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www.stahlheaders.com
1515 MT. ROSE AVE. - York, PA 17403
Phone: (717) 846-1632 Fax: (717) 854-9486 E-mail: sales@stahlheaders.com

Rod Length Relationships

You are invited to participate in this attempt to understand a part of internal combustion engines. I invite any/all criticisms, suggestions, thoughts, analogies, etc.-- written preferred, phone calls accepted from those too feeble or who have arthritis. Contributors are invited to request special computer printouts for specific combinations of interest to them.

In general, most observations relate to engines used for some type of competition event and will in general produce peak power higher than 6000 RPM with good compression ring seal as defined by no more than 3/16 CFM blowby per cylinder.

Short Rod is slower at BDC range and faster at TDC range.

Long Rod is faster at BDC range and slower at TDC range.

I. LONG ROD

A. Intake Stroke -- will draw harder on cyl head from 90-o ATDC to BDC.

B. Compression Stroke -- Piston travels from BDC to 90-o BTDC faster than short rod. Goes slower from 90-o BTDC to TDC--may change ign timing requirement versus short rod as piston spends more time at top. However; if flame travel were too fast, detonation could occur. Is it possible the long rod could have more cyl pressure at ie. 30-o ATDC but less crankpin force at 70-o ATDC. Does a long rod produce more efficient combustion at high RPM--measure CO, CO2? Find out!!

C. Power Stroke -- Piston is further down in bore for any given rod/crank pin angle and thus, at any crank angle from 20 to 75 ATDC less force is exerted on the crank pin than a shorter rod. However, the piston will be higher in the bore for any given crank angle from 90-o BTDC to 90-o ATDC and thus cylinder pressure could be higher. Long rod will spend less time from 90-o ATDC to BDC--allows less time for exhaust to escape on power stroke and will force more exhaust out from BDC to 90-o BTDC. Could have more pumping loss! Could be if exhaust port is poor, a long rod will help peak power.

D. Exhaust Stroke -- see above.

II. Short Rod

A. Intake Stroke -- Short rod spends less time near TDC and will suck harder on the cyl head from 10-o ATDC to 90-o ATDC the early part of the stroke, but will not suck as hard from 90-o to BDC as a long rod. Will require a better cyl head than long rod to produce same peak HP. Short rod may work better for a IR or Tuned runner system that would probably have more inertia cyl filling than a short runner system as piston passes BDC. Will require stronger wrist pins, piston pin bosses, and connecting rods than a long rod.

B. Compression Stroke -- Piston moves slower from BDC to 90-o BTDC; faster from 90-o BTDC to TDC than long rod. Thus, with same ign timing short rod will create less cyl compression for any given crank angle from 90-o BTDC to 90-o ATDC except at TDC. As piston comes down, it will have moved further; thus, from a "time" standpoint, the short rod may be less prone to detonation and may permit higher comp ratios. Short rod spends more time at the bottom which may reduce intake charge being pumped back out intake tract as valve closes--ie. may permit longer intake lobe and/or later intake closing than a long rod.

C. Power Stroke -- Short rod exerts more force to the crank pin at any crank angle that counts ie.--20-o ATDC to 70-o ATDC. Also side loads cyl walls more than long rod. Will probably be more critical of piston design and cyl wall rigidity.

D. Exhaust Stroke -- Stroke starts anywhere from 80-o to 110-o BBDC in race engines due to exhaust valve opening. Permits earlier exhaust opening due to cyl pressure/force being delivered to crank pin sooner with short rod. Requires a better exhaust port as it will not pump like a long rod. Short rod has less pumping loss ABDC up to 90-o BTDC and has more pumping loss from 90-o BTDC as it approaches TDC, and may cause more reversion.

III. NOTES

A. Rod Length Changes -- Appears a length change of 2-1/2% is necessary to perceive a change was made. For R & D purposes it appears a 5% change should be made. Perhaps any change should be 2 to 3%--ie. Ignition timing, header tube area, pipe length, cam shaft valve event area, cyl head flow change, etc.

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B. Short Rod in Power Stroke -- Piston is higher in the bore when Rod-Crank angle is at 90-o even though at any given crank angle the piston is further down. Thus, at any given "time" on the power stroke between a rod to crank pin angle of 10o and ie. 90-o, the short rod will generate a greater force on the crank pin which will be in the 70-o to 75-o ATDC range for most engines we are concerned with.

C. Stroke -- Trend of OEM engine mfgs to go to longer stroke and/or less over square (bore numerically higher than stroke) may be a function of L/R. Being that at slower engine speeds the effect of a short rod on Intake causes few problems. Compression/Power Stroke should produce different emissions than a long rod. Short rod Exhaust Stroke may create more reversion--EGR on a street engine.

D. More exhaust lobe or a earlier exhaust opening may defeat a longer rod. I am saying that a shorter rod allows a earlier exhaust opening. A better exhaust port allows a earlier exhaust opening.

E. Definition of poor exhaust port. Becomes turbulent at lower velocity than a better port. Flow curve will flatten out at a lower lift than a good port. A good exhaust port will tolerate more exhaust lobe and the engine will like it. Presuming the engine has adequate throttle area (so as not to cause more than 1" Hg depression below inlet throttle at peak power); then the better the exhaust port is, the greater the differential between optimum intake lobe duration and exhaust lobe duration will be--ie. exh 10-o or more longer than intake Carbon buildup will be minimal if cyl is dry.

IV. DEFINITIONS

Short Rod -- Min Rod/Stroke Ratio -- 1.60 Max Rod/Stroke Ratio -- 1.80

Long Rod -- Min Rod/Stroke Ratio -- 1.81 Max Rod/Stroke Ratio -- 2.00

Any ratio's exceeding these boundaries are at this moment labeled "design screw-ups" and not worth considering until valid data supports it.

Contributors to Date: Bill Clemmons, Jere Stahl

Connecting Rod Length Influence on Power

by William B. Clemmens

A spark ignition (SI) engine and a steam engine are very similar in principle. Both rely on pressure above the piston to produce rotary power. Pressure above the piston times the area of the bore acts to create a force that acts through the connecting rod to rotate the crankshaft. If the crankshaft is looked at as a simple lever with which to gain mechanical advantage, the greatest advantage would occur when the force was applied at right angles to the crankshaft. If this analogy is carried to the connecting rod crankshaft interface, it would suggest that the most efficient mechanical use of the cylinder pressure would occur when the crank and the connecting rod are at right angles. Changing the connecting rod length relative to the stroke changes the time in crank angle degrees necessary to reach the right angle condition.

A short connecting rod achieves this right angle condition sooner than a long rod. Therefore from a "time" perspective, a short rod would always be the choice for maximum torque. The shorter rod achieves the right angle position sooner and it does so with the piston slightly farther up in the bore. This means that the cyl pressure (or force on the piston) in the cylinder is slightly higher in the short rod engine compared to the long rod engine (relative to time).

Table 1
ROD LENGTH RELATIONSHIPS*
(with Crank @ 90 deg ATDC)
Piston Position Crankpin/Rod Angle

Stroke	Rod Length	Rod Angle	from TDC	ATDC
3.5	5.7	17.88	2.025	72.12
3.5	5.85	17.4	2.018	72.59
3.5	6	16.96	2.011	73.04
3.5	6.2	16.39	2.002	73.6

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Table 2
ROD LENGTH RELATIONSHIPS with CRANKPIN/ROD centerline @ 90o @ 7500 rpm

Stroke	Rod Length	Rod Angle	Piston Distance	Crank Angle	Piston Accel
3.5	5.7	17.07	1.487	72.93	2728.35
3.5	5.85	16.65	1.494	73.35	2504.72
3.5	6	16.26	1.5	73.74	2324.26
3.5	6.2	15.76	1.508	74.24	2097.27

*data from Jere Stahl

Another concern in selecting the rod length is the effects of mechanical stress imposed by increasing engine speed. Typically, the concept of mean piston speed is used to express the level of mechanical stress. However, the word "mean" refers to the average speed of the piston in going from the top of the bore to the bottom of the bore and back to the top of the bore. This distance is a linear distance and is a function of the engine stroke and engine speed, not rod length. Therefore, the mean piston speed would be the same for each rod length listed in Table 1.

Empirical experience; however, indicates that the mechanical stress is less with the longer rod length. There are two reasons for these results. Probably the primary reason for these results is that the profile of the instantaneous velocity of the piston changes with rod length. The longer rod allows the piston to come to a stop at the top of the bore and accelerate away much more slowly than a short rod engine. This slower motion translates into a lower instantaneous velocity and hence lower stresses on the piston. Another strong effect on mechanical stress levels is the angle of the connecting rod with the bore centerline during the engine cycle. The smaller the centerline angle, the less the side loading on the cylinder wall. The longer rod will have less centerline angle for the same crank angle than the shorter rod and therefore has lower side loadings.

Classical textbooks by Obert ( ) and C.F. Taylor ( ) provide little guidance on the rod length selection for passenger or commercial vehicles other than to list the ratios of rod length to crank radiuses that have been used by various engine designs. Race engine builders using production blocks have done quite a bit of experimentation and have found many drivers are capable of telling the difference and making clear choices along with similar results from motorcycle flat track racers/builders.

Because of recent developments in computer modeling of the engine cycle by R.D. Rabbitt ( ), another factor may be critical in selecting a given connecting rod length. This new factor is the cylinder head flow capability versus connecting rod length over stroke ratio (l/r) versus engine speed. To understand this relationship, let us first review previous techniques used to model air flow during the engine cycle which as Rabbitt points out is founded on principles initiated in 1862 and refined in 1920. These theories are documented in Taylor's textbook ( ). To calculate air flow throughout the cycle these models use such parameters as mean or average inlet mach number for the port velocity and an average inlet valve discharge coefficient which compensate for valve lift and duration. In these models a control volume is used to define the boundaries of the combustion chamber. The air flow determined by the previous parameters crosses this boundary to provide air (and fuel) for the combustion process within the control volume.

However, this control volume has historically been drawn in a manner that defines the boundaries of the combustion chamber in the area of the inlet and exhaust valves as if the valves were removed from the cylinder head (ie. a straight line across the port). With the valves effectively removed, the previously mentioned average port flow and valve discharge coefficient (ie. valve restriction) are multiplied within current computer models to quantify the air flow (and fuel) delivered for each intake stroke. But, as Rabbitt points out, this approach totally ignores the effect of the air flow direction and the real effect of valve lift on the total air flow that can be ingested on each intake stroke.

Rabbitt reaches two important conclusions from his study. One, because of the direction of the air flow (angle and swirl) entering the combustion chamber, three dimensional vorticies are set up during the intake stroke. Two, that above a certain piston speed, density of the mixture at the piston face is a function of valve geometry and valve speed. Rabbitt further discusses the effect of the first conclusion as it relates to the mass of air that is allowed to flow through the port and by the valve. Vorticies can exhibit different characteristics and in general conform to two general types--large scale bulk vorticies that could be described as smooth in nature and small scale eddies that are highly turbulant.

If one can consider that the vacuum produced by the piston on its downward travel to be the energy that causes the air to flow through the port when energy losses throughout the intake tract (including losses at the valve) are at a minimum, the flow delivered to the chamber will be maximized. If the area between the piston face and the valve is also included in the consideration of flow losses, the effect of the type of vorticies created can be more easily understood. Large scale bulk vorticies comsume less energy than highly turbulent eddy vorticies. Thus, more of the initial energy from the piston's downward movement is available at the port-valve-combustion chamber interface with which to draw the intake charge into the chamber. Small scale eddies eat up energy which reduces the amount of the initial energy that reaches the port-valve-combustion chamber interface which in turn, reduces the port flow.

Rabbitt's second conclusion follows that at some higher piston speed, the vorticies within the combustion chamber (which are assumed to be large scale bulk type at low speeds) transition from the bulk type to the small scale eddy type. At this point the flow into the combustion chamber ceases to increase in proportion to increases in engine speed. It is theorized that this flow transition point can be observed on the engine power curve as the point at which the power curve begins to fall off with increasing engine speed.

As indicated earlier, piston speed is normally viewed as mean or average piston speed. Thus for a given engine, the mean piston speed increases as the rotational engine speed increases. However, in Rabbitt's model the piston speed of concern is the instantaneous piston speed during the intake stroke near TDC. For any given engine, changing the rod length to stroke (l/r) ratio changes the instantaneous piston speed near TDC. For the purposes of flow visualization, the type of vortex formed should not care whether a given instantaneous piston speed had been achieved by a given rotational speed or changing the (l/r) ratio and operating at a new rotational speed. As long as the instantaneous piston velocity is the same, the type of vorticies formed should be the same and the amount of air inducted into the cylinder should be the same.

If other factors influenced by rotational speed such as the time distance between slug of intake air flow and valve opening rates relative to the acceleration of the air slugs were ignored, one should be able to predict the location (RPM) of the peak power as a result of a change in the (l/r) ratio. Note, that even though power is a funtion of air flow and air flow should be roughly constant for the same instantaneous piston speed (neglecting the afore mentioned factors), the power may not be the same because of the lever arm effect between the crank radius and the connecting rod. (As we noted earlier, the shorter rod should have the advantage in the lever arm effect.)

In reality, the analysis must be viewed by stroke (ie intake, compression, exhaust, power) the selection of exhaust valve opening time combined with the exhaust system backpressure and degree of turbulance the exhaust port experiences. If the exhaust port has good turbulance control then you may run a shorter rod which allows you to use more exhaust lobe which reduces pumping losses on the exh stroke.