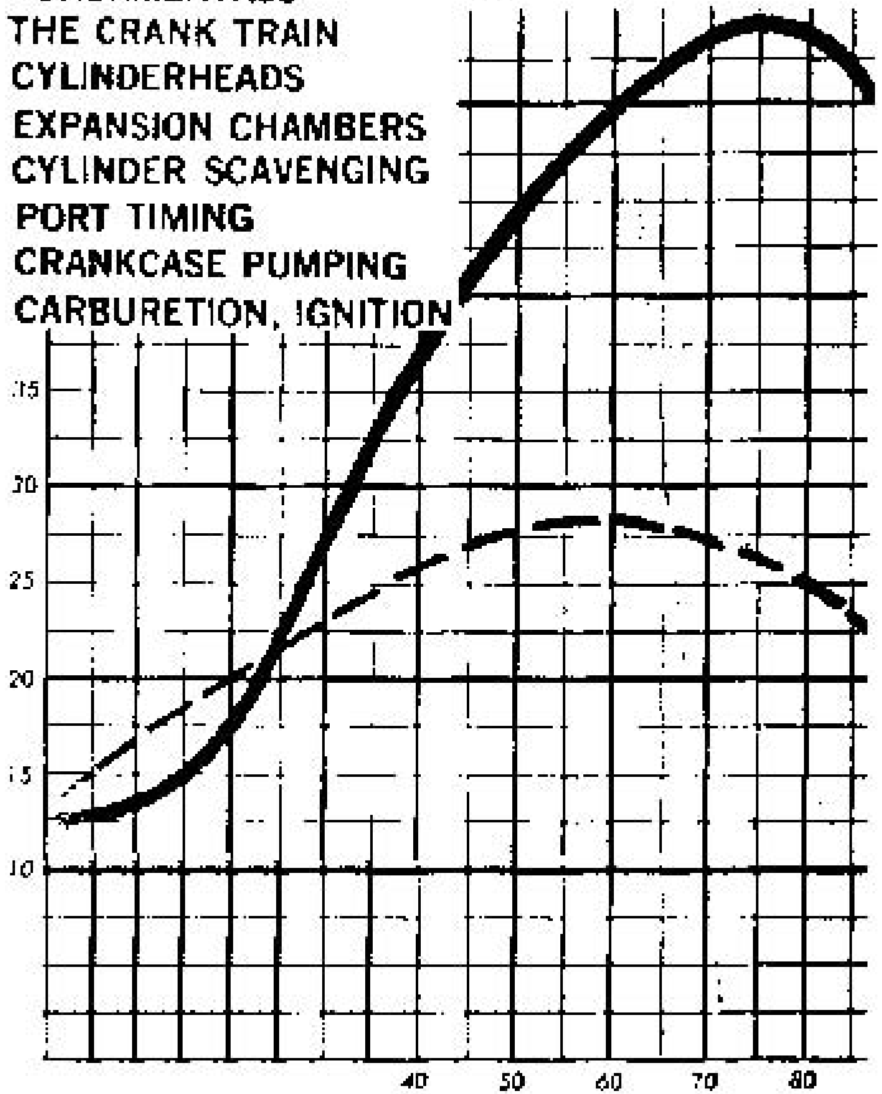
Two-stroke TUNER'S HANDBOOK

by Gordon Jennings

FUNDAMENTALS
THE CRANK TRAIN
CYLINDERHEADS
EXPANSION CHAMBERS
CYLINDER SCAVENGING
PORT TIMING
CRANKCASE PUMPING
CARBURETION, IGNITION



H. P. BOOKS

FOREWORD

Only ten years ago the two-stroke engine was widely and quite understandably thought to be a reasonable alternative to the four-stroke only when minimum weight and manufacturing cost were all-important considerations. The two-stroke was recognized as having substantial theoretical promise, as it delivered a power stroke for each 360 degrees of crankshaft rotation but the hard reality was that each individual power impulse was too feeble to amount to much when totalled at the output end of the crankshaft. A very few engines had begun to appear in which some of the theoretical promise was realized however, and this encouraged engineers at MZ, Yamaha and Suzuki to persist in their efforts to wring competitive power output from the racing two-stroke engine. To say that they were ultimately successful would be gross understatement.

Those engineers were motivated by the need to demonstrate that the two-stroke engine, *per se*, was worthwhile – as that would stimulate sales of their companies' ordinary touring models. My own interest in the two-stroke, which had reached the level of an obsession by 1963, was generated by comparative poverty. I like to tinker with engines, and the complexities of the poppet-valve four-stroke make modifications very expensive. One may think that a change in valve timing would do wonders for a four-stroke's power, but getting a camshaft made to order costs hundreds of dollars. In contrast, a two-stroke engine's valve timing may be altered simply by reshaping the holes in its cylinders, and its power output markedly changed by utilizing inertia and resonant effects in its intake and exhaust tracts. None of these modifications are costly.

On the other hand, while the two-stroke engine does not commonly require large dollar inputs to raise its power output, it does require an in-depth understanding on the part of the man doing the modifications. In an attempt to acquire that understanding I began a study of the high-speed, high-output two-stroke engine that has led to the collection of a minor library of text books and SAE papers. And to an endless series of experiments, some of them illuminating and many others raising more questions than they have answered. At this stage I have arrived at more or less satisfactory explanations for most of the gross phenomena, such as the general behavior of expansion chambers and port time-area values, and I flatter myself to think that just that much is an acceptable excuse for writing this book for the guidance of the layman experimenter. If it will not supply all of the answers it will at least take care of the fundamental problems and prevent the worst mistakes.

My special thanks to Mr. John Brooks, of McCulloch Engineering, who has done much to dilute my once pure ignorance (but should not be held accountable for the residue found herein). Also to the late Henry Koepke, who mistakenly assumed that I knew something about two-stroke engines and supported my early research; to my old friend Joe Parkhurst, who started me working on this book nearly ten years ago but never got it; and finally to Tom Heininger, who wheedled, needled, pleaded, complained and cajoled until I hammered my file of notes into publishable form.

CONTENTS

Throughout this book it will be assumed, inconvenient though that assumption may occasionally be, that the reader has progressed to at least a superficial knowledge of the manner in which a piston-type internal combustion engine – with particular reference to those operating on the two-stroke cycle principle – converts quantities of fuel and air into useful power delivered at the end of its crankshaft. People who need enlightenment in that regard will find a wealth of explanatory literature collected on the shelves of any public library; no real purpose would be served by lingering over the matter here. Neither will I attempt to instruct you in the elementary mathematics and physics required to grasp much of what follows, as again the public library is an entirely adequate source of information. What will be provided is a kind of "state of the art" report about high-speed, high-output two-stroke engines for laymen – who in most cases do not have access to the literature (SAE papers, etc.) available to engineers and thus must rely upon hunches (often wrong) and folklore (almost invariably wrong) for guidance. Many have learned, to their sorrow, that it is distinctly possible to lavish enormous amounts of time and money on the two-stroke engine without realizing a return appropriate to the investment. The information to be provided here will not make you a Kaaden, or Naito; it will help you to avoid some of the more serious mistakes.

The first serious mistake a layman experimenter can make is to assume that those who designed and manufactured his particular engine didn't know what they were doing. In point of fact, the professional engineer knows very well, and if the engine in question is something other than what the experimenter has in mind, there are excellent reasons: all engines are compromised, from what you might consider an ideal, in the interest of manufacturing economy and broad usefulness. For example, ports may derive their shape as much from what the design engineer intended to be a low scrap-rate casting as from consideration of flow characteristics. In other words, even something like ports-design always will be influenced by the demands of mass-production manufacturing. Similarly, designing for mass-market sales implies that an engine must be agreeable to many different uses – even though that inevitably means that it will do no single thing particularly well. In these areas will we find the latitude for "improving" an engine, and one should always be mindful that the real task is simply to tailor a mass-use product to a very specific application – and that in the tailoring process one inevitably will incur all the various expenses the engine's designer has avoided. Hours of labor may be required to finish rough-cast ports; dollars will be spent correcting other things that are the creatures of manufacturing economics; power added at maximum revs will be power subtracted at lower crankshaft speeds, while the increased speeds required to obtain large improvements in power output will be paid for in terms of reliability.

Another mistake commonly made, sometimes even by those who have enjoyed some success in modifying two-stroke engines, is to believe in a kind of mechanistic magic. Bigger carburetors, higher compression ratios, altered port timings and expansion chambers often do bring an improvement in power output, but more and bigger is not magically, instantly better. All must work in concert with the basic engine, directed toward the particular application, before they constitute a genuine improvement. You cannot treat them as a voodoo incantation, hoping that if you mutter the right phrases and stir the chicken entrails in the prescribed manner, your mild-mannered, all-purpose chuffer will be transformed into a hyperhorsepower firebreather. With a lot of luck, you might get that result; the chances heavily are that you won't.

With all the mysticism filtered out, horsepower at any given displacement is simply a function of average pressure in the cylinder during the power stroke and the rate at which power strokes occur, minus work absorbed by friction and scavenging. Raise pressure and/or the delivery rate of the power strokes, or reduce friction and pumping losses, and the engine's net output will rise. Unfortunately, there are limitations on all sides: Pressure must be limited because of thermal considerations (and is further limited by an engine's restricted ability to recharge its cylinder with a fresh air/fuel mixture between power strokes). The limit for power strokes per unit of time is established by what is tolerable in terms of crankshaft rotational speeds, and tolerable here is what the bearings, rod and piston will survive, in inertia loadings, for what *you* consider an acceptable service life; the design engineer has already expressed his opinion in this matter. Pumping losses can be reduced—relative to the mass flow through an engine—with a properly designed exhaust system, but otherwise are an inevitable and almost invariable consequence of pulling air from the atmosphere, moving it through the engine, and out the exhaust port. Some improvement in output may be obtained with reductions in friction, but the scope for such improvements is very small compared to what may be accomplished with cylinder pressure and engine speed.

Obviously, pressure in a cylinder will vary continuously throughout an engine's entire power stroke. Knowing what those pressures may be in a given engine is useful, but more useful still is knowing what they should and are likely to be, as such knowledge can keep you from that futile exercise commonly known as flogging a dead horse—and from believing a lot of lies about how much power various people are getting from their engines. Engineers have an overall efficiency rating called "brake mean effective pressure", which they calculate by working their way back through torque readings observed on the dynamometer, the leverage provided by crankpin offset, and piston-crown area. Thus, bmep says little about peak cylinder pressures (those measurements being taken with a pressure transducer and oscilloscope) but

it is an excellent relative indicator of performance and highly useful in projecting power output from a modified engine.

PREDICTING POWER

An average, well-developed stock engine intended for use in a sports/touring motorcycle will have a bmep of about 70 psi. It is possible, and I must stress that word "possible", to raise this to perhaps 115 psi—an improvement of some 64-percent, which (if accomplished) will yield a 64-percent increase in power output without raising the engine's operating speed. Similarly, a 64-percent increase in operating speed without a change in bmep would have the same effect on output. You will see this in the following formula for calculating horsepower:

$$BHP = \frac{PLAN}{33{,}000}$$

Where BHP is brake horsepower

P is brake mean effective pressure, in psi
L is piston stroke, in feet
A is the area of one piston, in square inches
N is the number of power strokes per minute

Obviously, when the values of L and A are held constant, as would be the case with an engine having a piston displacement at the limit established for a particular racing class, then increases in power may only be obtained by increasing the values for P and N—and you will find that in practice it is a lot easier to increase the latter than the former.

As already stated, bmep figures for stock, touring-type engines with flow-restricting aircleaners and mufflers, and with porting/carburetion compromised in favor of smooth low-speed running, will be around 70 psi. Typical figures for engines with porting and other plumbing arranged solely (and effectively) for maximum horsepower at peak revs would be about 115 psi—with a few small, highly-developed two-stroke engines operating up at 125 psi. The exact number will vary according to unit cylinder displacement and the width of an engine's useful power band, but one may reasonably expect that engines suitable for motocross will fall in the 85-95 psi range—with big cylinders tending toward the lower figure and small cylinders *vice versa*. Road racing engines, tuned to exert a maximum effort over a very narrow speed range, will usually show a bmep of 100-115 psi, and of course the same remarks regarding the influence of cylinder size apply.

These numbers have a usefulness beyond the mere satisfaction of vulgar curiosity: they may be used very profitably to determine an engine's suitability for some particular application. For example, they shed light on the future prospects of those who are trying to transform Kawasaki's F-5 "Bighorn" engine, a 350cc single, into a prime-mover capable of ending the Yamaha TD-2's absolute domination in road racing. Much has been made, by the Kawasaki's supporters, of the usefulness of a broader power range inherent with the F-5's disc-valve induction and the 100cc advantage it gets, over the TD-2, by having only a single cylinder (this, under the present American Motorcycle Association rules). Now while it is true that a racing motorcycle having a wide power band is easier for its rider to manage, and may offer an absolute if very slight advantage on short, extraordinarily twisty circuits, one must not overlook the fact that the TD-2 has been blessed with an excellent close-ratio transmission and a number of riders quite capable of coping with any problems introduced by the need for frequent gear changes. Viewed realistically, the situation facing any serious challenger to Yamaha's supremacy is one in which horsepower must be met with horsepower. And what are the Kawasaki's prospects of developing that kind of horsepower? Let's have a look at the numbers:

Assuming that the man who modifies the Kawasaki F-5 knows his business, but doesn't have all the development time in the world, (probability favors the latter far more than the former) then he very likely will arrive at a combination of porting, etc., good for a bmep of about 105 psi—which is about all that can be expected with a single cylinder of 350cc displacement. To expect more would be to ignore the considerable difficulties in scavenging efficiently the F-5's large-bore (3.17-inch) cylinder. Further assuming (and as we shall see later, this assumption is far from safe) that the F-5 engine will remain in one, working piece for the duration of a longish race with its rider observing a red-line of 9000 rpm, with a power peak at 8500 rpm, then,

$$\text{BHP} = \frac{105 \times .223 \times 7.59 \times 8500}{33{,}000}$$

$$\text{BHP} = 47.6$$

So, a well developed F-5 would deliver 47.6 brake horsepower. How does that compare with the Yamaha TD-2? With all the years that have gone into the TD-2's development, and giving due thought to Yamaha's proven expertise in these matters, it seems safe to assume that this engine would be operating with a bmep of 115 psi at its power peak—which seems to be at 11,000 rpm. Thus, working from those numbers and the 250cc Yamaha twin's bore/stroke dimensions of 56mm and 50mm, respectively,

$$\text{BHP} = \frac{115 \times .164 \times 3.81 \times 22{,}000}{33{,}000}$$

$$\text{BHP} = 48.0$$

Clearly then, those who would try to beat the Yamaha with a Kawasaki F-5 have taken upon themselves a task of considerable magnitude. The only bright spot in the picture, for them, is that while they are 0.4 bhp down on the Yamaha (assuming near-optimum work on their part) they probably will have the advantage in terms of average horsepower, figured from the moment a gear is engaged—when revs fall somewhat below those for peak horsepower—until the red-line is reached and it is time for a change to the next higher gear. There will be no advantage in frontal area, for although the F-5 engine is narrower than that of the TD-2, the fairing must be wide enough to shroud the rider, and the minimum width that requires is sufficient to encompass either engine. Moreover, moving from the theoretical to the practical for a moment, it is highly unlikely that the Kawasaki could be made as reliable at 8500 rpm as is the Yamaha at 11,000 rpm, and not because the F-5 engine is badly designed or shoddily constructed. The simple truth is that any single-cylinder 350cc engine with the F-5's bore/stroke dimensions and red-lined at 9000 rpm is going to be stressed very near its absolute limit—a limit imposed by the properties of available materials.

PISTON SPEED

All this asks the question, "how does one determine the limit, with regard to engine speed"? Unfortunately, establishing this limit with any precision is not only extremely difficult in terms of the mathematics involved, but also requires data concerning metallurgy, etc., seldom available outside the record-rooms of the factories from which the engines originate. Still, there are guide-lines which, if lacking in absolute precision, do at least have the virtue of simplicity, and will provide an indicator to keep us away from certain trouble. It is almost impossible to establish the point, in engine speed, between zero trouble and the possibility of trouble; there is much less difficulty in determining a red-line between *some* trouble and nothing *but* trouble.

A quick and easy method of establishing a limit for crankshaft speed is by working with piston speed. Actually, with "mean" piston speed: pistons do not travel at uniform velocity; they move from a dead stop at each end of their stroke, accelerate up to a maximum speed that often is in excess of 120 mph, and then brake to another complete stop. For convenience, we use just the mean piston speed and the safe limit for that, for engines having bore-stroke dimensions within the range considered normal for motorcycles, is about 4000 feet per minute. And mean piston speed may be calculated very easily by applying the following formula:

$$C_m = .166 \times L \times N$$

Where C_m is mean piston speed, in feet per minute
L is stroke, in inches
N is crankshaft speed, in revolutions per minute

Thus, using again the Kawasaki F-5 engine as an example, with L being 2.68-inches and N given as 9000, we find that

$$C_m = .166 \times 2.68 \times 9000$$
$$C_m = 4000 \text{ ft/min}$$

Here we have a theoretically-predicted limit that seems to agree quite closely with observable reality in the field: Reports from those actually racing modified F-5 Kawasakis indicate that the engine does in fact retain acceptable (within the framework of that word's meaning in racing) reliability when red-lined at 9000 rpm, and ravels with horrifying abruptness if pressed further. Of course, it must be stressed here that few engines, the F-5 not excepted, retain more than marginal reliability at mean piston speeds of 4000 ft/min, and even this presupposes frequent replacement of the piston and the crank/rod bearings.

You will be on far more solid ground if your engine is not asked to endure mean piston speeds above 3500 ft/min. Anything above that takes an engine into the twilight zone of reliability, and the ground between 3500 ft/min and the near absolute limit of 4000 ft/min is covered with unpleasant possibilities, but these often may be minimized with the proper selection of materials and lubrication. I should note here that there are exceptions to this rule among some of the old-fashioned, long-stroke engines, which tend to have very light (and strong) reciprocating parts relative to their absolute stroke. An example that comes to mind is the Bultaco 125cc TSS, which had a stroke of no less than 2.36-inches (decidedly long for a 125) but which would, in "factory" road racing trim run up to 11,500 rpm, just like the Yamaha TD-2 (with a much shorter, 1.97-inch stroke), and that represents a mean piston speed of 4500 rpm. Obviously, Bultaco held the opinion that the resulting thinnish margin of reliability was acceptable, but their TSS never was as predictably trouble-free as Yamaha's TD-2, which at the same crankshaft speed (11,500) has a mean piston speed of only 3775 ft/min.

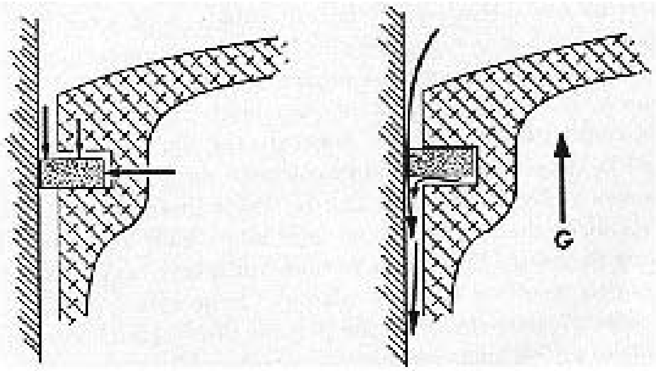
While on the subject of bore/stroke dimensions, I would like to say that there is much in favor of long stroke two-stroke cycle engines in many applications. They are not superior (as many people seem to think) compared to the presentday short-stroke designs in terms of low-speed torque, as torque is entirely a function of displacement and bmep, and wholely unrelated to bore/stroke ratios. With a long stroke, there is (at any given displacement)

a reduction in bore, and with it a loss of piston area against which gas pressure can exert its force, that exactly balances the loss of leverage in a short-stroke engine (which is, in turn, compensated by a gain in piston area). The only thing wrong with the long-stroke engine is that its crankshaft speed is limited by inertia loadings, and that in turn limits its absolute power potential as compared with the "modern" short-stroker. On the other hand, it is compensated by having a much more compact combustion chamber, which makes for more efficient burning, and by lower thermal loadings on the piston as a result of the smaller crown area into which heat from the combustion process may soak. Finally, there is an advantage in port area for the long-stroke design resulting from its relatively large cylinderwall area. This area increases in the long-stroke engine because displacement rises only in direct proportion to stroke, but is increased by a factor of 3.14 (the constant, pi) with enlargements in bore. These are very real advantages, but they are not enough, usually, to prevail against the short-stroke engine's sheer ability to rev. Crankshaft speed is the only thing subject to much juggling in the horsepower equation—and is a far more potent factor in determining power output than the relatively slight improvements in bmep obtainable with the marginally better combustion chamber and porting in the long-stroke engine. A 10-percent improvement in our Kawasaki F-5 engine's bmep (a large improvement indeed) would raise its output to 52.3 bhp; leave the bmep unchanged, but shorten the stroke and spin it 11,000 rpm and you would have [illegible] bhp. There is indeed no substitute for revs.

PISTON ACCELERATION

Sadly, while there is no substitute for revs, there are plenty of barriers: piston speed is one, as was already noted. But that is a rather indirect limit, as it ignores the fact that it is not speed so much as all the starting and stopping of pistons that does the damage, or at least the worst of any damage. The acceleration forces generated by the starting and stopping are felt even in an engine's mainbearings, but they are at a peak in the connecting rod and piston and have a particularly disastrous effect on the latter, as any attempt to make a piston stronger is apt also to make it heavier—which aggravates the very situation the strengthening of the piston should improve. Even so, an engine's one Achilles heel, the problem that may most strongly resist solution, often is the disastrous effects piston acceleration may have on the piston's rings.

It often is thought, and quite wrongly, that rings maintain a seal between the piston and the cylinder's walls simply through their properties as springs. A little thought should convince you that such cannot be the case, for most rings, compressed in the process of installation, press outward against the



Normally gas pressure in the upper cylinder holds the ring down against the bottom of its groove and out against the cylinderwall, forming a seal (shown left). But piston acceleration, "G", can lift the ring, shut off pressure behind the ring and break the seal.

cylinder with a force amounting to about 30 psi. Gas pressure in that cylinder may easily exceed 750 psi, and it should be obvious that a 30 psi force will not hold back one *circa* 750 psi. Still, equally obviously, piston rings *do* form an effective seal. How? Because they get a lot of help from the cylinder pressure itself: gas pressure above the ring forces it down against the bottom of its groove in the piston, and also (acting behind the ring, in the back of the groove) shoves it out hard against the cylinderwall. Thus, in the normal course of events, sealing pressure at the interface between cylinder wall and ring always is comfortably higher than the pressure it must hold back.

This very desirable situation will be maintained unless something happens to upset things, and most-insistent among the several "somethings" that may intrude is excessive piston acceleration. When piston acceleration exceeds the sum total of gas pressures holding the ring in place, the ring will lift upward (as the piston nears the top of its stroke, and is being braked to a halt). Instantly, as the ring lifts, the gas pressure previously applied above and behind is also applied underneath the ring, at which point its inertia takes over completely and the ring slams up hard against the top of its groove. This last action releases all pressure from behind the ring, leaving it entirely to its own feeble devices in holding back the fire above, and as its 30 psi outward pressure is no match for the 750 psi pressure in the upper cylinder, it is blown violently back into its groove. The ring's radial collapse opens a direct path down the cylinderwall for the high temperature and pressure combustion gases—but only for a microsecond, for the action just described instantly applies gas pressure once again behind the ring and that sends its snapping back into place against the cylinderwall. Unhappily, it cannot remain there, as gas pressure immediately bangs it back into its groove again—to repeat the process over and over until the piston is virtually stopped and the ring's inertia is no longer enough to counter gas pressure.

The net result of all this activity is that over the span of several degrees of crank rotation, immediately preceding the piston's reaching top center, the ring will be repeatedly collapsed radially and at the same time hammered hard against the top of its groove. Understandably, the ring is distressed by this, as it not only receives a fearful battering but also is bathed in fire while being deprived of the close contact with piston and cylinder that would otherwise serve to draw off heat. Equally damaging is that the piston is having much the same problem, with high-temperature gases blowing down past its skirt to cause overheating, to burn away the film of oil between itself and the cylinderwall, and with its ring, or rings, all the while trying to pound their way up through the piston crown. A mild case of what is quite accurately termed "ring flutter" eventually results in the destruction of the ring and sometimes the dimensional integrity of its groove; a more serious case is certain to lead rapidly into lubrication failure, overheating, and piston seizure.

Fortunately, this drastic problem can be avoided, thanks to the work of the researcher Paul de K. Dykes, whose investigation of the ring flutter phenomenon yielded most of what we know about it—and who invented the flutter-resistant ring that bears his name. Dykes showed us the cause of ring flutter, and engineers' understanding of the cause is reflected in their designs of the modern piston ring, which is very thin, axially, with a very considerable width, radially. Thus, gas pressure bears down on a large surface, providing an equally large total downforce, but is opposed by a relatively small upward load as the ring, being thin, is light and in consequence has little inertia. Still, even with very thin rings, flutter will occur if inertia loadings are high enough. To settle the question, with regard to any given engine, apply the following formula for determining maximum piston acceleration:

$$G_{max} = \frac{N^2 \times L}{2189}\left(1 + \frac{1}{2A}\right)$$

Where G_{max} is maximum piston acceleration, in feet per second squared
N is crankshaft speed, in revolutions per minute
L is stroke, in inches
A is the ratio of connecting rod length, between centers, to stroke

To illustrate how high these forces may sometimes be, let's use as an example the Yamaha TD-2, using 11,000 rpm for N. The formula tells us that