Engine Balance
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This article is intended to give the reader an appreciation for the processes necessary to build balanced engines.

GENERAL BALANCE INTRODUCTION

"The development of present day balancing technique has been fostered by the demand for high operating speeds and quiet vibration less machinery and equipment. No longer is the quality of high-speed equipment measured by its "power roar" or the associated numbing vibration. Rather, the detrimental effects of vibration have been realized and the balancing of rotating parts has become a widespread practice."

Balance Engineering Manual #31

Balancing improves quality and performance of rotating parts. Balancing is the process of adding or removing weight thereby minimizing vibration, noise and bearing wear. This is done by aligning the natural axis the part wishes to rotate about with the actual axis about which it does rotate.

Balanced engines generally:

Use less fuel	Run more consistently
Generate less friction and heat	Operate more smoothly
Last longer	Exhibit fewer leaks
Produce more horsepower	Have fewer component failures

Most engine spin fast enough to damage bearings, moving parts and cylinder walls if the reciprocating assembly is not balanced. correctly.

Just about everyone has been in a car with a wheel out of balance. The resulting feeling is often vibration, shimmies, and lack of complete control. Imagine this feeling, now partially hidden within the confines of your engine. In many instances (extreme out of balance) you will see or feel the engine vibrations. But most cases of engine unbalance go unnoticed but can lead to its demise.

Any engine that is not balanced or balanced incorrectly will most likely hurt internal engine components.

The speeds in which the internal parts reciprocate creates stress loads, friction, and heat; all of which are trying to tear the engine apart. The smoother the engine runs, the less these negatives have an effect. The parts most affected by this inept balancing are the ones that the engine needs the most to survive. Some components affected by imbalance and their symptoms are:

- Piston Rings ... fail to seal
- Bearings ... early and irregular wear on connecting rod and main bearings
- Damper ... the device that tries to assist in controlling harmonic shock gets overworked and begins to deteriorate.
- Oil Pumps ... Chatter and bounce, which can also create spark chatter and early ignition part failures (oil pumps driven off same drive as distributor)
- Timing Sets ... Early chain stretching as chain has to make up for damper failures
- Valve Springs ... Valve instability, spring harmonic failures (worse with gear drives)
- Transmission **...** Front Pump failures in automatics, early pressure plate and clutch spring failures

DEFINITIONS:

Axis of Rotation: The axis along with a body rotates, usually defined by center position of two or more bearings in a straight line.

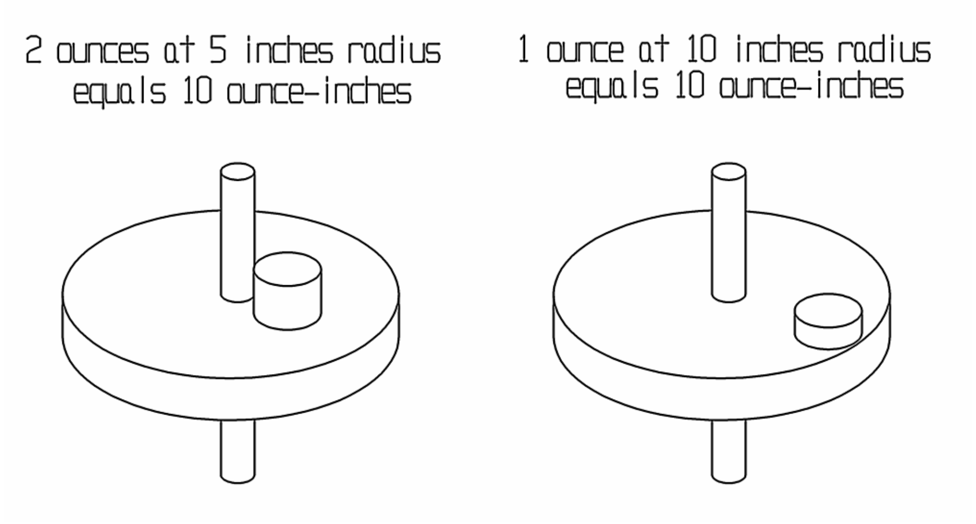
Center of Gravity: The point at which all the mass of the body can be considered to be concentrated. More precisely, the center of gravity is the point at which the sum of the moments of all particles comprising the body equal zero.

Mass Axis: The axis a body would naturally wish to rotate about if it were freely suspended with no restriction. If an object were placed in zero gee, the axis it would naturally rotate around.

Unbalance: Condition where the Axis of Rotation is not identical to the Mass Axis. Since the center of the mass is rotating around the Axis of Rotation, centrifugal force will create a radial force on the body. Unbalance can be divided into 'Static' and 'Dynamic' unbalance which will be described later.

UNITS OF BALANCE:

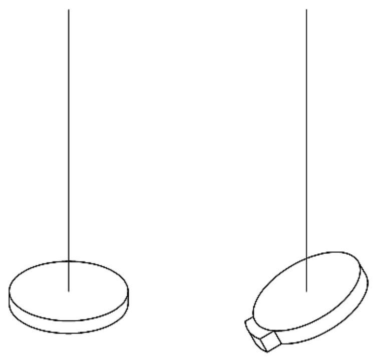
From the definition of unbalance above, we can see that the amount of unbalance is described by the total weight of the object multiplied by the distance of the Mass Axis from the Rotating Axis. This can be given in units such as gram-centimeters, millimeter-kilograms or inch-ounces. If a perfectly round, consistent and uniform disc were spun on its geometric axis, it would be perfectly in balance. Attaching a weight of 2 ounces at a distance of 5 inches from the axis of rotation would create an unbalance of 2 ounces x 5 inches or 10 inch-ounces. A weight of 1 ounce attached 10 inches from the axis would create an unbalance of 10 inch ounces. When dealing with Static Balance, these terms are definitive; Dynamic Balance will require further qualification of terms.



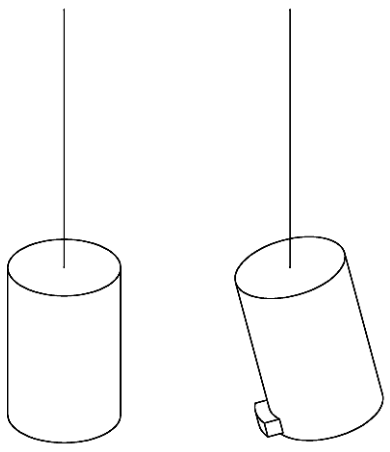
STATIC AND DYNAMIC BALANCE:

Static balance can be measured with a process that does not require revolving the part while measuring dynamic unbalance always requires that the part be rotated. Note that some static balancers do rotate the part, but this is to obtain faster and more accurate results, not because it is inherently necessary to obtain the static balance measurement.

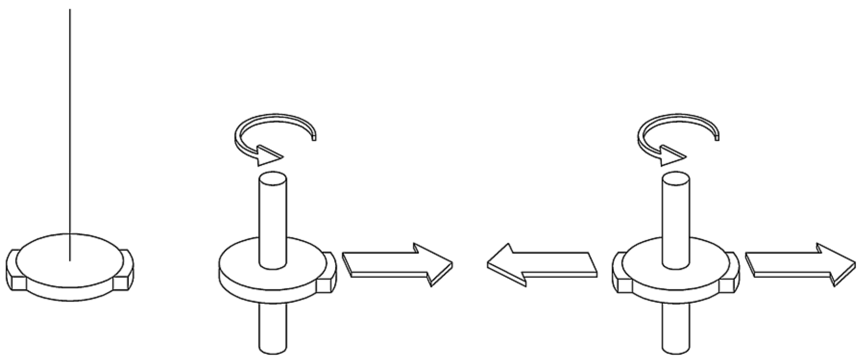
Let us examine the perfectly consistent, uniform and round rotating disc once again. No matter where we place the unbalance weight around the disc, it is always essentially in the plane of the disc (or as referred to in balance engineering, the rotor). Any unbalance of this rotor will occur in a single plane. Therefore, a single sensor can measure the amount of centrifugal force developed in this plane and determine the angle this force is developed at. If we suspended the disc by a string from the geometrical center, it would rest perfectly level and be in balance. Attaching a weight anywhere along the disc will cause the disc to rest with the weight inclining towards the ground and would indicate a static unbalance. This is a case of static balance measurement.



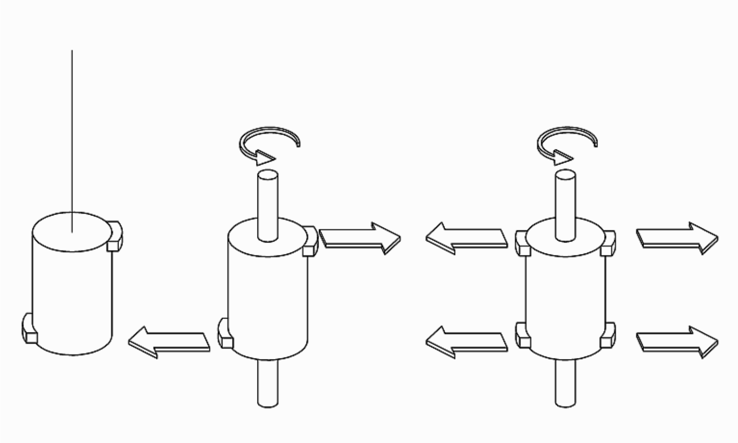
Now let us examine a perfectly consistent, uniform and round cylinder. If we suspend it from the same string on the center of the face it will also rest perfectly level and it will also incline if we add a weight to one side. The key difference here is that with the disc we can only add weight in one plane, that of the disc, while on the cylinder we can add weight anywhere along its length. The first cylinder shown below is statically balanced while the second is statically unbalanced.



Returning to our imaginary disc, if we were to add identical weights directly opposite each other at the same distance from center, the disc would still hang level and be statically balanced, as the two weights would offset one another. Were we to rotate the unbalanced disc, it would wish to wobble. If we rotated the balanced disc, there would be no centrifugal forces trying to cause it to wobble since the weights would equally offset each other.



Now let us examine the cylinder. If we suspend the cylinder from a string and put one identical weight on the top end and another identical weight on the bottom end, but place these two weights absolutely opposite one another, then the cylinder will still hang level. The cylinder is statically balanced. If we put the cylinder on a shaft and rotate it, we see a different phenomenon. The weight on the top end will create centrifugal force in one direction and the bottom weight will create centrifugal force in the opposite direction. The two forces create a rotating **'COUPLE'** and try to twist the cylinder around its axis as it rotates. (A couple is when the unbalance forces on each of a part are not identical. Couples can be rotating or rocking.) As we can see in the drawing below, static balance measurement is not sufficient in this example to assure a smoothly rotating part. In cases like this, where the mass can be distributed along the length of the axis, we resort to dynamic balance. Dynamic balance measurement is achieved by designating TWO planes at which to measure the unbalance and the angle at which it occurs. The last figure shows the cylinder in a state of dynamic balance.



Static Unbalance Measurement can be considered to be **One Plane Measurement** while **Dynamic Unbalance Measurement** can be defined as **Two Plane Measurement**. In practical terms, rotating items that have most of their mass distributed in a plane (such as a flywheel) can be statically balanced, whereas components that have mass distributed along their length (such as crankshafts and drive shafts) must be dynamically balanced.

ENGINE UNBALANCE

In the above introduction we looked at how rotating forces cause unbalance. In a piston engine we also have reciprocating masses that create their own forces. The following is a brief discussion of the nature of these forces and approaches for dealing with them.

The parts of an engine that move in a reciprocating fashion are the piston, rings, bushings or bearings and the upper part of the connecting rod. As these masses accelerate along the cylinder bore they create **INERTIA FORCES** that react with the engine structure. **INERTIA BALANCING** is the process of selecting cylinder and crankshaft configurations in such a way that the inertia forces largely cancel one another out. The complete cancellation of these forces is known as **INHERENT BALANCE** and exists in such engines as V-8s and Inline 6s.

Inertia forces propagate along the line of piston travel, and they fluctuate in force and direction proportionately to the degree of crankshaft rotation.

The **PRIMARY SHAKING FORCE** changes direction every revolution and is caused by acceleration and deceleration of the reciprocating masses. The force takes the form of equation:

$$C = .0000284\ WRN^2$$

where

C= Inertia Force (pounds)

W = Weight of reciprocating parts in pounds

R = Crank Radius (crank throw) in inches

N = rpm

The primary shaking force reaches its peak at the top and bottom of the piston travel. For intermediate positions the force is equal to the inertia force calculated above times the cosine of the crankshaft angle.

As you can see from the above, the force (C) goes up as the square of the rpm (N). A typical tolerance for a crankshaft would be 0.25 ounce-inches maximum allowable unbalance. The following table describes the force this applies to the engine as rpm increases.

RPM	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000
FORCE (LBS)	0.44	1.77	3.98	7.08	11.06	15.93	21.68	28.32	35.84	44.25

The **SECONDARY SHAKING FORCE** is one that changes direction twice every revolution. It is caused by the fact that piston acceleration is not regular. When the piston has traveled half the length of the cylinder, the crankshaft has not yet rotated 90 degrees. The amount of this distortion is proportionate to the ratio of the connecting rod length to the crankshaft throw. The formula for the secondary shaking force is:

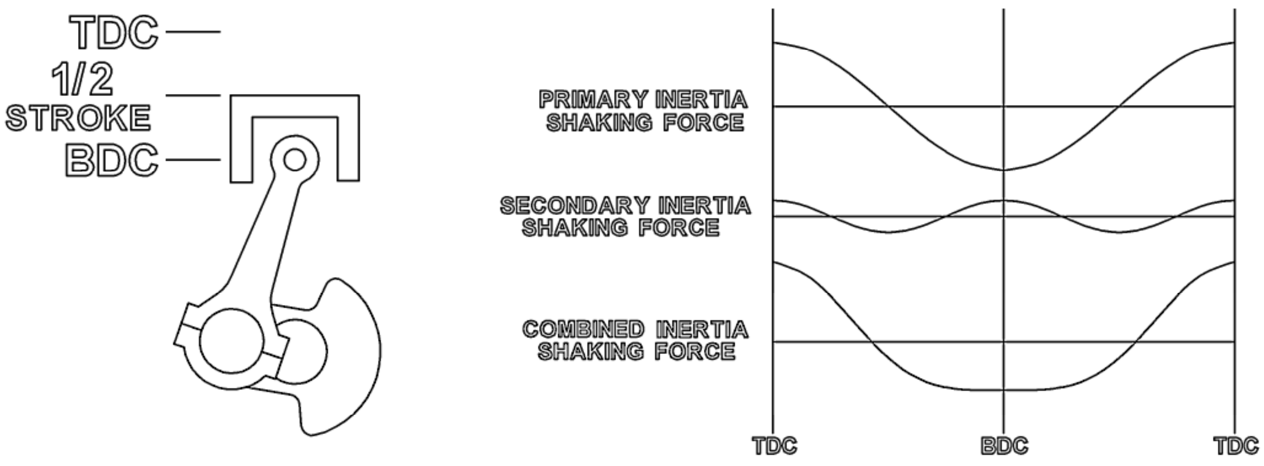
$$C_s = PC$$

P = Crankshaft Throw/Connecting Rod Length

The secondary shaking force also reaches peaks at TDC and BDC as well as at 90 and 270 degrees of rotation. Intermediate forces can be determined by multiplying the secondary shaking force by the cosine of 2 times the crankshaft angle.

SINGLE CYLINDER ENGINE

In a single cylinder engine the primary shaking force assumes the value of a cosine wave along the line of the piston travel. The secondary shaking force also assumes a cosine wave, of smaller value, and twice the frequency. The two forces interact to form a ‘lumpy’ shaking force along the line of piston travel**.**

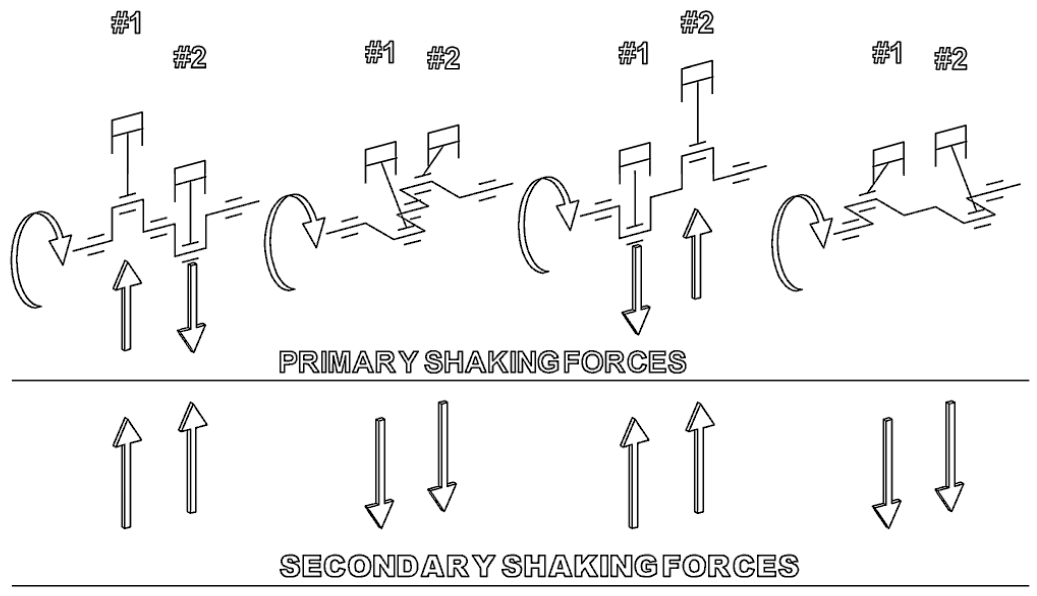


TWO-THROW SHAFT

For our second example, let us look at an inline two-cylinder engine such as used on some small engines. They comprise two crankshaft throws spaced 180 degrees apart.

Since the primary shaking force reverses every 180 degrees, when piston 1 is at TDC it creates a primary upwards shaking force of 1. At the same time piston 2 is at BDC and it creates a primary downwards shaking force of 1. The **total primary shaking force** on the engine is zero, since the two forces are equal in opposite directions; neither suffices to actually move the engine from its rest position. On the other hand, piston 1 lifts the left side of the engine upwards while piston 2 pushes the right side downwards. This leads to a **primary rocking couple** equal to the primary shaking forces multiplied by the distance between cylinder centers. The secondary shaking forces propagate at twice the frequency of the primary. They are peak at TDC, zero at 45 degrees, peak in the reverse direction at 90 degrees, zero at 135 degrees and peak again at BDC in the original direction. Therefore the engine has a **total secondary shaking force** equal to that of both combined cylinders, or 2PC. As the engine rocks back and forth every revolution, it will also bounce up and down twice every revolution. The drawing below shows a two throw crank as it rotates in 90 degree increments. The arrows directly below each crank throw illustrate the primary shaking force for that piston while the arrow beneath that illustrates the smaller secondary shaking force for that piston.

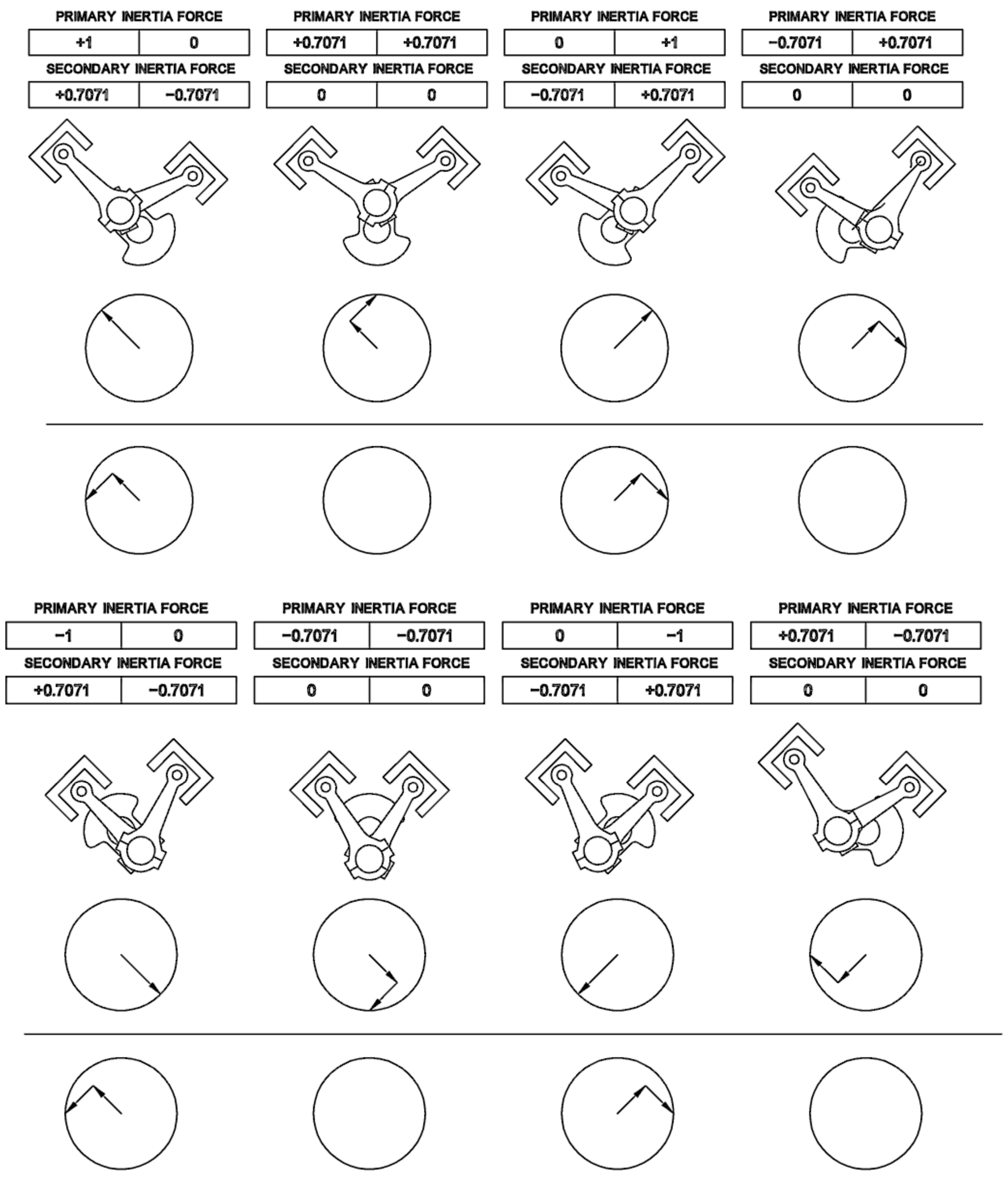
In the drawing below, the secondary shaking forces are smaller than the primary forces by the factor "P", the crankshaft throw divided by the connecting rod length. They are illustrated as having the same magnitudes in the drawing below and in future drawings since rendering the secondary forces to scale would cause them to be difficult to read.



90 DEGREE 2 CYLINDER

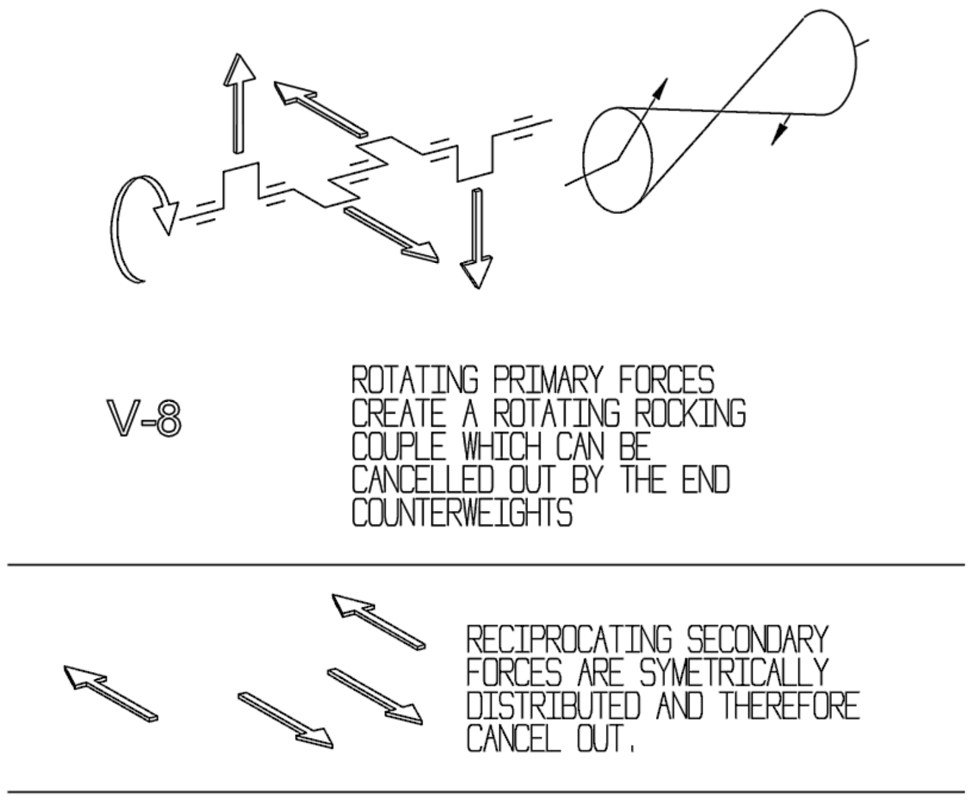
This configuration is the basis of the popular V-8 engine and many V-6s. By examining the force diagrams, below, you can see how the primary shaking forces in both cylinders cancel each other out in such a way as to cause a **rotating shaking force** that propagates at the crankpin angle. This is actually quite advantageous because such a rotating force can be readily canceled by placing additional mass in the counterweight sufficient to generating an opposing force. For all practical purposes, the 90-degree vee should have no primary shaking force if properly designed and constructed. By the same token, the secondary shaking forces also interact in an interesting manner. In a 90-degree V-2 they combine to act as though a single cylinder engine. The secondary shaking force propagates along the line perpendicular to the plane bisecting the angle of the cylinders and has a frequency twice that of the engine rpm. The drawings below illustrate the forces on a 90 degree V-2 bank in 45 degree rotation increments.

Note that for each instance the upper vector force diagram (circle drawing) represents primary inertia forces and the bottom diagram the secondary forces. Eacb piston is shown with its own arrow representing force. The radius of the circle itself is equivalent to one unit of the applicable inertia force. The secondary shaking forces are smaller than the primary forces by the factor "P", the crankshaft throw divided by the connecting rod length.



V-8

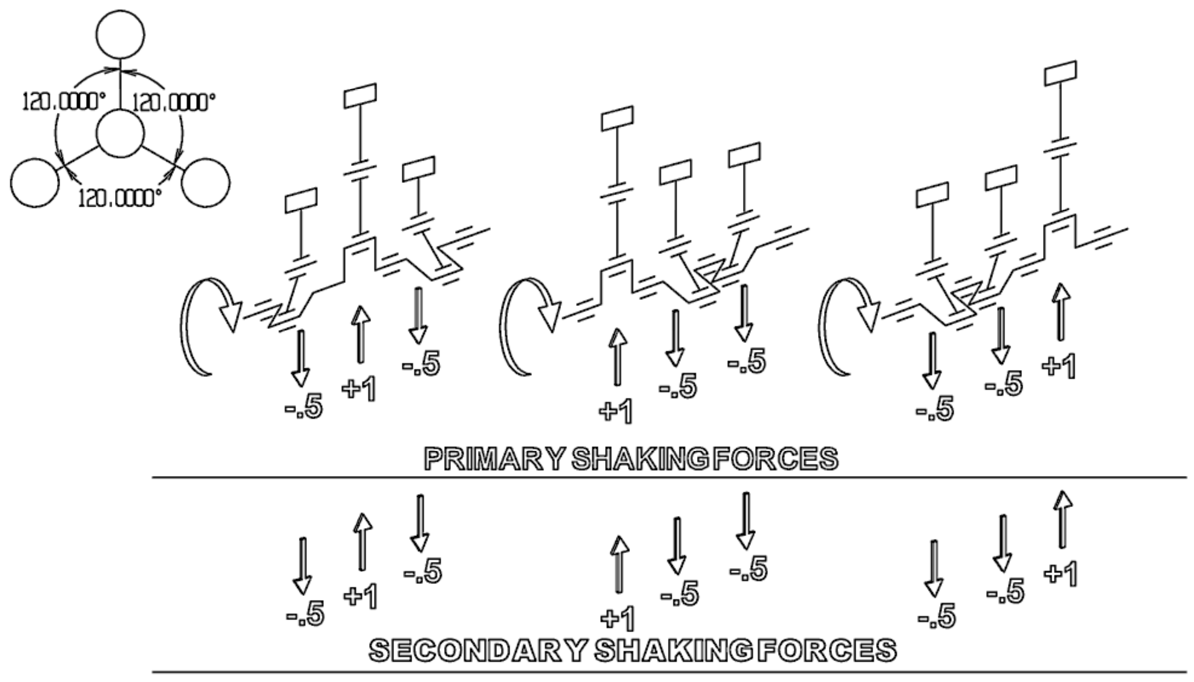
From the above example, it is easy to work out how a V-8 engine is inherently balanced, that is balanced so that all the inertia forces effectively cancel out. Since each pair of V-2 cylinders has no primary shaking or rocking couples, there is no primary shake in any engine using 90-degree Vee banks. This leaves the secondary shaking forces. In a 90 degree Vee the crankshaft throws are placed at 0, 90, 270 and 180 degrees, as you look down the crank, or something equivalent depending on direction of view and rotation. Secondary shaking forces reverse every ninety degrees, and we have determined that each 90-degree Vee bank acts as though it were a single cylinder as far as secondary forces are concerned. Therefore, if #1 crank throw were at TDC the secondary shaking force would be an horizontal value of 1. Throw 2, being 90 degrees away from throw 1, would have a horizontal value of -1. Throw 3, also being 90 degrees away from throw 1 in the opposite direction would likewise have a horizontal value of -1. Throw 4 being 90 degrees from Throws 2 and 3 would have an equal and opposite horizontal value of 1. This gives us an engine with an upward secondary shaking force of 1 on each end and a downward shaking force of 2 in the middle. The upward and downward forces are equal so there is zero secondary shake to the engine. Likewise, the upward and downward forces are symmetrically distributed along the shaft so there is no secondary rocking couple. With all forces being cancelled, we can say this engine is **inherently balanced**.



INLINE 3 CYLINDER

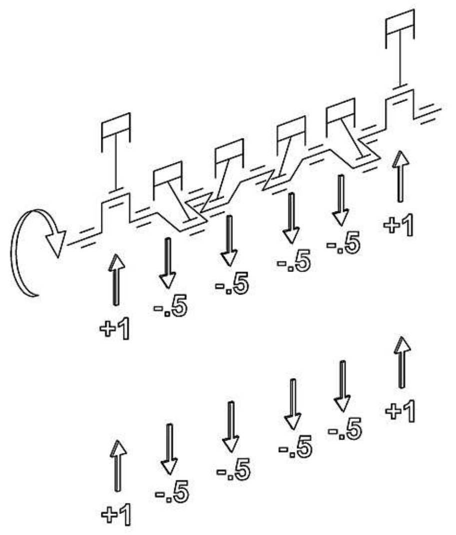
This engine will have 3 crank throws spaced 120 degrees apart. Whenever any piston is at TDC the other two pistons will be 120 degrees ahead or behind. The cosine of 120 degrees is –0.5. The combined primary shaking force of the two pistons at 120 degrees from TDC will equal the upward shaking force of the TDC piston. Therefore there is no primary shake to this engine. When the center piston is at TDC, the equidistant spacing of the two outer pistons cancels out any rocking couple. When the piston at either end is at TDC, however, the center and far end pistons both create a downward shaking force. We can therefore conclude that inline 3 engines have a primary rocking couple once per revolution in the cylinder plane.

The secondary shaking force occurs at twice the frequency of the primary. As it so happens, the cosine of twice the crankshaft angle returns much the same values as for the primary shaking force. Therefore, inline 3 cylinders have no secondary shaking force but do have a secondary rocking couple twice per revolution in the cylinder plane.



INLINE 6 CYLINDER

The inline 6-cylinder engine is basically two inline three cylinders arranged back to back. This engine is inherently balanced because both the primary and secondary rocking couples for each 3-cylinder segment are symmetrically distributed and thus cancel each other in much the same fashion as the secondary couples cancel in a V-8.



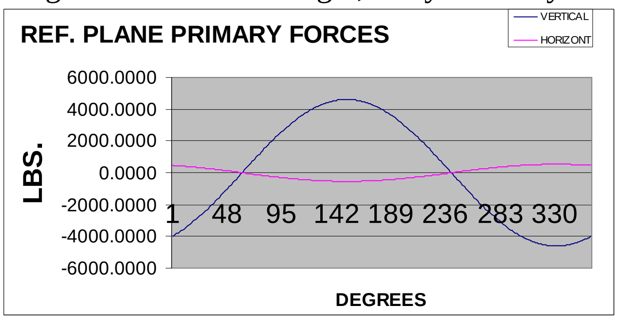
60 DEGREE V-2

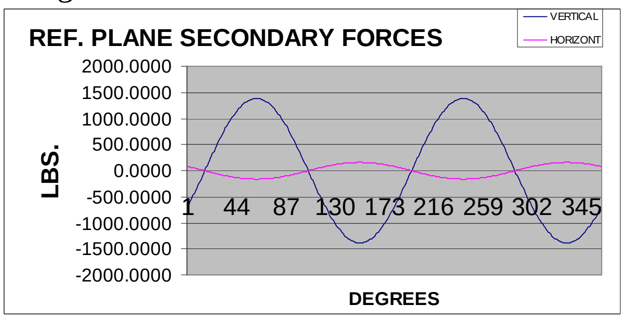
When pairs of cylinders are arranged in a 60-degree Vee bank, something interesting occurs. The combined primary shaking forces resolve out as a single shaking force with a frequency of one cycle per revolution propagating along the angle that bisects the two cylinders. The secondary shaking force resolves out as a **rotating** force that revolves twice per crankshaft revolution. (Actually, this isn't so since the pistons are displaced by the thickness of a connecting rod, but it's a useful fiction). Some very successful V-6 and V-12 engines have been built around 60 degree Vee banks since the secondary forces are mostly cancelled and the primary forces can be readily dealt with using auxiliary balancing techniques covered below.

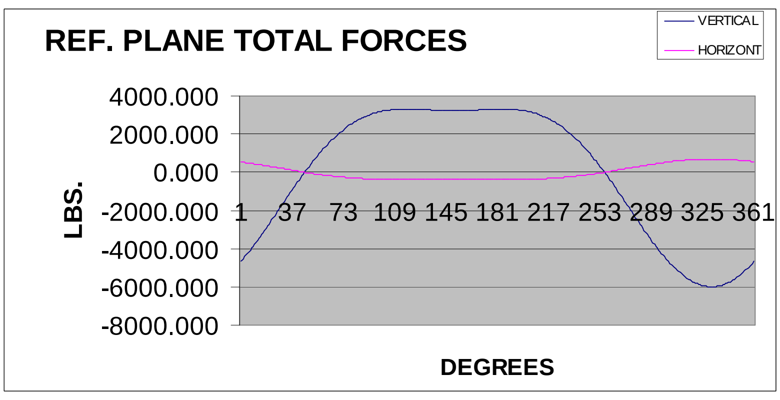
60 DEGREE SPLIT PIN V-2

GM has built a number of 'split pin' V-6 engines over the years. Rather than having two connecting rods on the same crankpin as in a typical Vee engine, the crankpins are 'split' by the angle of the cylinder bank. Since split pins are used on 60 degree V-6 gas engines (and upcoming V-8 diesel), the pins are 60 degrees apart.

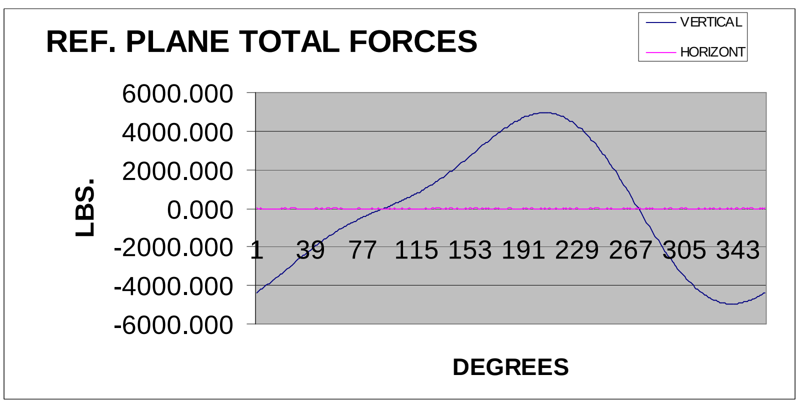
The graphs below were taken from a spreadsheet designed to calculate the unbalance forces on various engine configurations. As you can see, the horizontal primary and secondary forces are negligible. There are significant primary and secondary vertical forces, but something interesting happens when we look at the bottom chart. Primary and secondary forces are both sine waves, with the secondary propagating at twice the rate of the primary. They are out of synch enough, however. The result is that the peak total effect of the forces is less than that of the primary force, so some cancellation is occurring. The combined force diagram has only one peak per period and is thus susceptible to reduction simply by adding one half the reciprocating weight to the counterweight, or by auxiliary balancing means such as balance shafts.



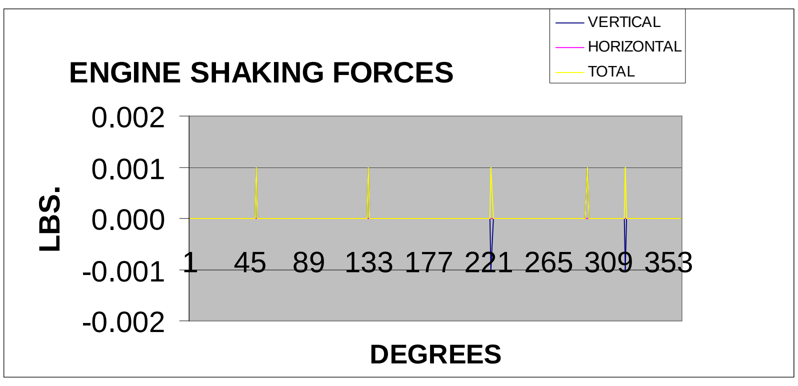




In a split pin V-6 the combination of the shaking forces on each end is shown below, with the two ends 180 degrees out of phase.

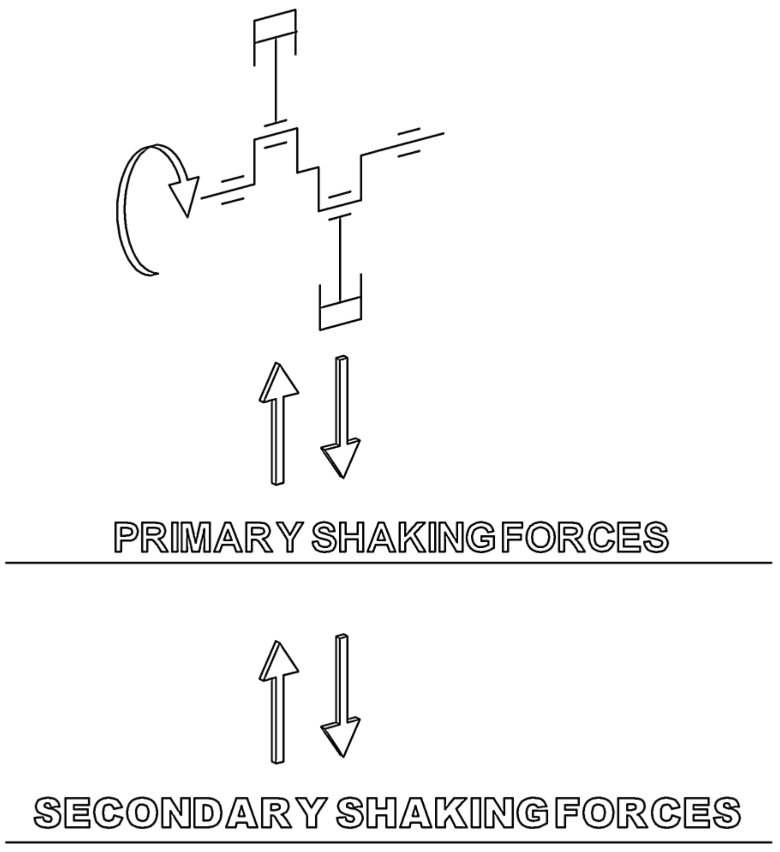


The result is that the split pin V-6 has no primary or secondary shake but does have a vertical rocking couple.



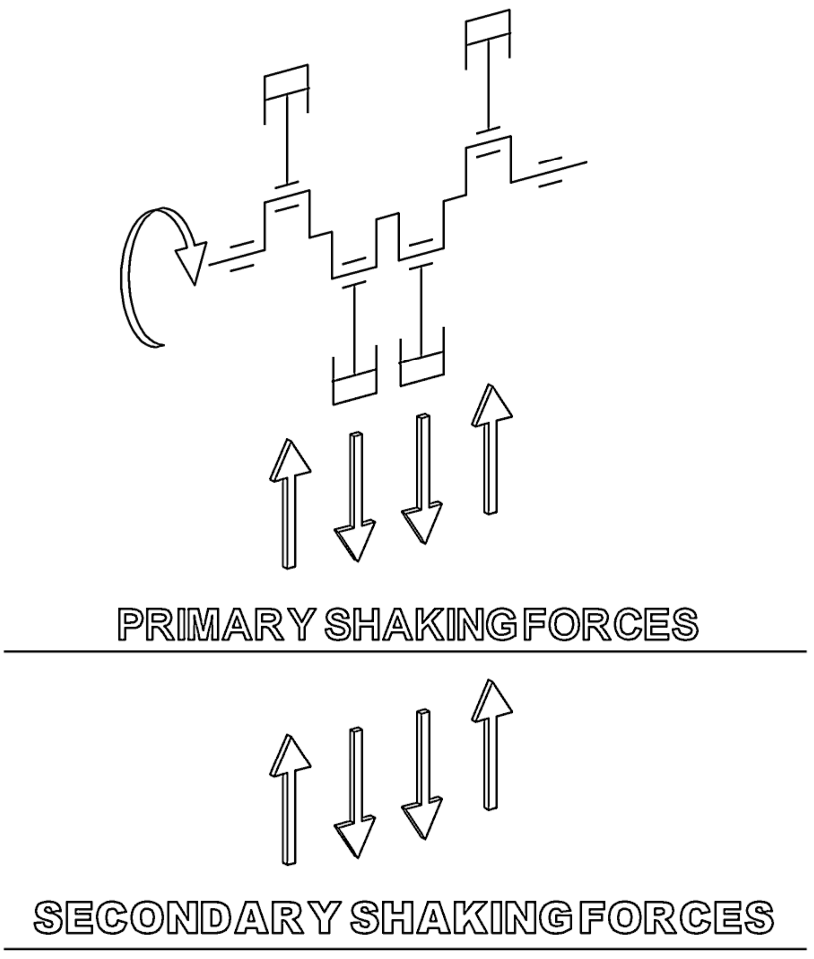
2 CYLINDER BOXER ENGINE

Or the two cylinder horizontally opposed engine. Both cylinders are at TDC at the same time, so both have a primary shaking force of 1. Since the cylinders are in opposite directions, the shaking forces cancel out, leaving no total primary shake to the engine. There is a significant primary rocking couple since the crank throws are located alongside each other rather than coaxially. The situation is exactly identical for the secondary rocking forces. Since both cylinders are at TDC these forces also cancel out the secondary shaking force, but produce a secondary rocking couple twice per revolution.



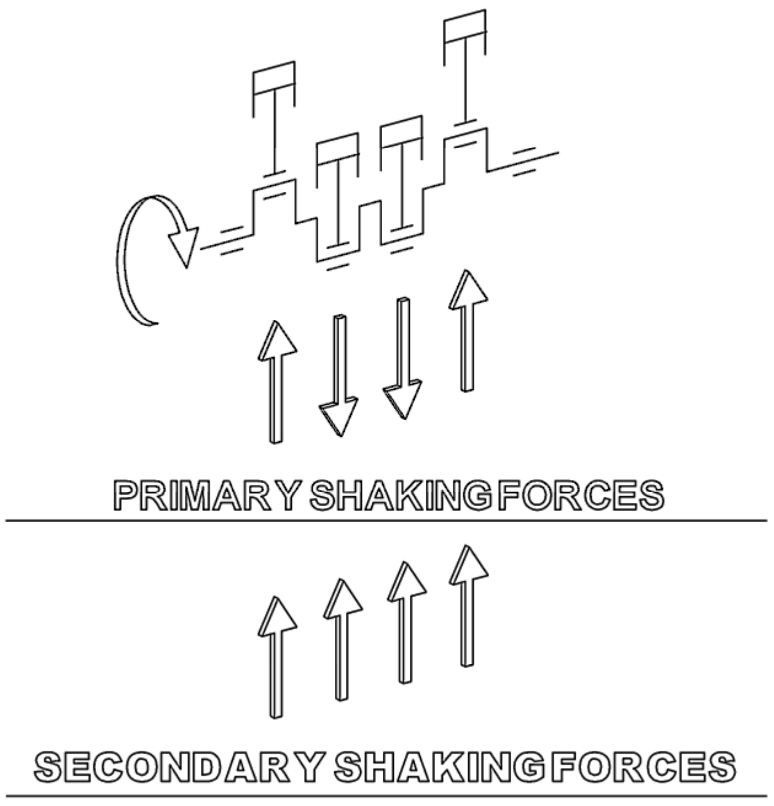
4 CYLINDER BOXER ENGINE (VW-SUBARU-LIGHT AIRCRAFT)

The situation with 4 cylinder horizontally opposed engines is reminiscent of the inline 6 and V-8 engines. We can consider it to be a pair of 2 cylinder boxers mounted back to back. The primary and secondary rocking couples for each pair of 2 cylinder engines cancel each other out, leaving an inherently balanced engine.



INLINE 4 CYLINDER

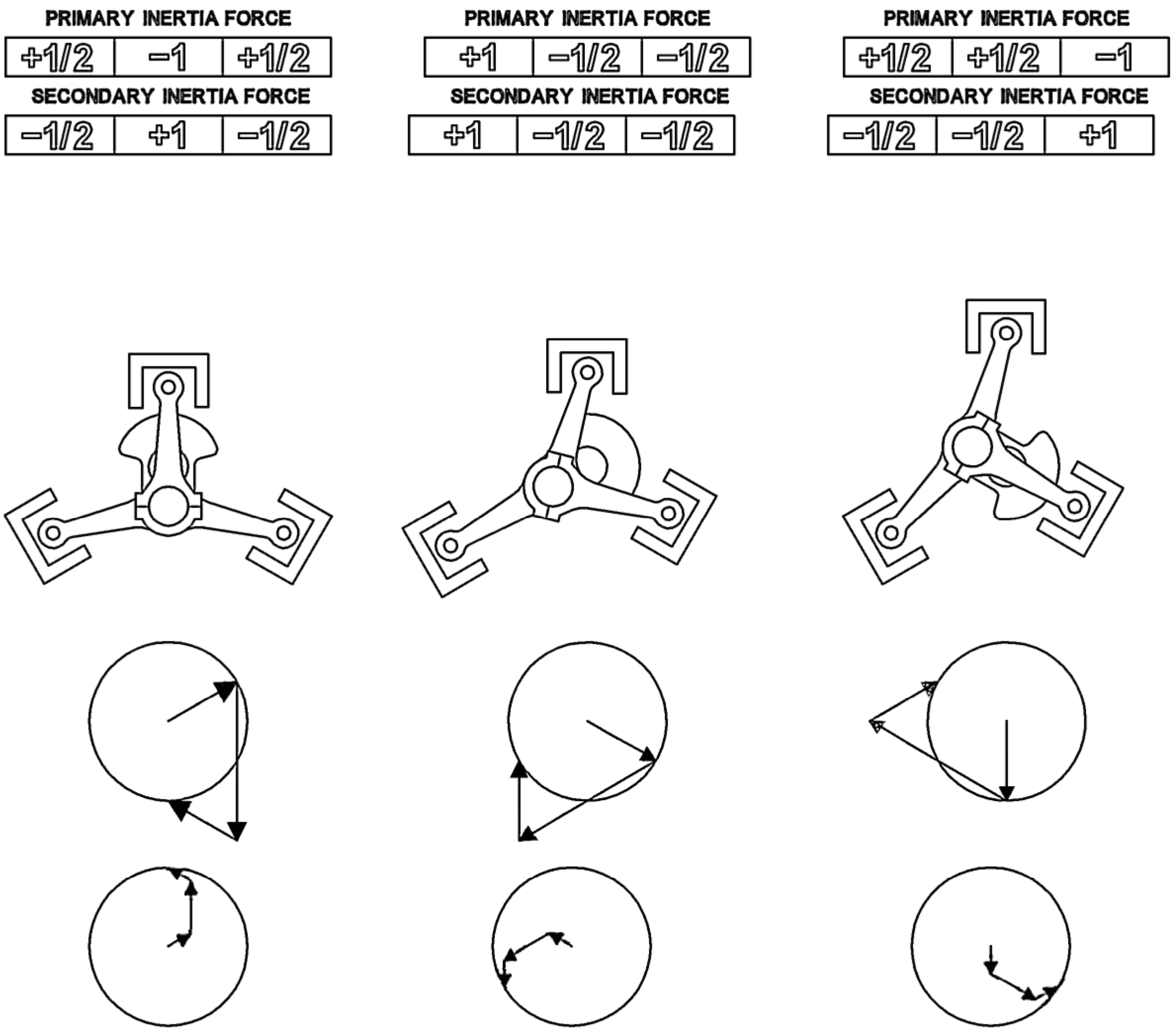
This engine is essentially a pair of inline 2 cylinders mounted back to back. As we demonstrated earlier, this crank has a primary rocking couple and no primary shaking force. Both cylinders contribute to a secondary shaking force. By placing these cranks back to back we can cause the primary rocking couples to cancel out. This leaves us with an engine having no primary shaking force or rocking couple. All four cylinders, however, contribute fully to a secondary shaking force.



RADIALS

In all the radial engine geometries I have examined, primary shaking forces resolve out as a rotating force. This can readily be cancelled out with appropriate extra mass in the engine counterweight. For all practical intents and purposes radials have no primary shake or couple.
In 3 cylinder radial engines the secondary shaking forces resolve out as a rotary force propagating at twice engine rpm, but in the opposite direction of crankshaft rotation. When more

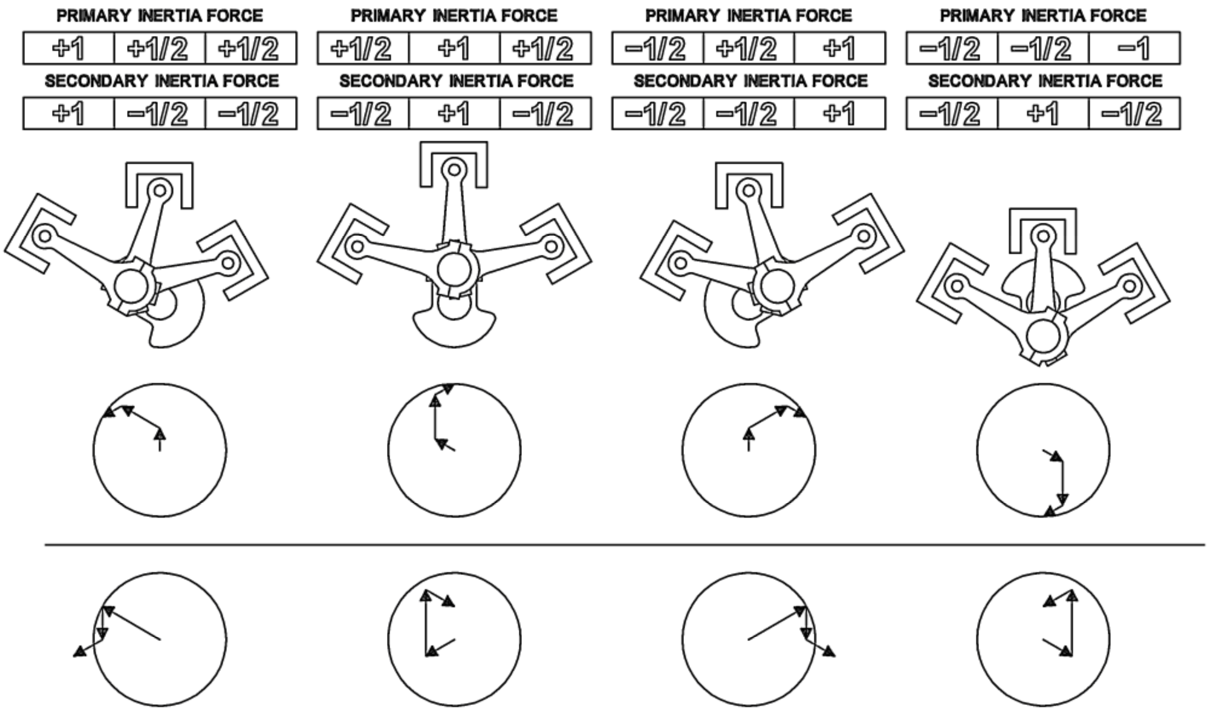
than 3 cylinders are used, secondary forces also cancel out entirely creating an inherently balanced engine.



In the interests of accuracy it should be noted that other radial configurations that I have examined exhibited no secondary shaking forces. The three cylinder radial is unusual in that the force is rotary and also OPPOSITE the rotation of the primary force.

60 DEGREE W ENGINE

I've tossed this in because some air compressors use this scheme, Abner Doble seems to have used it on a successful compound for Sentinel trucks before WW2, VW has played around with a W-12 configuration and a source tells me Ward manufactured an effective marine steam triple in this configuration. In a double acting configuration the power pulses are evenly spaced, yet the engine occupies a much smaller volume than a 3 cylinder radial. Without going into the mathematics, suffice it to say that the primary shaking forces resolve out into a constant rotating force. One again, this is good news in that we can cancel it with the counterweight, making the 60-degree W engine balanced for primary shaking forces. The secondary forces resolve into both horizontal and vertical components at a frequency double that of the rpm, with the component moving in the plane of the center cylinder being the smaller of the two forces. The vector solution for the secondary forces would be an elliptical force moving in the direction of the crank travel but at twice the speed.

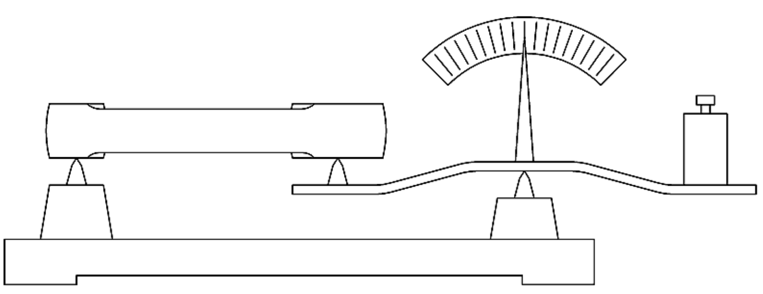


BALANCING THE ENGINE

Up to this point we have covered some fundamental concepts related to balance, defined some terms and taken a general look at the inertia balance of various engine geometries. Now it is time to take a look at the processes required to actually balance an engine. As they say on TV, "Do not try this at home, these are trained professionals".

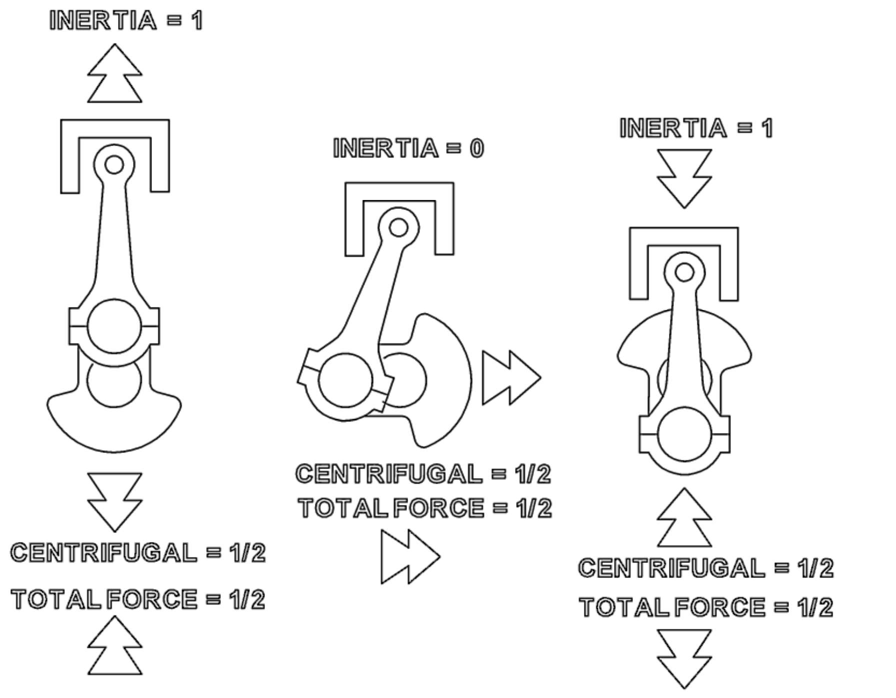
Crankshafts are balanced using a machine called a dynamic balancer. Any reasonable description of how this works would take more space than this article can likely tolerate, as it is already becoming something of a novel. Suffice it to say that this machine spins the crank in a cradle that is mounted on flexible steel reeds or some other similar flexible suspension arrangement. Accelerometers on each end of the cradle measure side to side motion created by the centrifugal forces developed by the crankshaft and through computer wizardry separate these measurements into unbalance force readings at each of the two balance planes on the crank. Before computers there were various mechanical and electrical means of achieving the same results, but computerization has boosted ease of operation, accuracy and reliability far above earlier methods.

When balancing the crank, we have to look at two different masses in the engine, the rotating masses and the reciprocating masses. Reciprocating masses include the piston, rings, wrist pin, and upper end bushing and so on as they all move back and forth. The big end connecting rod bearings are a rotating mass, as they go round with the crank instead of up and down. The connecting rod presents a challenge. The big end goes round and the little end goes up and down. Is it reciprocating or is it rotating mass? The answer, of course, is 'yes'. Part of the connecting rod mass is rotating and part is reciprocating. To determine which part of the weight is which, the rod is weighed with one end on the scale and the other supported on the same level as the scale pan. The weight recorded when the crankshaft end is on the scale is considered rotating mass and the weight when the piston end is on the scale is considered reciprocating mass.



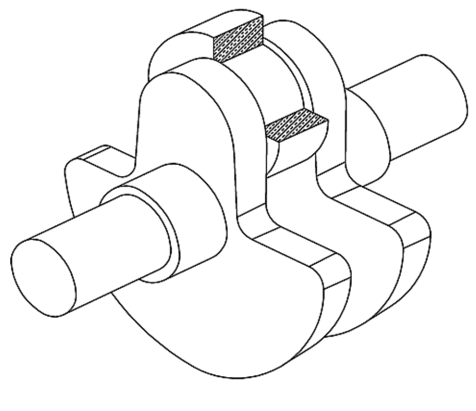
MEASURING ROTATING WEIGHT OF CONNECTING ROD

Balancing rotating masses is easy; you just add a mass to the opposite side that will create an identical force but in the opposite direction. Canceling out reciprocating masses is more difficult. The rule of thumb developed is that a force equal to one half that created by the reciprocating mass added to the counterweight directly opposite the crankpin is the optimal solution. Logically, this makes some sense. As the crank rotates, this mass creates a rotating outwards force. At TDC and BDC it cancels out half the force of the reciprocating mass. By the same token, when the crank rotates 90 degrees it adds a horizontal shaking component that was not present before. The result is that the engine shakes in all directions, but the peak shaking force is only half as extreme as it would be otherwise. Since the peak forces cause the most damage, this is a fair improvement in our situation. Most light single cylinder engines are balanced to this standard.



ONE-HALF RECIPROCATING MASS USED TO REDUCE PEAK INERTIA FORCE

The question now is, "how do you balance the crankshaft in such a way that the counterweight is appropriately heavy enough to cancel out the rotating mass plus one half the reciprocating?" This balance is achieved by employing a device called a bob weight or ring weight. These are rings or other devices that clamp onto the crankpin. These weights are designed so that their center of gravity is in the dead center of the opening so that rotating the ring has no effect on where the weight falls on the crankpin. They are also carefully adjusted so that they weigh exactly as much as the rotating mass plus one half of the reciprocating mass. Typically, crankshaft counterweights are made larger than necessary so that they can have holes drilled in them to make them lighter until the crank is bought into balance.



SINGLE THROW CRANKSHAFT
WITH RING WEIGHT

When a crankshaft with ring weights affixed to the crankpins is spun in the balancer, the machine gives a reading that reflects the crank balance with the appropriate rotating and reciprocating masses applied. The operator then drills out the counterweights until the crankshaft is bought into perfect balance. When the ring weights are removed, the counterweights are now precisely heavy enough to create a canceling force equal to one half the reciprocating mass plus the rotating mass. Production processes do not use ring weights but employ masses attached to the machine rotating assemblies that are carefully engineered to produce the same dynamic results as the ring weights. This eliminates attaching and removing the ring weights and greatly speeds production.

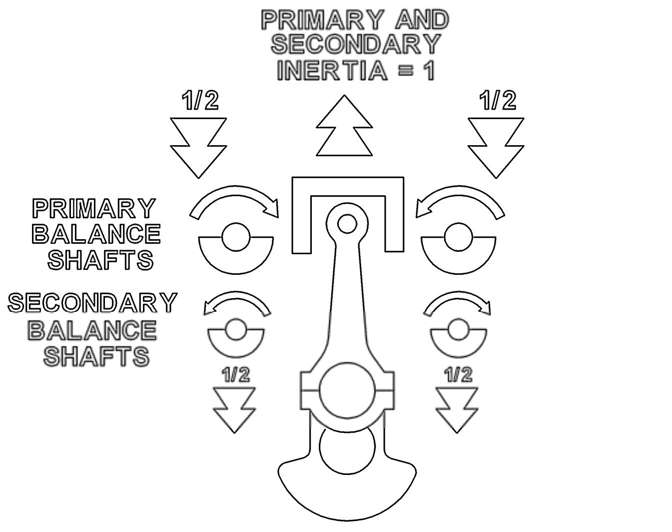
AUXILIARY BALANCING

Up to this point we have seen that by selection of correct crankshaft and cylinder geometry some engines such as V-8s and I-6s can be inherently balanced for both primary and secondary forces. We have also seen that other engine geometries can either cancel one or the other force out, or perhaps partially cancel a force. The I-4 has no primary forces but a full set of secondary shaking forces. The single cylinder engine has both primary and secondary shaking forces acting upon it.

By adding extra masses to the engine, it is possible to partially or completely cancel forces in engines with geometries that are not inherently balanced. The single cylinder engine is perhaps the worst victim of all, so let us look at how shake can be eliminated in a simple one lung engine.

The most common means of vibration cancellation are **balance shafts**. Generally speaking, they are shafts with offset weights that are spun either at engine speed to cancel primary forces or at twice engine speed to cancel secondary forces. In a single cylinder, the primary shaking force is along the direction of piston travel. At TDC it would be desirable to have a peak canceling force developed towards the bottom of the cylinder.

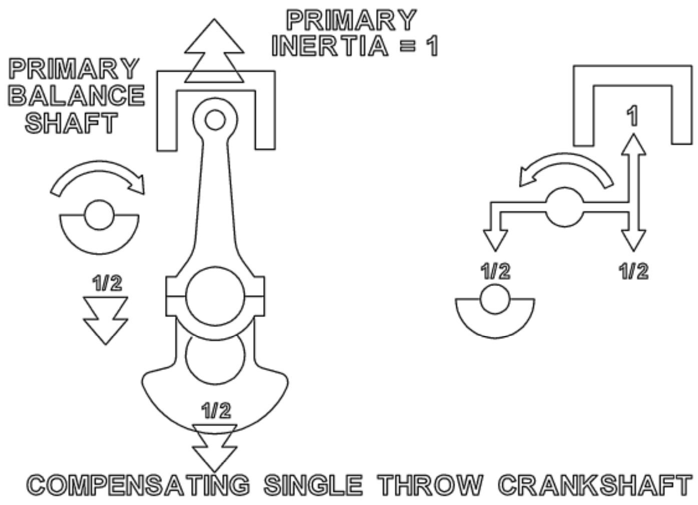
The theoretically optimum method of employing balance shafts is to have two shafts each with enough mass to generate half the canceling force located equidistantly on opposite sides of the crankshaft spinning in opposite directions. When the rotating masses were aligned at TDC or BDC they would create a force equal to the desired opposing force. As the crank rotates these shafts would create a force that fluctuates along the cylinder plane with a value equal to the cosine of the angle, and with no sideways force component. Two counter rotating balance shafts running at engine speed would therefore perfectly eliminate the primary shaking force of the single cylinder engine. Exactly the same arrangement geared up to twice the crankshaft speed would suffice to perfectly cancel out the secondary shaking forces as well.



COMPENSATING SINGLE THROW CRANKSHAFT
FORCES WITH ROTATING WEIGHTS

While this design would completely cancel out the shaking forces on a single cylinder engine, one has to question the practicality of adding four geared shafts with attendant bearings and weight, to what is probably a fairly simple and economical engine.

In most cases, the secondary shaking forces are smaller than the primary; so canceling them is of lesser importance. If we decide to try to reduce the primary forces to a reasonable level rather than totally cancel them, we can simplify the engine even further. If we assume the crankshaft is already rotating, we can place a second counter rotating balance shaft alongside it. If the balance shaft and crankshaft are properly balanced, they will create a reciprocating canceling force just like that of the two counter rotating primary force balance shafts we discussed a moment below. While these two shafts can create a canceling force that fluctuates according to the cosine of the crank angle, the solution is not perfect. The primary shaking force is traveling along the cylinder bore, but now our balancing shafts are not equally spaced on each side of the bore, in fact, one of them is directly in the line along which the shaking force propagates. While the combination of crankshaft and balance shaft will cancel out the shaking force, it will create a rocking couple it its place. If the total distance between the balance shaft and the crankshaft can be kept small enough, the leverage of this rocking couple will be proportionately small and a great reduction in overall vibration will still be obtained at modest cost.



COMPENSATING SINGLE THROW CRANKSHAFT
FORCES WITH ONE ROTATING WEIGHT
LEADS TO A ROCKING COUPLE
(SECONDARY FORCES UNCOMPENSATED)

The number and arrangement of auxiliary balancing devices will depend on the engine speed, weight of reciprocating and rotating masses, ratio of connecting rod length to crank throw, weight of engine, engine geometry, economic and engineering factors and so on.

CONCLUSION

In this article we have looked at some of the basic terms used in balance, and the units used to measure unbalance. The inertia of reciprocating masses and use of engine geometry to balance inertia forces were discussed. Introduction to the basic concepts used to perform actual engine balance were briefly examined and the employment of auxiliary means to cancel out unbalance forces were also considered. This was a brief glimpse into engine balancing; in practice there is much that was not covered.