The Aspirin-Taker's Guide To Two-Stroke Port Timing

BY GORDON JENNINGS

• Mathematics never has been one of my favored pastimes, and that probably would account for my mixed feelings of fascination and loathing for two-stroke engines. They are fascinating for anyone with an inquiring mind, because under their superficial simplicity the infernal things are a mystery shrouded in an enigma. Yet, in me at least they also inspire loathing, as their behavior—while not without order nor beyond reason—is in many respects too complex to be rendered predictable without recourse to the subtlety and precision of mathematics. And math gives me headaches.

Knowing this, you will appreciate that my aspirin intake reached record highs a few years ago as I tried to lend some kind of order to the matter of port timings. It had become obvious that to concentrate on timing itself would be fruitless, for even a brief study of the information at hand showed that no clear pattern of timings and engine speeds existed. Further, logic insisted that if a pattern was to be found, it would necessarily involve not only timing and engine speed, but also the relationship between port area and cylinder volume. Why? Because the passage through a port of some given volume of gases depends on both time and area. Blowdown through an exhaust port, for example, is influenced by the port's area just as much as by the time it remains open. A narrow exhaust port would therefore be fairly high, and require a lot of duration; a wider port would need less time to do the same scavenging job. The same principle applies to the intake and transfer ports. Thus, if there was a universal rule for the two-stroke engine's port timings, it would have to be one expressed in time-area values per unit cylinder volume.

Developing that concept was easy. Coming up with universally-applicable numbers was an entirely different matter. To do that required accumulating a small mountain of data not easily obtained. Cylinder volume, port timings and engine speeds presented no special problem, but I would also need port-window areas and connecting rod lengths. Why the latter? Because nominal port area isn't very meaningful in this context. A port is fully-open only when the piston is at the end of its stroke; at all other times it is in some degree masked by the piston. The extent of this masking, relative to the nominal port-open duration, is influenced by the ratio of connecting rod length (on centers) to stroke, and also by the duration itself.

Has all the progress in two-stroke engine output been accidental? Or is there a universal rule for port timing? There's a rule.

Anyway, I needed a bunch of almost-unobtainable information, and even had I been able to collect it there was some question in my mind about the next step. The only means I then had at hand for solving time-area was a nasty bit of integral calculus, which meant just one enormous headache after another while working up numbers for all the ports in a necessarily large sampling of engines. Working with just two or three engines and drawing broad assumptions from that narrow base would have been just asking for serious error. But there wasn't time for anything else, so I finally closed my file on the whole matter and went back to the less-satisfactory but familiar educated hunch.

That's where the time-area question rested until fairly recently, when I acquired an SAE paper authored by Yamaha's Naitoh and Nomura. There, the time-area values I had sought were given, albeit without any explanation of their derivation. On a hunch, (that most-useful tool of the ignorant) I made some measurements of a cylinder from Yamaha's new TR3 racing engine, and worked up a set of time-area numbers based on the "mean" port-open areas. That is to say, the effective port aperture presented with the piston positioned halfway, in terms of crank-angle, from the point of port-opening to the end of its stroke. It should be noted that the piston will not be halfway down the port window when it is correctly positioned for measuring mean port area. For example, in an engine having exhaust opening/closing 90-degrees from bottom center, mean port area is taken another 45-degrees of crank-angle down—with, in most engines, about 70 percent of the total port window area exposed.

Working from this mean-area base, I developed the following numbers for the TR3 cylinder's ports: Exhaust, .000145 sec-cm^2/cm^3. Transfer, .000081 sec-cm^2/cm^3. Intake, .000148 sec-cm^2/cm^3.

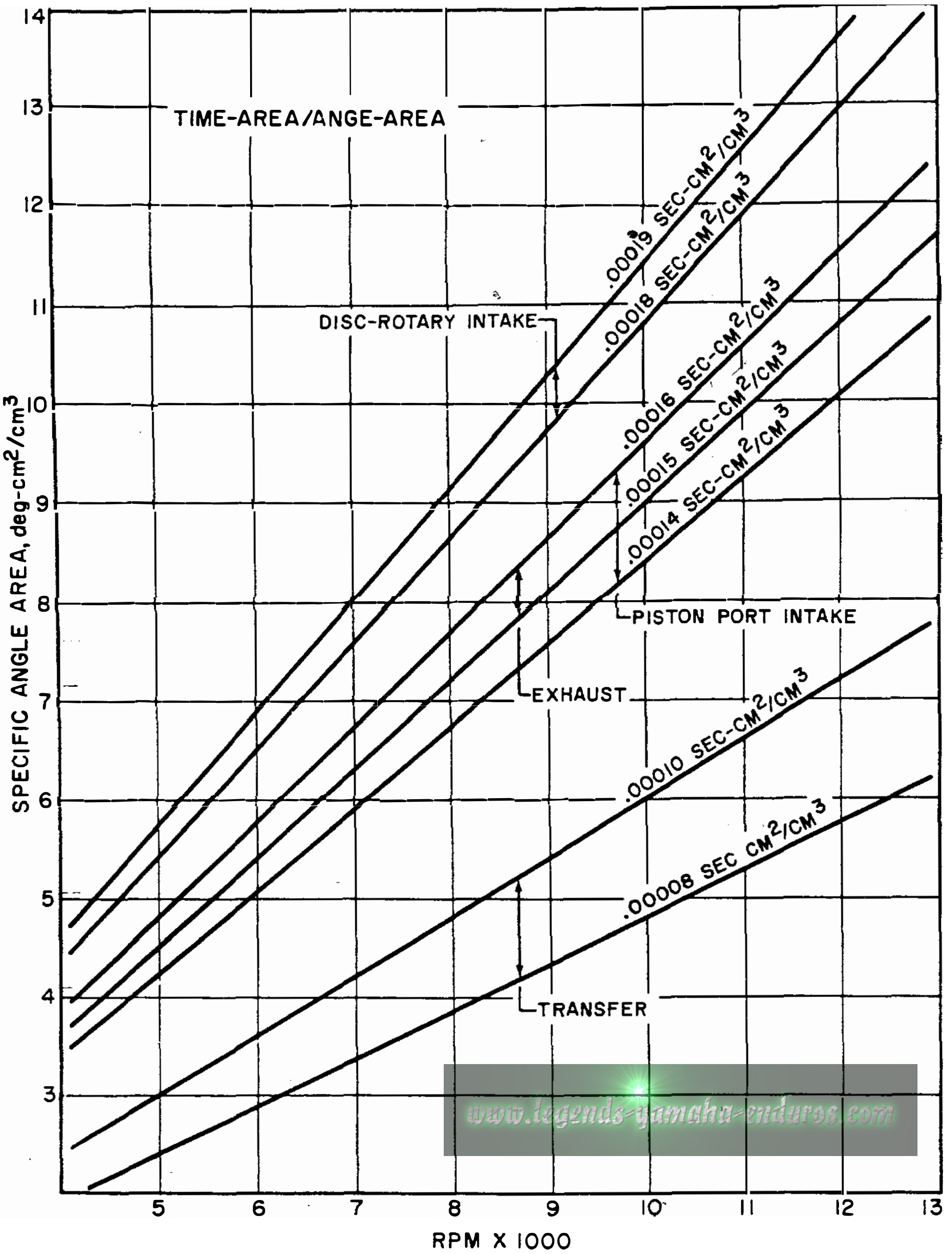
And referring to the Naitoh-Nomura paper, I found the following time-area values, for any two-stroke motorcycle engine, of whatever cylinder displacement and crank speed:

For the exhaust port, between .00014 and .00015 sec-cm^2/cm^3.

For the transfer ports, between .00008 and .00010 sec-cm^2/cm^3

For a piston-controlled intake port, between .00014 and .00016 sec-cm^2/cm^3; the suitable value for rotary-disc intake valves is between .00018 and .00019 sec-cm^2/cm^3.

That expression, "sec-cm^2/cm^3", may look both mysterious and intimidating, but it simply says "time-area per unit displacement" and means that the number it follows was derived by dividing cylinder volume, in



cubic centimeters, (cm^3), into mean port area, in square centimeters, (cm^2), and then multiplying the resulting number by the total time, in seconds, (sec), during which the port is open. Combining the formula for calculating time from the port-open period, in degrees, and engine speed, we have:

$$\text{Time-area} = \frac{60}{N} \times \frac{\theta}{360} \times \left(\frac{A}{D}\right)$$

Where, N = engine speed, in rpm
θ = port-open period, in degrees
A = mean port area, in cm^2
D = cylinder volume, in cm^3

That was my method in developing time-area numbers for the Yamaha TR3, and Yamaha's engineers must be working along the same lines, because there is very close agreement between Naitoh-Nomura's suggested values and those for the TR3. Moreover, additional checks made with various other two-stroke engines lend further verification. Finally, experiments with engines having port time-area deficiencies relative to those given by Naitoh-Nomura have shown that power increases as those deficiencies are corrected. On the basis of such evidence as I have, which admittedly is still too scanty to eliminate the possibility of some degree of error, the values presented are an extraordinarily valuable guide for anyone working with two-stroke engines.

Of all the factors one must insert into the formula I have provided, mean port area is the most difficult to obtain. The most direct method is simply to bolt a degree-wheel on an engine's crankshaft, set it for TDC, then find the point at which the port opens and, finally, rotate the crankshaft halfway from porting-opening to the appropriate end-of-stroke. You then measure the aperture exposed above the piston crown (or below the skirt, on the intake side).

A better method, I think, is to do all this with pencil, compass and protractor—working on fine-grid graph paper. To construct a working model, on paper, you begin by drawing a vertical line representing the cylinder axis, and a circle for the path followed by the crankpin. Then measure up from the top and bottom of the circle distances equal to the on-centers connecting rod length, and at those points add a pair of horizontal lines for the limits of piston travel. Port windows are then drawn flanking the cylinder axis, with the exhaust port and one transfer port against the lower piston-travel line; the intake port is drawn-in at the top. Finally, mark the mean port-open points on the crank circle, using a protractor, and connect these points to the cylinder axis with lines precisely the length of the connecting rod. The points-of-connection along the cylinde axis will fall inside the drawn-in port windows, and the areas above these points will be the mean exhaust and transfer port-window areas; the intake port area is below the marking point.

Obviously, the accuracy of either metho I have outlined depends very heavily on faithful reproduction of the port-window shapes. You may well have your own ideas about technique here, but the best approach I have found is to make a close-fitting sleeve out of stiffish paper, slide it down inside the cylinder, and then rub around the port edges with the side of a pencil to mark the window shapes on the paper. After this is done, the paper sleeve is withdrawn from the cylinder and reversed, so that the port-window pattern shows on the outside of the sleeve. Measurements may then be taken directly from the paper, but only after it is re-rolled into a cylinder, because the port widths are exaggerated when the paper is layed out flat.

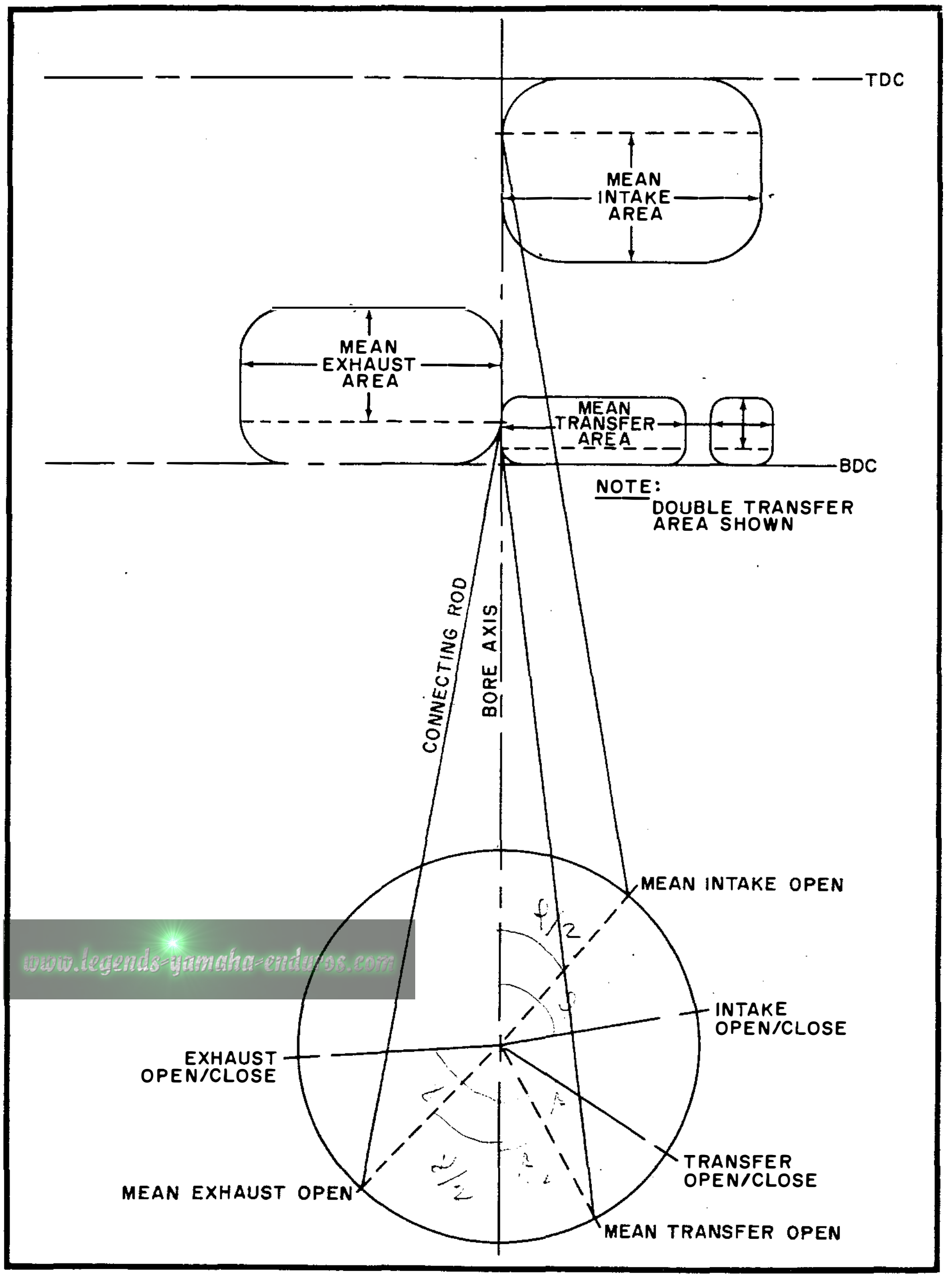
The reason for using grid-lined graph paper is primarily for ease in determining tl areas of irregularly-shaped ports. If you assign the grid lines a value of one-millimeter, and draw everything to scale, area may be found simply counting all the grid squares within the port window. This is a very grea help when, as is often the case, port windows are more nearly elliptical than square.

Another aid in the inevitable juggling of port timings and areas needed to get a given engine within the time-area limits suggested here is to work with angle-area instead of time-area, which simplifies the mathematics involved because it eliminates the steps needed to convert degrees and rpm into time. Of course the time factor cannot be ignored, so I have provided a chart showing the time-area/angle-area relationship. To find the angle-area for any port, you simply multiply port-open duration, in degrees, by mean port area divided by cylinder volume, giving you an answer in deg-cm^2/cm^3. To determine if the value derived is within limits, you mark a vertical coordinate for engine speed, and a horizontal coordinate for angle-area. The point of intersection should fall within the upper and lower limit-lines for the appropriate port on the chart.

The chart is also useful in demonstrating graphically that there is some latitude permitted in the time-area/angle-area of all the ports. At 8000 rpm, for instance, the exhaust port's angle-area is within limits with values from 6.7 deg-cm^2/cm^3 to 7.2 deg-cm^2/cm^3, and you may very well be moved to wonder about the influence variations within this range may have. As a matter of fact, there is a very considerable difference in an engine's power characteristics as combinations of port angle-areas are chosen from within the indicated limits.

Large-displacement motocross engines, for instance, should have angle-area values very near the upper limit for their transfer ports, but down at the lower limit for their exhaust ports. In large measure, this is a reflection of their exhaust-system characteristics: expansion chambers for such engines should return long-duration negative and positive wave pulses to the engine to provide the broadest possible useful power range, and when you extend the duration of these pulses, their amplitude is reduced. As a result, there never is a strong negative pulse to help move the fresh air/fuel charge up through the transfer ports, and high angle-area values are simply required for efficient scavenging.

Neither are exhaust-plugging pulses very strong with such expansion chambers, so it is also necessary to minimize charge loss by holding the exhaust port's angle-area at a minimum value. This measure will, of course, pay an added dividend in reducing to a minimum the amount of piston travel borrowed from the power stroke—thus improving torque over the entire speed range.



Road racing engines stand at the opposite extreme in terms of both power characteristics and angle-area combination. That is to say, for maximum power and ignoring power range, you should establish maximum angle-area values for the exhaust port (while still working within the suggested limits) and minimum transfer angle-areas. Again, the expansion chamber characteristics have much to do with this: road racing chambers commonly deliver pulses of rather short duration, but of extraordinary amplitude. Indeed, the scavenging pulse may well drop pressure in the cylinder to minus 7.0 psi while the piston is near bottom center and the transfer ports are almost fully-open. With that condition existing, it is neither necessary nor desirable to provide more than a minimum transfer angle-area. Extending the transfer period only increases the opportunity for charge-less out the exhaust port, and for back-flow into the crankcase, via the transfer ports, after the expansion chamber's scavenging pulse has faded away. This last is more important than you might think, for the crankcase pressure in an engine with a good expansion chamber will be pulled below atmospheric by the scavenging pulse. There is another very powerful reason for limiting transfer timing, in degrees, in the road racing engine, but that properly belongs in the overall discussion of intake port timing—of which much will be said later.

Carried to extremes, the combination of a strong-pulse expansion chamber, large exhaust port angle-area and minimum transfer port angle-area can elevate and narrow an engine's power curve remarkably. Yamaha's TR3 road racing engine has just such a chamber, with exhaust and transfer time-areas very near the maximum and minimum values, respectively. It produces something in the order of 63-65 bhp, from a displacement of only 350cc, but has a power band so narrow that a 6-speed transmission with ultra-close ratios is required to keep it operating within its effective power range. And that is something to keep firmly in mind, if you are reworking some engine with road racing intentions. Before you decide upon an angle-area combination that will maximize horsepower, look into the problem of transmission ratios. Should there be only the stock gear-set available, you will have to make adjustments in the engine's output curve accordingly.

Similarly, exhaust and transfer angle-areas for medium and small-displacement motocross engine must be biased at the expense of power range to get the kind of power needed to be competitive. In the 125cc class particularly, engines of near road racing specification are being used, in conjunction with 5- and 6-speed close-ratio transmissions. These small motocross bikes may not be pleasant to ride (you pedal them around with the gear-shift lever) but when that's what you need to be competitive, that's what you get. Or, as Mark Twain so succinctly put it, "When it's steamboat time, you steam." Just don't steam off in the wrong direction. The time-area and angle-area values given for exhaust and transfer ports will get you within limits; you still must do some juggling to suit your particular application.

When you are trying to establish suitable angle-area values in a given cylinder, you should begin by making increases over the stock values all in width. Curiously, widening an exhaust port has precisely the same effect on an engine's power characteristics as the same increase in angle-area gained by raising its upper edge—which is the approach most commonly taken. Maximum output will be raised, and the speed at which peak power is obtained. The only difference is that increases in exhaust-open duration tend to narrow the effective power band more than a similar increase in angle-area obtained by widening the port. But you also will find that there are sharp limits to increases in port width, if serious reliability problems are to be avoided. Widening an exhaust port beyond a width representing 62 percent of bore diameter and you are asking for trouble, because beyond that point the unsupported side of the ring will surely bulge out into the port far enough to snag on its upper edge as the piston moves up past the port window. This trouble can be minimized by chamfering around the window edges, and by making the window shape more nearly elliptical than rectangular, but the limit is still there. Actually, some racing engines, with near-round ports, give acceptable ring life with exhaust-port widths representing about 70 percent of bore, but it is difficult to work that kind of shape around an existing, stock exhaust port.

Transfer ports usually are not wide enough to cause any great difficulties with ring-snagging, but the possibility exists and modifications should be done with caution. Also, in widening either the exhaust or transfer ports, care should be taken not to crowd them too closely together, or you may create a condition in which a lot of the mixture flowing up through the transfer ports turns abruptly and short-circuits out the exhaust port. From all the evidence I have at hand, exhaust and transfer ports should be separated by at least .350-inch—and this holds true for cylinders of 125cc capacity and up.

You can also get into trouble with excessive width in an intake port, for the lower edge of the piston skirt can snag on the port's floor. This problem is not usually severe, however, as you will be forced to work within the limits imposed by the location of cylinder hold-down studs, etc., and even a slight rounding of the port floor will do much to relieve any problem that does occur. There is every reason to widen the intake port window if you are interested in maximum flow coefficient, for low and wide ports are best in that regard. But you should also know that such modifications alter the intake tract's reasonant frequency—which can lead to very large problems indeed, as the piston-port two-stroke engine depends very heavily on resonance and inertia effects in order to compensate for its symmetrical intake timing.

Unquestionably, the piston-controlled intake port is attractive for reasons of simplicity, but the fact that its intake-open period is symmetrically disposed before and after top center is a considerable disadvantage. Mixture drawn into the crankcase during the time between intake-opening and TDC is partially pumped back when the piston again descends, before intake-closing. Only the combined activities of sheer charge inertia, and sonic waves, hold the mixture in the crankcase while the piston closes the intake port. Ideally, intake-closing should occur at the precise moment when intake-tract ramming is at its peak, and that peak should coincide with pressure in the crankcase at the moment of intake-closing. But because resonance, particularly, occurs independently of engine speed, it can be truly effective only over a rather narrow speed range; at other speeds there will be blowback through the intake port.

Thus, all recommendations of angle-area notwithstanding, the most important aspect of establishing intake timing is to make it such that the negative-pressure wave created by intake opening moves out to the intake tract's end, inverts, and returns to the port window as a positive-pressure wave shortly before the piston closes the port. Predicting this wave motion theoretically is possible, but extremely difficult. Calculations must be based on flask-resonance, with the crankcase as the flask and the intake tract as the neck connected to atmosphere—a problem complicated enormously by the fact that this flask's volume changes continuously, and its neck has a cross-sectional area that also varies down its length. All things considered, it is a lot easier to establish tuned intake length by actual test, fitting a stub exhaust to eliminate exhaust system effects and making a series of tests with different intake lengths.

Clearly these resonant effects will be at a minimum and charge-inertia maximized with comparatively brief intake-open periods and small-diameter, lengthy intake pipes. And that's what you need for motocross engines: a very wide port window, to provide the necessary angle-area without excessive angle, with a smallish carburetor and long intake manifold. The reasons for using a small carburetor are not what you think; you need large changes in cross-sectional area in the intake tract to damp resonance. Indeed, when the carburetor throat area is made 35 percent or less of port window area, there is no measurable pulsation effect. You don't have to restrict the intake nearly that much, as considerable wave damping occurs below 50 percent throttling.

Inertia in the incoming charge will work wonders for a motocross engine, but not if the intake duration is greater than 160 degrees—which is the upper limit for even small, high-speed motocross engines. In general, I would say that anything less than 120 degrees will not give adequate angle-area values, but you should definitely get as much angle-area as is possible with port width and restrict port-open duration to maintain a broad power band. Fiddling with the intake port width will change the optimum intake tract length, but if you want optimum results, that is the price to be paid.

Road racing engines depend much more on intake tract resonance to prevent blowback through the carburetor, which is another reason why their power bands are so narrow. But even with the strongest inertia effects to hold the charge in the crankcase, these engines are still limited in terms of maximum intake-open duration because if there is too much they simply will not start. If you think about it, you'll see that at cranking speeds the volume of mixture pushed up through the transfer ports will be equal to that displaced by the piston between transfer-closing and intake-opening. At cranking speeds, the upward motion of the piston will not begin to pull a depression in the crankcase until after the transfer ports close, and everything drawn in after the intake port opens will be pushed back out by the descending piston until the intake port closes. For example, Yamaha's TR3 engine has transfer ports that close 120 degrees before top center, and an intake port that opens and closes 94 degrees from top center. Thus, crankcase pumping at cranking speeds is confined to the scanty 26 degrees of piston travel between the transfer and intake periods. I think that must be very near the minimum for starting even with a motorcycle that is forced into life with vigorous pushing, and it certainly would limit any efforts to increase power, or power range, by altering timing.

For various reasons, the proper angle-area for disc-type rotary intake valves is larger than for a piston-controlled port. Also, it is much easier to get adequate angle-area because the disc-valve engine is almost entirely

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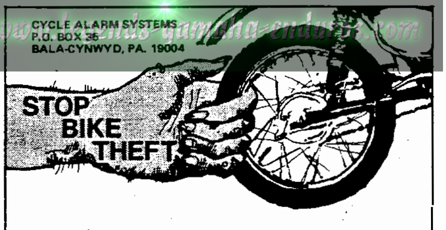

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PORT TIMING *Cont'd from page 52*

free of carburetor blowback. All of the current enthusiasm for reed valves aside, the rotary-disc valve still is the best induction method for crankcase-scavenged, two-stroke engines.

Like all other two-stroke engines, the disc-valve type's timing is very sensitive to exhaust system characteristics, and it can make maximum use of a good expansion chamber. For example, when the scavenging pulse lowers crankcase pressure below atmospheric in a piston-port engine, the only means of preventing back-flowing through the transfer ports is to close them before it happens. In the disc-valve engine, you simply arrange the port-open timing so that the crankcase can balance its pressure with air drawn through the carburetor—which not only sharply reduces the back-flow problem, but also gives overall crankcase charging a headstart. Generally speaking, the optimum port-open point will be between 130 and 145 degrees before top center, but the exact point must suit expansion chamber characteristics and should be determined experimentally.

In the same manner, some experimentation is needed to establish the optimum point for port-closing. Inertia effects in the intake tract will keep airflow moving into the crankcase some several degrees after the piston has passed TDC, and there is no reason to close the port until after the point of maximum charging has been reached. In most engines, this will occur at about 65 degrees after top center. However, port-closing may not be delayed quite so much in a low-speed engine, and more delay may be required for very high crankshaft speeds.

By now, you should be aware that while port time-area/angle-area values can be established within reasonably narrow limits, theoretically, final results with an engine still depend very heavily on sonic wave and gas-inertia effects at the intake and exhaust sides of an engine—and that all these factors must be in balance if maximum engine output is to be obtained. There are other influences, such as differences in port flow-coefficients and crankcase compression ratios, I have ignored entirely just because there is little variation in these things within the normal range. Casting techniques are about the same everywhere, which means that flow is substantially a function of port-window area, and crankcase compression ratios have all settled at about 1.5:1—which is what you get with a crankcase and flywheels in the normal proportions, and has proved to be best for nearly all engines anyway. Minor variations do make a difference, but they are inconsequential as compared with the differences realized by having correct time-area/angle-area values for a cylinder's ports. Yamaha's engineers have done us all an enormous service by providing the means for finding those values. ◉