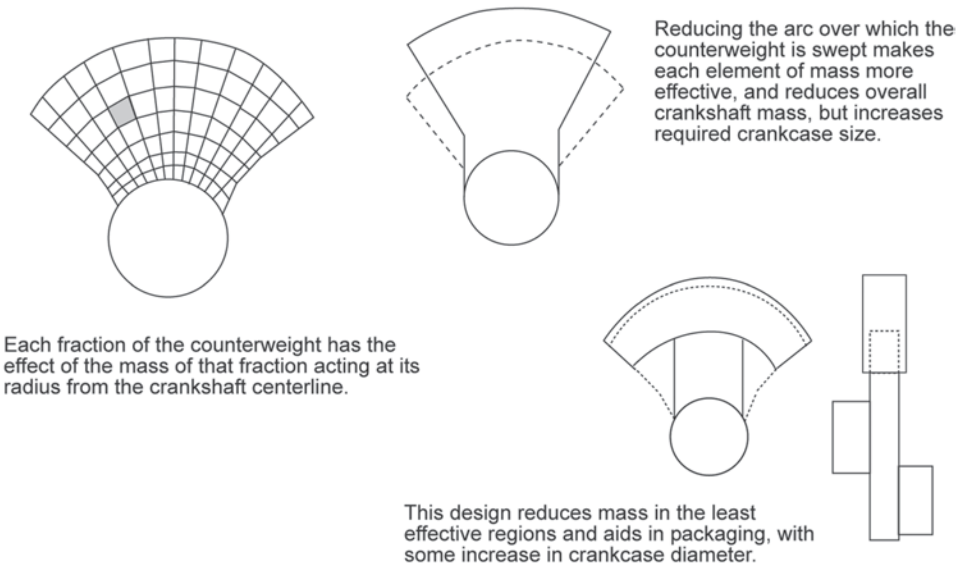

Fig. 6.12 Counterweight design considerations

important to consider the force being transmitted from the crankshaft to the block at each bearing. While the balanced forces and moments result in zero net force attempting to translate or rotate the block (and thus having to be countered by the engine mounts) the two internal moments result in crankshaft deflection and additional main bearing loads at certain bearings. The axes about which the moments rotate are centered at the number two and four main bearings and these two bearings will not see any loads induced by the moments. Bearings one, three, and five see additional loads as a result of the moments. Counterweights, as shown in Fig. 6.11c, are therefore added to this four-cylinder crankshaft, not to achieve rotational balance but to reduce crankshaft deflection and main bearing loads.

It is important to emphasize again that the discussion so far has considered only rotational forces. For the simple case of the four-cylinder engine it was demonstrated that each centrifugal force or dynamic couple that attempts to translate or rotate the engine is completely balanced, and no net force remains that would be transmitted to the vehicle at the engine mounts. This is not to say that no vibration forces whatsoever are transmitted to the engine mounts—only that none resulting from rotational forces are transmitted. Reciprocating and gas pressure forces have not been introduced in this discussion and will in fact transmit forces through the engine mounts.

Several design trade-offs pertaining to the counterweights are summarized in Fig. 6.12. While the counterweights too can each be represented as a single mass acting at some local center of gravity, the actual counterweights consists of elements of mass at various distances from the crankshaft centerline and from the plane about which the mass is to be centered. Mass located at positions furthest from the shaft centerline, and closest to the center plane is most effective. Locating more mass in these most effective regions allows the total crankshaft mass to be reduced, but at the penalty of larger crankcase size. In some engines, especially those designed to operate at high speed it may be advantageous to devote attention to the aerodynamics of the counterweight. This too may result in the need for greater total crankshaft mass.

The four-cylinder engine taken as the example for the initial discussion of rotating balance and couples has a “planer” or “flat” crankshaft—the crankshaft is centered about a single plane, and all of the forces and moments act in that plane. This is not the case with various other engine configurations of interest for automotive applications. The calculations become more complicated, but the methodology remains the same. As an example, consider the inline six-cylinder engine whose crankshaft is represented in Fig. 6.13. The typical firing order is 1-5-3-6-2-4. The crank throws are each 120° apart, with the throws for cylinders one and six in the same plane, five and two another plane, and three and four in a third plane. Equal, outward centrifugal forces are acting at 120° intervals, and this

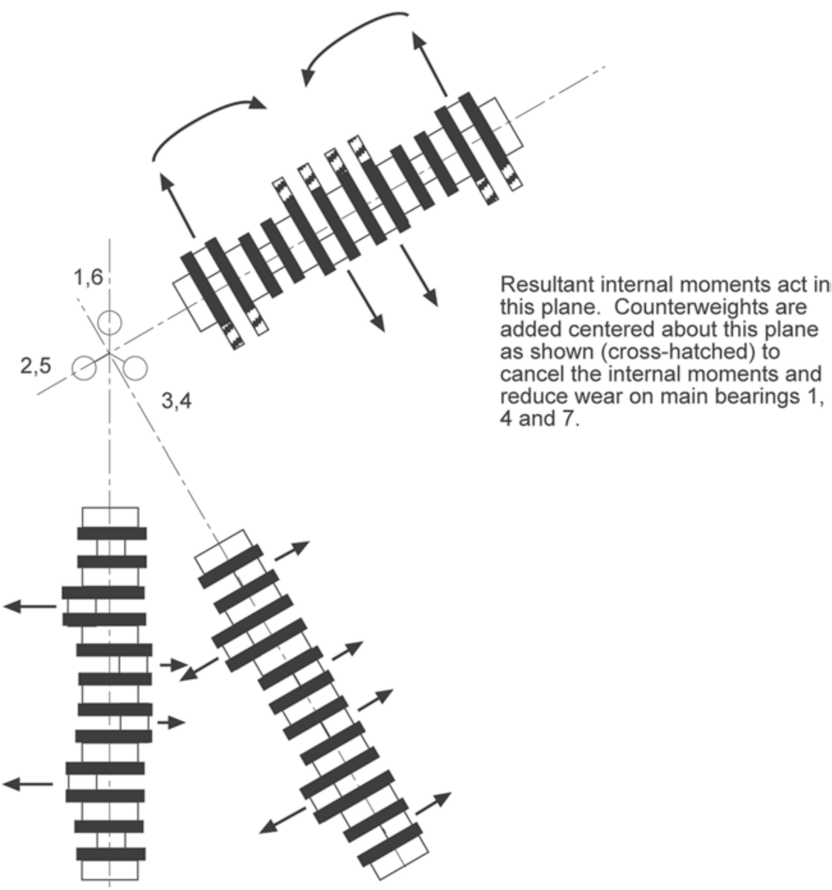

Fig. 6.13 Determining the plane of the resultant moment, using the example of an in-line six-cylinder crankshaft

too results in balanced forces. The reader is encouraged to verify this fact by selecting a plane through any of the pairs of cylinders just identified, and calculating all of the force components acting in that plane. They will be found to sum to zero.

Again selecting any plane through the crankshaft and assessing the force components in that plane allows the moments to be calculated. It is found for this engine configuration that the moments balance across the engine as a whole, but that the front three cylinders and the back three cylinders each generate an equal and opposite internal moment. The counterweights required to counteract these moments must be centered about the plane in which the moments are maximized. The moments can be assessed over a variety of planes in order to determine the plane in which to center the counterweights, as further depicted in Fig. 6.13. In the case of the in-line six-cylinder engine the counterweights are centered in the plane 90° from that of cylinders two and five. Similar exercises can be done for any cylinder configuration.

Finally, it should be noted that an alternative to the approach just discussed is to counterweight each cylinder individually with mass opposite to each throw. For example, instead of the counterweights shown in Fig. 6.13 all of the cylinders would be equally counterweighted, with mass placed on each web and centered 180° from the centerline of the particular crank throw. Because each cylinder is balanced individually the resulting mass at each cylinder (12 counterweights in the in-line 6-cylinder example) is less than that shown at the 8 locations in the figure. This provides the advantage of a smaller crankcase, and both the advantages and disadvantages of a stiffer and heavier crankshaft. This approach can be taken relatively easily with a cast crankshaft, but if the crankshaft is to be forged the additional counterweights would have a detrimental impact on forging process complexity and cost.

The variety of engine configurations conceivable for automotive applications is too great to allow detailed discussion of each configuration. The principles discussed here can be applied for any configuration of interest. A summary of the results for a number of popular configurations is provided in Sect. 6.6.

6.6 Reciprocating Forces

The reader is reminded of the discussion in Sect. 6.4, where it was stated that while the engine simultaneously experiences rotating and reciprocating forces it is convenient to separate these for calculation purposes. Rotating forces and moments and their balance were the topic of Sect. 6.5. In this section reciprocating forces and balance will be taken up.

The piston, piston pin, rings, and the upper portion of the connecting rod are subjected to reciprocating motion, and thus repeated acceleration and deceleration. The reciprocating forces act along each cylinder centerline, and their magnitude varies continuously with crank angle. The magnitude of the reciprocating forces at any given crank angle position can be determined from Newton's Second Law as the product of the reciprocating mass and the instantaneous acceleration at that crank angle.

Piston velocity reaches zero at both the TDC and BDC positions, following a non-symmetric trace determined by the slider crank geometry between those positions. The peak velocity does not occur midway between TDC and BDC, but is skewed toward TDC. If the connecting rod were infinitely long the velocity trace would be sinusoidal, but as the connecting rod is made shorter the velocity trace becomes increasingly skewed. The resulting velocity versus crank angle is the following series expression:

$$V = -\omega^2 r[\sin\theta + 2a_2 \sin 2\theta + 4a_4 \sin 4\theta + \ldots] \tag{6.10}$$

Where:

ω = Angular velocity of crankshaft, in radians per unit time
r = Radial distance of rod bearing axis from crankshaft centerline (stroke/2)
θ = Crank angle relative to TDC
L = Connecting rod length

$$a_2 = \frac{L}{r}\left[\frac{1}{4}\left(\frac{r}{L}\right)^2 + \frac{1}{16}\left(\frac{r}{L}\right)^4 + \frac{15}{512}\left(\frac{r}{L}\right)^6 + \ldots\right] \tag{6.11}$$

$$a_4 = -\frac{L}{r}\left[\frac{1}{64}\left(\frac{r}{L}\right)^4 + \frac{3}{256}\left(\frac{r}{L}\right)^6 + \ldots\right] \tag{6.12}$$

Of current interest is the instantaneous acceleration, which is simply the velocity derivative with respect to time (in this case crank angle):

$$\frac{dV}{d\theta} = -\omega^2 r[\cos\theta + 4a_2 \cos 2\theta + 16a_4 \cos 4\theta + \ldots] \tag{6.13}$$

The instantaneous reciprocating force at any given cylinder can now be calculated as:

$$F_{\mathrm{Re}ciprocating} = M_{\mathrm{Re}ciprocating} \frac{dV}{d\theta}$$

$$F_{\mathrm{Re}ciprocating} = -M_{\mathrm{Re}ciprocating}\,\omega^2 r[\cos\theta + 4a_2 \cos 2\theta + 16a_4 \cos 4\theta + \ldots] \tag{6.14}$$

Equation 6.14 can be greatly simplified by making two observations. First, unless the connecting rods are very short it is observed that the higher order terms are quite small and only the first two must be considered. Second, the term ‘$4a_2$’ is observed to be almost identically ‘$r/_L$.’ These observations result in the simplified acceleration expression,

$$F_{\mathrm{Re}ciprocating} = -M_{\mathrm{Re}ciptrocating}\,\omega^2 r\left[\cos\theta + \frac{r}{L}\cos 2\theta\right] \tag{6.15}$$

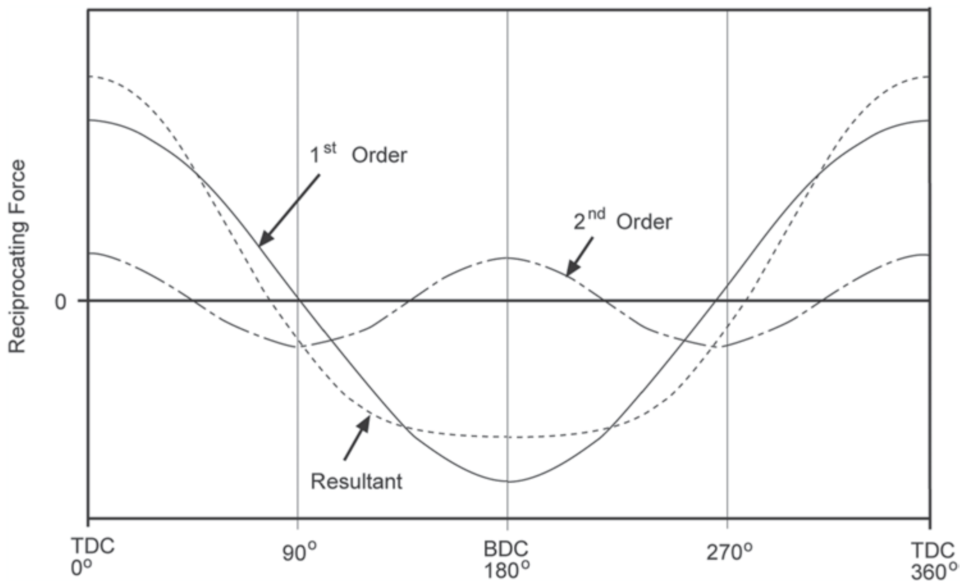

Fig. 6.14 First- and second-order reciprocating forces and the resultant force versus crank angle for an individual cylinder

It may now be recognized that of the two remaining crank angle terms the first varies at the rate of once per crankshaft revolution (first order), and the second varies at the rate of twice per crankshaft revolution (second order). While the reciprocating force seen by the engine at each cylinder is that calculated by this entire equation it is helpful to break the equation down into first- and second-order terms, and consider each separately. The first and second order terms and the resultant reciprocating force are each plotted in Fig. 6.14.

A physical understanding of the first and second order terms can be gained by returning to the in-line four-cylinder engine. If a second horizontal axis is added to Fig. 6.14, as shown in Fig. 6.15a, the net reciprocating forces from a bank of cylinders can readily be seen. On the second axis the first cylinder is located at its TDC position. Each remaining cylinder is located at the position it occupies when the first cylinder is at TDC. For the case of the four-cylinder engine the standard firing order, 1-3-4-2, results in pistons two and three at BDC, and piston four along with piston one at TDC. The reciprocating forces generated by each piston assembly at this moment in time are readily seen. For this engine, the two cylinders (one and four) decelerating to zero velocity as they approach TDC transmit their maximum first order reciprocating force upward on the cylinder block. At this same moment in time, cylinders two and three are decelerating to zero velocity as they approach BDC, and exert the same reciprocating force downward into the main bearing caps. The net result is zero, and the engine has balanced primary reciprocating forces. Looking now at the second order forces it can be seen from Fig. 6.15a that all four cylinders simultaneously reach their maximum second order force and this engine thus has a second

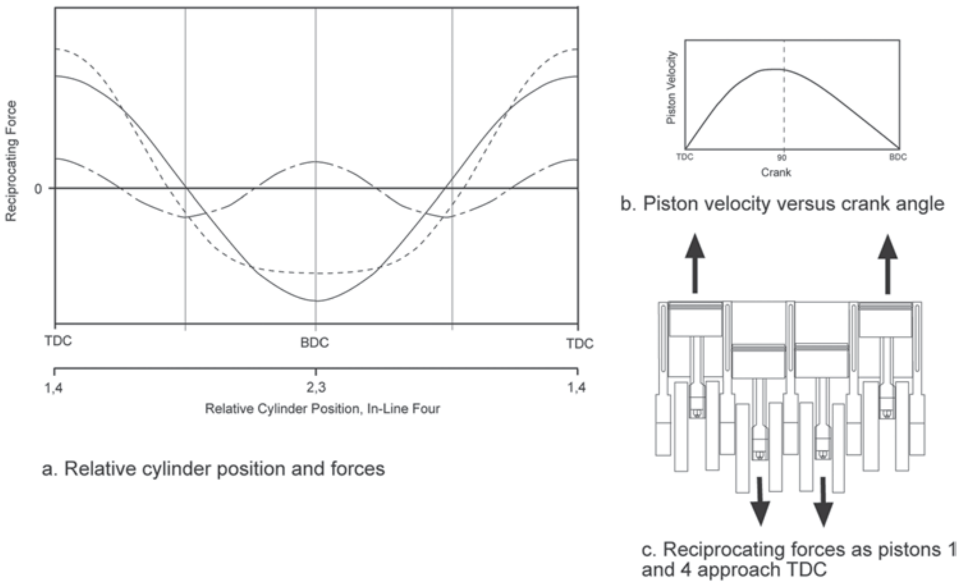

Fig. 6.15 Reciprocating forces in an in-line four-cylinder engine

order imbalance. The physical explanation can be seen with reference to Fig. 6.15b, c. In Fig. 6.15b the velocity profile discussed earlier is plotted. Since the maximum velocity occurred nearer to TDC than to BDC the two pistons approaching TDC were decelerated at a greater rate than those approaching BDC. This resulted in a greater upward reciprocating force than downward. One half revolution later cylinders two and three approach TDC as cylinders one and four approach BDC, and the same force imbalance occurs.

Finally, it should be recognized that the reciprocating forces may result in first- and second-order moments within a bank of cylinders. Depending on the relationship between cylinders within the bank these moments may be counteracted or remain unbalanced.

6.7 Balancing the Forces in Multi-Cylinder Engines

The calculation of rotating and reciprocating forces and resulting moments was summarized in the two preceding sections. These equations can now be applied to consider the balancing of any reciprocating piston engine configuration. Several common examples will be discussed further in this section. The discussion will not attempt to include all configurations that may be of interest for automotive engines. The reader is encouraged to apply the equations to look at configurations that have not been discussed.

The *in-line four-cylinder engine* has been considered in previous examples concerning both rotating and reciprocating balance. It was shown that the rotating forces create opposing dynamic couples that cancel one another out but result in internal crankshaft deflection and bearing wear. Counterweights are used as were shown in Fig. 6.11 to eliminate the

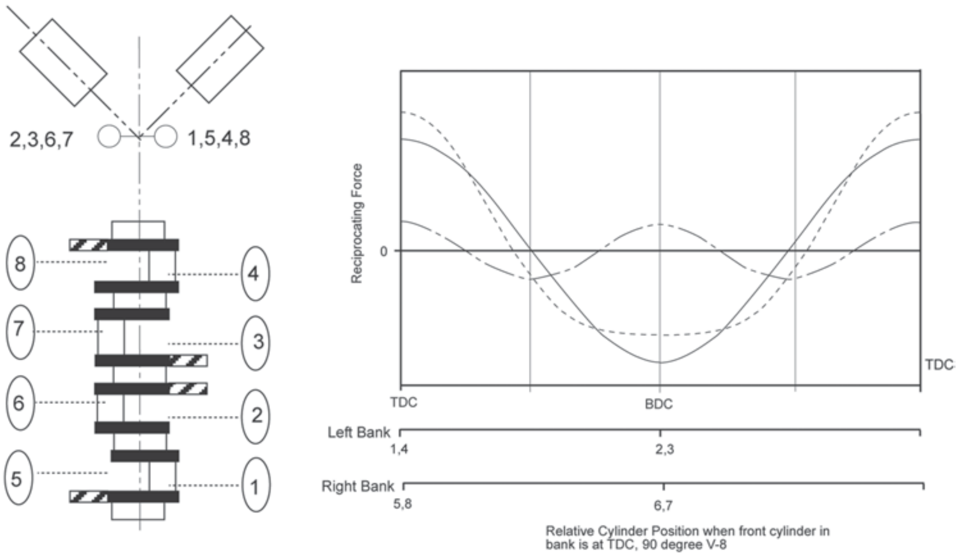

Fig. 6.16 Forces and balance in a 90° V-8 engine using a planar crankshaft

internal couples. The primary reciprocating forces are balanced, but there is an inherent second order imbalance that was explained with reference to Fig. 6.15. The symmetric relationship between the cylinders results in balanced reciprocating moments—both first-order and second-order. In most smaller four-cylinder engines the second-order force imbalance is accepted, and the mounts are designed to minimize its transmission to the vehicle. In order to eliminate the second order imbalance counter-rotating balance shafts are typically used in larger displacement (above two liter) engines. Two shafts are located at the same height in the block, and are driven such that they spin in opposite directions at twice engine speed. Each shaft contains an equal off-centered mass that generates a centrifugal force. Because the shafts are counter-rotating the horizontal components cancel one another out and the vertical components add. They are sized such that the vertical forces add to exactly cancel the second order forces generated by the piston assemblies.

Because the *V-8 engine* consists of two banks of four cylinders each it will be reviewed next. At first glance the forces acting on vee engines would seem far more complicated to analyze, but this is not necessarily the case. While there are now reciprocating forces acting in two different planes each bank of cylinders can be treated individually. If the reciprocating forces can be balanced within a bank they need not be considered further. The V-8 engine almost invariably uses a 90° angle between the two banks of cylinders, allowing even spacing between cylinder events and shared rod bearing throws. Two different crankshaft configurations can be used. The first is a planar crankshaft similar in appearance to that of a four-cylinder engine, but with two connecting rods sharing each throw. This configuration is summarized in Fig. 6.16. Each bank of cylinders appears exactly like the inline four-cylinder engine, and as a result the primary reciprocating forces are balanced within each bank. The two banks cancel the horizontal component of the second

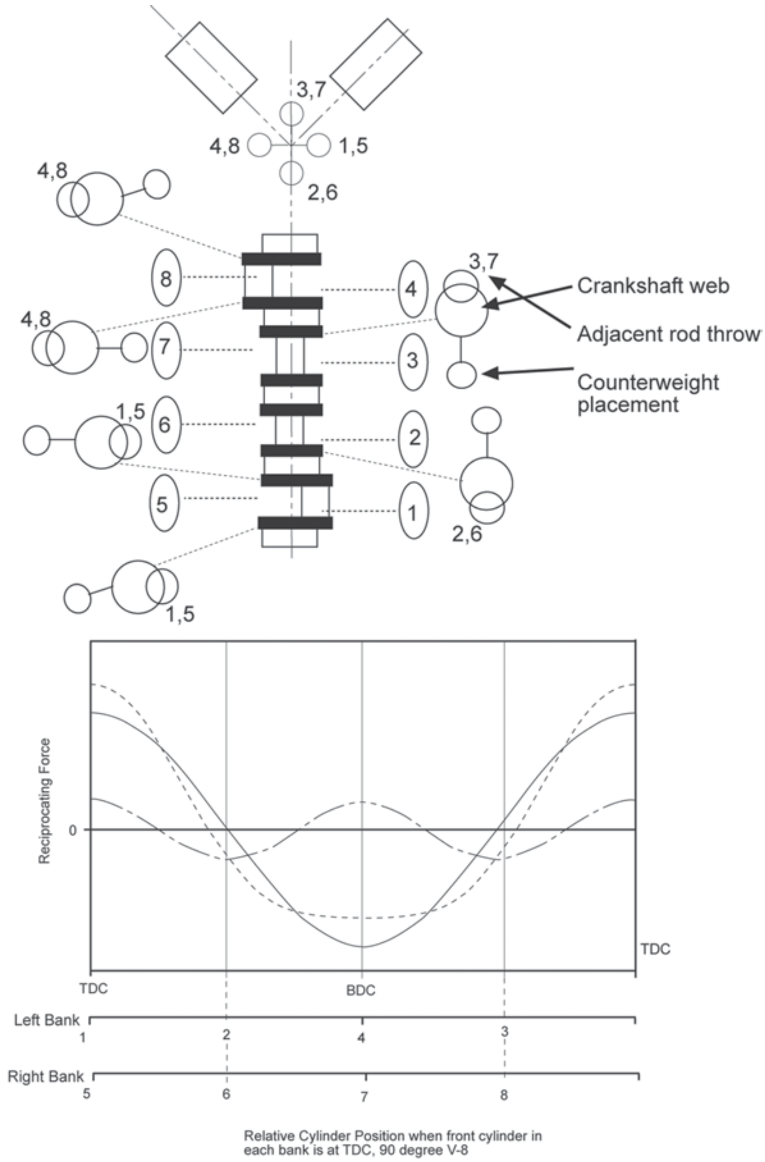

Fig. 6.17 Forces and balance in a 90° V-8 engine using a cruciform crankshaft

order reciprocating force, while the resulting vertical component is 1.414 times that of a single bank. Rotational forces are identical to those of an in-line four-cylinder engine, and the resulting counterweight placement is also identical.

Although it adds crankshaft mass and complexity the more common crankshaft configuration is a two-plane design generally termed "cruciform." This arrangement is shown in Fig. 6.17. The reciprocating force diagram for each bank of cylinders in this engine is

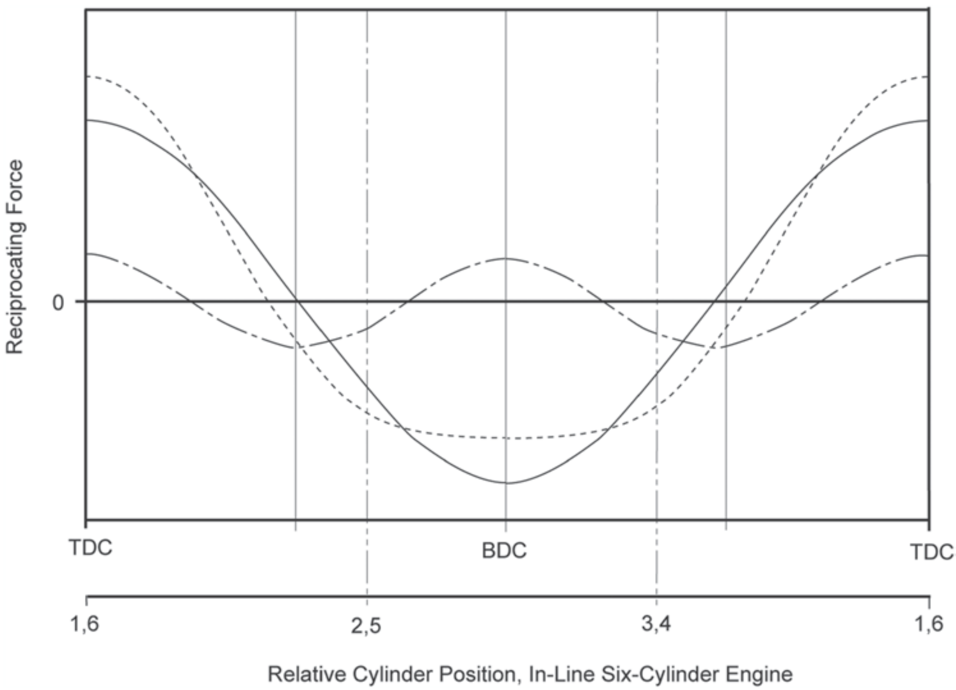

Fig. 6.18 Reciprocating forces in an in-line six-cylinder engine

shown in the figure, and demonstrates both primary and secondary balance. Note that the horizontal axes show each bank at a moment in time when one piston is at TDC, and thus the position of maximum reciprocating force. These positions do not occur simultaneously on each bank, but are shifted in phase by 90°. The rotational forces result in a moment acting over the length of the crankshaft between the first and fourth crankshaft throw and a smaller moment acting across the second and third throw in a plane 90° from the larger moment. The plane in which the resultant moment acts is approximately 18° from that of the first and fourth throw. In order to avoid prohibitively large counterweights at either end of the crankshaft it is typical to place counterweights at each of the throws shown in the figure, acting in the planes of the crankshaft throws. These counterweights are used to offset between one-half and two-thirds of the moment. Balance is then completed with counterweights offset by 18° from the front and rear throws as shown.

Turning now to the *in-line six-cylinder engine* the reciprocating forces can be reviewed with reference to Fig. 6.18. The conventional firing order of 1-5-3-6-2-4 places the throws for cylinders one and six in the same plane. The lower axis in Fig. 6.18 depicts both of these cylinders at TDC. When these two cylinders are at TDC cylinders two and five will be at the position of 120 crank angle degrees, and cylinders three and four will be at 240°. The reciprocating force at cylinders one and six will be at its maximum in the positive direction, while the reciprocating force at each of cylinders two, three, four, and five will be at one-half their maximum in the negative direction ($\cos[120]=\cos[240]=-0.5$). The net first order reciprocating force is thus zero. The net second order force is also found to be zero, since cos2[120] = cos2[240] = −0.5. The rotational forces for this engine were discussed previously, and were shown to result in counteracting internal couples balanced with couterweights centered in a plane 90° from the crank throws for cylinders two and five. This was depicted previously in Fig. 6.13. The symmetric placement of the cylinders about the fore and aft center of the crankshaft results in complete balance of any moments resulting from the reciprocating forces.

The *V-12 engine* will be discussed next because of its similarity to the in-line six. The V-12 consists of two banks of six cylinders. The crankshaft looks exactly like that of the six-cylinder engine, but with each connecting rod throw made to accept two connecting rods. It follows that the engine has first- and second-order reciprocating balance within each bank. The rotational forces generate opposing internal moments identical to those of the in-line six-cylinder engine, again requiring counterweight placement identical to the in-line six. Note that the discussion did not mention the vee angle between the cylinder banks. Since the reciprocating forces in each bank are independently balanced the vee angle has no impact on mechanical balance. Vee angle does impact the firing forces, and this effect will be discussed further in the next section.

Several firing orders are seen with the *V-6 engine* but in each case it is important that the three cylinders within each bank are spaced such that the crankshaft throws are 120° apart. This results in first- and second-order reciprocating balance within each bank. In most V-6 engines the crankshaft has six separate throws, supported on four main bearings, with two throws between each bearing. The front and rear throw are in the same plane, 180° apart, with the remaining throws spaced symmetrically about this plane. The resultant moment is thus in the plane of the front and rear throws, and the counterweights are placed opposite these throws as shown in Fig. 6.19.

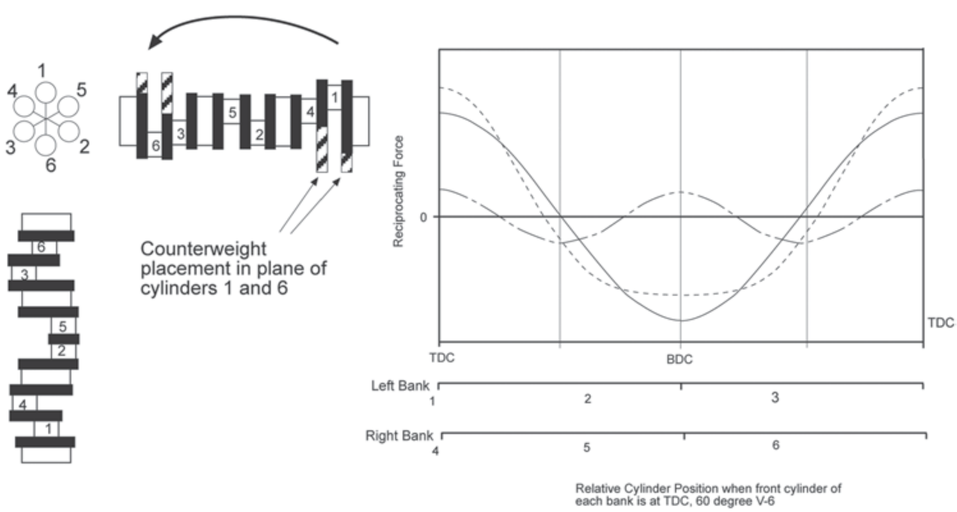

Fig. 6.19 Forces and balance in a V-6 engine

Table 6.3 Balance of reciprocating forces in in-line 5-cylinder engine

Cylinder	First-order	Second-order
1	cos[0] = 1.0	cos2[0] = 1.0
2	cos[72] = 0.309	cos2[72] = −0.809
4	cos[144] = −0.809	cos2[144] = 0.309
5	cos[216] = −0.809	cos2[216] = 0.309
3	cos[288] = 0.309	cos2[288] = −0.809
Sum	0.0	0.0

There are now several examples of both *in-line five-cylinder and V-10 engines*. The firing order commonly used with the inline five-cylinder engine is 1-2-4-5-3, and the cylinders are equally spaced at 72° intervals around the crankshaft. Both the first- and second-order reciprocating forces balance as shown by the following calculations in Table 6.3:

A first-order moment remains. Rotating balance is achieved by equally counterweighting each cylinder to individually balance it along the crankshaft centerline. This results in a relatively stiff, heavy crankshaft. The V-10 engine consists of two five-cylinder banks placed at either a 72 or 90° angle from one another. The 72° angle provides even spacing between firing pulses, but the 90° angle is more common since most V-10 engines are built from modified V-8 block tooling.

The discussion of multi-cylinder engine mechanical balance concludes with a look at *horizontally-opposed engines*. Beginning with the four-cylinder engine, a single plane crankshaft is used as depicted in Fig. 6.20. The throws are positioned identically to an in-line four cylinder engine, and therefore rotational balance is achieved with the same counterweight positions described for that engine. The peak reciprocating forces seen as the pistons approach TDC and BDC are also shown in Fig. 6.20. The two pistons approaching TDC (one in each bank) provide counteracting forces, as do the two pistons approaching

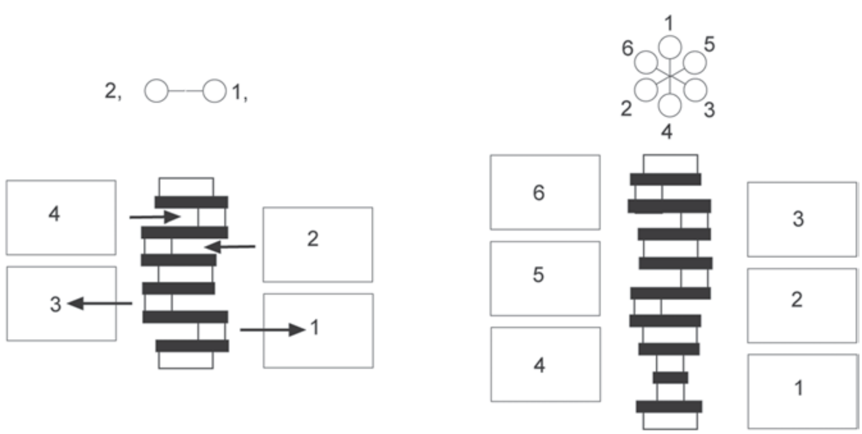

Fig. 6.20 Forces and balance in four- and six-cylinder horizontally opposed engines

BDC. Thus both first- and second-order reciprocating forces are balanced. Because the pistons approaching BDC generate a lower reciprocating force than those approaching TDC a second order moment results.

The horizontally-opposed six-cylinder engine is also depicted in Fig. 6.20. Like the V-6 reciprocating forces within each bank are completely balanced; although the typical firing order is different the throws within each bank are equally spaced at 120° intervals. First- and second-order couples are also completely balanced with this configuration.

6.8 Gas Pressure Forces

Another important category of forces acting internally on the engine is that due to gas pressure within each cylinder. The in-cylinder pressure varies continuously with crank angle, reaching its peak shortly after TDC during combustion. These forces cannot be balanced, and the engine mounts are designed to minimize their transmission into the vehicle. In a four-cycle engine the force generated at each cylinder goes through a complete cycle every two revolutions, and is thus half-order. If all of the cylinders in a multi-cylinder engine contribute equally, and the spacing between combustion events is equal, the resulting vibration order generated by the pressure force is one half the number of cylinders.

The angle between the banks of cylinders in vee engines becomes important when one considers the gas pressure forces. Since one cylinder from each bank shares each crankshaft throw the crank angle spacing between TDC positions in each cylinder pair is the same as the vee angle. Unless further design modifications are made the product of the vee angle and the total number of cylinders must equal 360 or 720° if an even spacing between the gas pressure forces is to be achieved. A 90° vee angle results in evenly spaced firing pulses for a V-4 or V-8 engine. A 60° vee angle provides evenly spaced pulses in a V-6 or V-12 engine. A V-10 engine would require a 72° vee angle.

Since the gas pressure forces cannot be balanced it is legitimate to question whether evenly spaced firing forces are required, and in fact they are not. There are many production examples with uneven firing pulse spacing. In some cases the decision to use a different vee angle is driven by manufacturing costs—a 90° V-6 or V-10 engine is often made based on modifications of an existing V-8 design. In other cases the decision might be driven by package dimensions—a narrow vee angle may be chosen to reduce engine width. In these cases it is most common to accept the uneven firing pulse spacing. Another alternative is to modify the crankshaft throws, offsetting the two shared bearings to maintain even firing. As an example consider the V-6 engine discussed with reference to Fig. 6.19. Even firing requires each firing pulse to be placed 120° apart. Sharing crank throws would require a 120° vee, resulting in a very wide engine. In Fig. 6.19, a web is shown between the two cylinders in each crankcase section, or bay. If the vee angle of the engine is 60° the two throws must be placed 60° apart and an unsupported web, or "flying web" is used to link the throws. On a 90° V-6 the two throws in a given bay are 30° apart. The crankshaft can be manufactured with an offset between the throws, and the flying web is not needed.

6.9 Recommendations for Further Reading

Engine balancing and configuration calculations were established many years ago, and few recent publications cover these concepts. For a recent detailed explanation of these concepts, using a helpful graphical analysis technique, the reader is referred to the following text (see Heisler 1995).

References

Heisler, H.: Advanced Engine Technology. SAE Press, Warrendale (1995)