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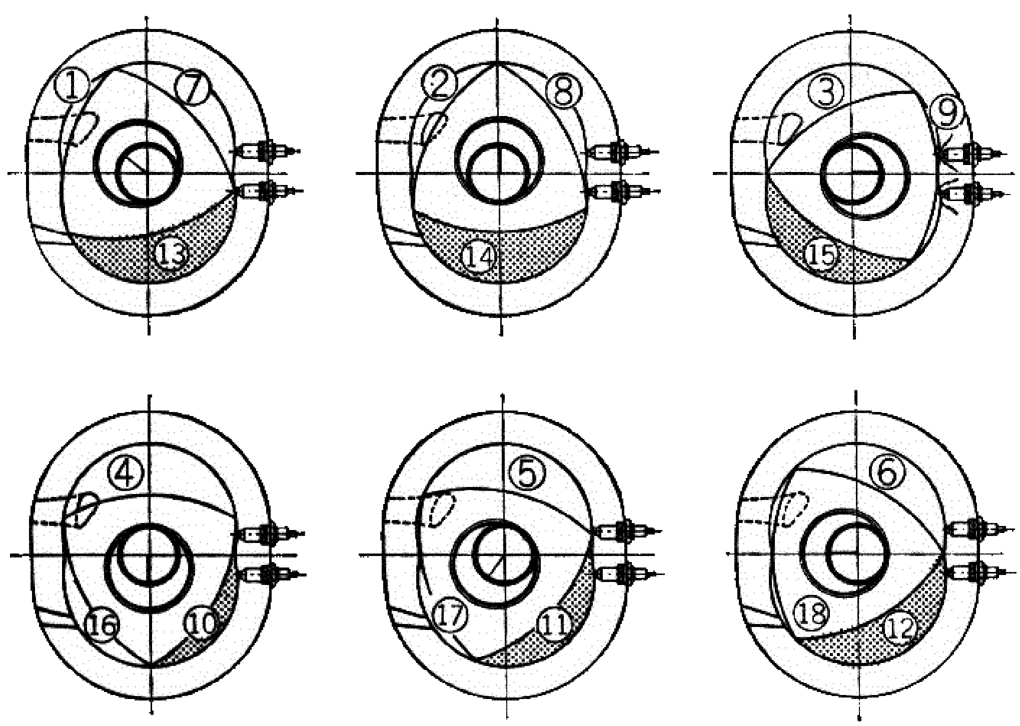
Port Timing Basics

After a great deal of thought, I decided that this first article should cover the basic workings of the rotary engine. In my experience, most people have the hardest time understanding port timing, and how it relates to engine operation. The accompanying illustration from "The Rotary Engine" by Kenichi Yamamoto will make this much easier to understand. At first, it may seem a bit confusing, but if you simply follow the numbers in order it is actually quite simple.

Before going into detail, it is critical that the reader understand some basic terminology. The various timing events of an internal combustion engine are typically stated in degrees of crankshaft rotation. In our case, output shaft, or eccentric shaft rotation. This terminology comes from the piston engine. Top dead center, or TDC refers to the working chamber being at its smallest possible volume. In a reciprocating piston engine, this occurs when the piston is at the very top of its stroke, hence the term top dead center. Bottom dead center, or BDC refers to the chamber being at its largest possible volume. In a reciprocating piston engine this occurs when the piston is at the very bottom of its stroke. All chamber volumes between TDC, and BDC, are referred to as Before TDC (BTDC), after TDC (ATDC), before BDC (BBDC), and after BDC (ABDC). For instance, 45° ATDC refers to the point at which the eccentric shaft has rotated 45° beyond top dead center. This is the situation in the first picture, looking at the chamber numbered 1. The line in the center of the picture extending from the crosshairs illustrates the angle of the eccentric shaft. This line coresponds with the keyway in the front of the shaft.



Below is a description of the complete process. Each description corresponds to the number in the illustration.

1. 45° ATDC The intake stroke is just beginning. The exhaust port has just closed, and on a stock or street ported engine, the intake port has been open for approximately 15°.

2. 90° ATDC The intake port is almost completely open, and the chamber is starting to expand at a fairly rapid rate.

3. 180° ATDC The intake port is all the way open, and has just passed the point of maximum flow. Maximum flow occcurs at approximately 135° ATDC, which corresponds with the maximum rate of chamber volume increase.

4. BDC of the intake stroke. The intake chamber is now at its largest possible volume. The intake port is partially open, and the port is still flowing in the forward direction, even though the chamber is no longer increasing in volume. This is due to the inertia of the column of air flowing in the induction system. This effect is referred to as inertial supercharging, and is described in further detail in the airflow section of my webpage. This will also be addressed in a later article.

5. 45° ABDC The chamber has started to decrease in volume, and with the exception of a stock US model 12A, which has an intake port closing of 40° ATDC, the intake port is still partially open. At high rpm, the intake port is still flowing in the forward direction due to inertial supercharging. At low rpm, airflow in the port has reversed, and some of the intake charge is being squeezed back into the induction system by the pressure of the intake chamber which is decreasing in volume. This is the result of the low velocity in the induction system. This is a very important point to consider, as this alone affects the operating range of the engine more than than any other factor.

6. 90° ABDC The intake port is completely closed, and air fuel mixture is being compressed.

7. 135° ABDC Same as #6.

8. 180° ABDC More of the same.

9. TDC of the compression stroke. The mixture is fully compressed, and ignition has started.

10. 90° ATDC The expansion cycle has started, and is already 45° past the point of maximum torque transfer to the eccentric shaft, which occured at 45° ATDC.

11. 135° ATDC The expansion stroke continues, but the torque transferred to the output shaft is now down to about 35% of its peak.

12. 180° ATDC The exhaust port is still closed, and the torque transfer to the eccentric shaft is approximately 15% of its peak.

13. 225° ATDC At this point, the exhaust port has been open for approximately 30°, and exhaust flow is quite high.

14. BDC of the exhaust stroke. This is typically the point of maximum flow through the exhaust port. Even though the chamber volume is not decreasing at an appreciable rate, the chamber pressure is very high, and this is responsible for a large percentage of the total exhaust flow.

15. 90° ABDC The chamber volume is decreasing, and is 45° away from the point of maximum rate of decrease of the chamber volume.

16. 180° ABDC The exhaust chamber volume continues to decrease, and at approximately this point, a bridge ported, or peripheral ported engine will have started to open the intake port.

17. 225° ABDC The exhaust port is still open, and the chamber volume is decreasing at a relatively slow rate. At this point, a mildly bridge ported engine will have just opened the intake port.

18. TDC of the intake stroke. Here we are at the beginning, ready to start all over again. Note that the exhaust port is still open, but the intake port, for a non bridge ported engine has not opened yet.

I have included the port timing for all RX-7 engines, and some alternative ports, so that you can make comparisons, and gain a greater understanding of how the rotary engine operates.

This information may seem very basic to some readers, but it is critical to the understanding of performance tuning. As most of you know, changing the port timing of the rotary engine can result in large horsepower gains. Further articles will discuss this in detail, and without this knowledge base, the upcoming articles will make very little sense.

Next months article will cover the exhaust cycle, and its effect on engine performance and efficiency.

Paul Yaw.

Port Timing

IO = Intake opens
IC = Intake closes
EO = Exhaust opens
EC = Exhaust closes

US Model First Generation RX-7

IO 32° ATDC
IC 40° ABDC
EO 75° BBDC
EC 38° ATDC

European Model Model First Generation RX-7

IO 32° ATDC
IC 50° ABDC
EO 75° BBDC
EC 48° ATDC

First and Second Generation 6-Port 13B

Primary intake (Part throttle/cruise)
IO 32° ATDC
IC 40° ABDC

Secondary intake (Part to full throttle)
IO 32° ATDC
IC 30° ABDC
Auxiliary high speed ports (Full throttle above approximately 4000 rpm)
IO 45° ATDC
IC 70° ABDC
EO 71° BBDC
EC 48° ATDC

Second and Third Generation Turbo 13B

IO 32° ATDC
IC 50° ABDC
EO 71° BBDC
EC 48° ATDC

Racing Beat "Street Port"

IO 25° ATDC
IC 60° ABDC
EO 84° BBDC
EC 48° ATDC

Racing Beat "J-Bridge Port"

IO 115° BTDC
IC 72° ABDC
EO 88° BBDC
EC 57° ATDC

Mazda Factory Peripheral Port

IO 86° BTDC
IC 75° ABDC
EO 73° BBDC
EC 65° ATDC

Exhaust Cycle Part One

Last months article described the basic internal workings of the rotary engine. The next several articles will break this down into separate cycles, and descirbe them in detail. I will begin with the exhaust cycle because it has the greatest effect on the power output of the engine. If the engine cannot exhaust itself completely, further modifications will result in very little improvement. This is true of naturally aspirated, and turbocharged engines. This first article will explain a few basic terms and concepts. Next months article will present some more new information, and then describe how all of this comes together to affect the complete exhaust cycle.

When attempting to increase the power output of the rotary engine, there are three basic aspects that can be improved upon. Volumetric efficiency, combustion efficiency, and reduction of pumping losses. As most of you know, the rotary engine has four separate cycles. Intake, compression, expansion, and exhaust. Of the four, only the expansion cycle contributes to the power output of the engine by exerting force on the output shaft. The other three cycles actually reduce horsepower by resisting the rotating force. This reduction in power is referred to as pumping loss. Pumping losses occur in both the intake and exhaust cycles. This article, and the next will deal with the importance of reducing pumping losses during the exhaust cycle.

Blowdown Period

Early internal combustion engines opened the exhaust valve at BDC of the expansion cycle. This required the piston to pump, or physically force the exhaust gasses from the cylinder during the period from BDC to TDC. The force required to pump the gasses from the cylinder considerably reduced the power output of the engine. As performance, and rpm requirements increased, it was discovered that by opening the exhaust valve before BDC the residual combustion pressure could be used to help evacuate the cylinder at the beginning of the exhaust cycle. This is referred to as the blowdown period, and is responsible for approximately half of the exhaust flow. In theory, this will reduce thermal efficiency by releasing pressure that is still applying force to the crankshaft. In practice however it was determined that the reduction in pumping losses far outweighed the loss of pressure at the end of the expansion cycle. Since most of the useful work is done in the first third of the expansion cycle, the pressure loss caused by early exhaust valve opening is minimal. This also applies to the rotary engine. Referring to last months article you can see that the exhaust port of a stock engine opens approximately 75 degrees before BDC.

Pressure Wave Tuning

Pressure wave phenomena is probably the least understood aspect of exhaust tuning. Right now I am thinking that it is also the hardest to explain! Entire books have been written on this subject, but I will try to boil it down to a few paragraphs.

Any time there is a pressure change in an elastic meduim (like air for instance) a series of resonances or vibrations will occur. Any time you hear a sound, it is the result of a pressure disturbance in the air. For instance, if someone across the room claps their hands together, the air pressure between their hands will increase. This rise in pressure will be transferred from one group of molecules to the next (at the speed of sound of course) until it finally reaches your ear. While this energy transfer is invisible, you can easily picture it by dropping a stone into an undisturbed pool of water. Pressure waves radiate outward from the center of the disturbance. This same thing happens in the exhaust system, but because of the higher pressures involved it is more like an elephant doing a belly flop in your swimming pool.

The main difference between the swimming pool analogy, and the exhaust system is that the pressure waves cannot travel outward in all directions from the source of the pressure disturbance, beacause they are enclosed by the tubing itself. In the case of the exhaust system, the initial pressure wave, or pulse caused by the exhaust port opening will travel towards the open end of the tube.

So far I have only referred to pressure waves as being positive, or caused by an increase in pressure. In fact, pressure waves can be negative, or caused by a decrease in pressure. Picture a wave in the ocean with the highest point of the wave being positive, or above sea level, and the trough between two waves being negative, or below sea level. This is analogous to the pressure waves in the exhaust system. These waves can also be referred to as high pressure, and low pressure.

These pressure waves can be used to our advantage because they have the effect of moving gas particles along with them. A positive, or high pressure wave will propel gasses in the same direction that it is travelling. A negative, or low pressure wave will propel gasses in the opposite direction that it is travelling. Take a moment to let this sink in, because this simple fact is at the heart of exhaust system tuning. Although the pressure wave is moving at the speed of sound, it will propel the gasses at a much slower speed. An example of this is a boat that catches a wave from another boat that is motoring by. As the wave passes it will propel the boat in the same direction the wave is travelling, but at a much slower speed, and the wave will eventually pass the boat completely. This is the same thing that happens to the gas molecules in the exhaust system as a pressure wave passes through them.

These pressure waves respond in an interesting manner when they reach a sudden area change in the pipe. An example of a sudden area change is the collector, where the two pipes empty into a larger diameter pipe, a megaphone, or the end of the exhaust where the pipe empties into the atmosphere. When a pressure wave reaches a larger cross sectional area, it will reverse its sign (positive becomes negative, and negative becomes positive) and its direction. For instance, when the exhaust port first opens, a strong positive wave will travel to the end of the pipe, change to a negative wave, and travel back to the exhaust port. This is called a reflection. Both the positive wave travelling towards the end of the pipe, and the negative wave travelling towards the exhaust port will propel exhaust gasses towards the end of the exhaust system which is exactly where we want them to go. The amount of time that this cycle takes is dependant on the total distance that the wave has to travel.

By changing the length of the header pipes, you can time the cycle so that the negative return wave arrives at the exhaust port at the end of the exhaust cycle where it is most beneficial. Assuming that the negative return wave is timed correctly for a given engine at 6000 rpm, lengthening the headers will further delay the return wave so that it is timed appropriately for a lower rpm, and shortening the headers will time the return wave so that it is timed appropriately for a higher rpm. The key to header length tuning is simply timing the low pressure return wave to give the greatest benefit for a given rpm.

This is a VERY basic description of pressure waves, and how they affect the exhaust system of an internal combustion engine. For a more detailed analysis, I would suggest researching two stroke exhaust system design. There is a great deal of information in print, and much of it can be found at public, or university libraries.

Velocity

Velocity refers to the speed at which the exhaust gasses are travelling. The exact speed is not important to this discussion, but an uderstanding of how velocity affects exhaust flow is. There are two ways that velocity can be increased. One, by decreasing the cross sectional area of the orifice that the gasses are flowing through. (Making the headers or exhaust ports smaller) Two, by increasing the volume of air that is flowing through the orifice. (Increasing engine rpm) Velocity will increase proportionally with an increase in rpm. In other words, if you double the rpm, the velocity will also double. Velocity is inversely proportional to an increase in cross sectional area. Doubling the cross sectional are will halve the velocity, and halving the cross sectional area will double the velocity.

Velocity is important for one simple reason. Inertia. Websters dictionary describes inertia as "The property of matter by which it retains its state of rest or velocity so long as it is not acted upon by an external force." In other words, once it is moving, it will continue to move until some external force stops it. If you apply this theory to the gasses in the exhaust system you can see that once they have been accelerated by the pressure in the combustion chamber, It will take a given amount of energy to stop them, and even more to cause them to reverse direction. Since energy equals mass times velocity squared, you can see that doubling the velocity of the gasses will quadruple the amount of energy required to stop them. This is important because the flow of exhaust gasses is not steady. During each exhaust cycle, the gasses are accelerated, and decellerated rapidly. Often in the forward and reverse direction.

Next months article will take all of these concepts, and describe how they affect the exhaust cycle. Like everything else that seems complex, it is just a combination of many very basic theories. If you take the time to fully understand this months article, next months article will leave you with a thorough understanding of the exhaust cycle.

My goal in writing these articles is to inform you, the reader so that you can go faster, and make appropriate decisions when modifying the rotary engine, or buying performance parts. Until next month, have fun, and thank you for reading.

Exhaust Cycle Part Two

Hopefully all of you have digested last month's information, and are ready for more. Last month I described the importance of velocity, pressure wave tuning, and opening of the exhaust port before BDC. Now it is time to take this information and see how it can be used to optimize the exhaust cycle. Let's start by looking at the effects of a less than optimum exhaust cycle.

A motor has fully exhausted itself (When it is really tired?) when the pressure in the chamber is equal to, or below atmospheric at the end of the exhaust cycle. Several things happen when the motor cannot fully exhaust itself. If the pressure is above atmospheric at the end of the cycle, the result is lowered volumetric efficiency, increased pumping losses, and reduced combustion efficiency as compared to an optimized exhaust cycle.

Swept Volume, Clearance Volume, and Compression Ratio.

In December's port timing article, I stated that top dead center, or TDC refers to the point at which the chamber is at its smallest possible volume. The space in the chamber at TDC is referred to as the clearance volume, and this in part determines the compression ratio. The compression ratio is specified as (Volume at BDC/Volume at TDC) Using an '87 13B as an example, the chamber volume at BDC is 9.4 times greater than the volume at TDC, for a compression ratio of 9.4 to 1. The difference between the volume at TDC, and BDC is referred to as the swept volume, or displacement. This is the volume of gasses that will be displaced in one complete cycle assuming 100% volumetric efficiency. A little bit of high school algebra shows that the volume at BDC is 44.66 cubic inches, and the volume at TDC is 4.75 cubic inches, or 10.6% of the total volume.

Volumetric Efficiency

The exhaust gasses that occupy the clearance volume will be carried around into the following intake stroke. As you can see, even at 100% volumetric efficiency the mixture will still only be 89.6% fresh intake charge. If the chamber pressure does not reach atmospheric by the end of the cycle, this 10.6 %, or 4.66 cubic inches of exhaust gasses will be pressurized, and will take up even more space once they are allowed to expand as the chamber volume increases during the intake stroke. This will reduce volumetric efficiency considerably, as the exhaust gasses will occupy space that could be used for fresh mixture. These exhaust gasses effectively "take away" from the swept volume, or displacement of the motor. The goal of the exhaust system then, should be to evacuate as much of the spent gasses as possible.

Inertial Scavenging

Inertial scavenging is easiest to understand if you think of the gasses in the exhaust system as a big piece of elastic. While they are not directly connected, a change at one end of the system will have an effect on the gasses at the other end of the system. For instance, towards the end of the cycle, the flow through the exhaust port slows down, but the high velocity gasses from earlier in the cycle are still travelling through the system. (Note: A system made up of 100" long, 1/34" inside diameter header tubes, as you might see on a race car, will contain about six complete cycles worth of exhaust gasses per pipe.) These high velocity gasses will "pull" on the slower moving gasses near the exhaust port, helping to evacuate the chamber. This is inertial scavenging. Just imagine two cars rolling down the road, connected to each other by a bungee cord. If the car at the back slows, it will not immediately be jerked back to speed, but rather gently pulled back up to speed by the car in front. As some of you may have guessed, a series of resonances will then occur, with each car alternately pulling at the other. This is very much like what happens to the gasses in the exhaust system.

Pumping Losses

Well, here we are at pumping losses again! Luckily this is quite easy to explain and understand. It all comes down to exhaust flow. Not just the airflow capability of the exhaust port, but of the entire system from the port to the end of the exhaust pipe. Quite simply, if the exhaust flow is insufficient, the blowdown period will only account for a small amount of the total exhaust gasses, and the remainder will have to be squeezed out by the rotor itself. Physically forcing the gasses from the chamber through a restrictive exhaust system requires a substantial amount of horsepower. So much in fact that many diesel truck engines have a mechanism which blocks the flow of exhaust gasses to slow the vehicle down, thus saving wear on the brakes. Just think about slowing an 18 wheeler with nothing but exhaust pressure, and you get an idea how much this can affect your engine.

Combustion Efficiency

We have already discussed how insufficient exhaust flow reduces volumetric efficiency, but the presence of exhaust gasses in the intake charge (Exhaust gas dilution) causes other problems as well. The rotary engine is known for its poor combustion characteristics. Due to the shape of the chamber, and the location of the spark lugs, a large percentage of the intake charge does not burn in the chamber. The end result is a fair amount of unburned gasses, or hydrocarbons being passed into the exhaust system. This reduces power output, because a portion of the mixture that we tried so hard to put into the engine did not burn. This also reduces fuel economy, and increases emissions. Another effect that is not often realized is excessive exhaust gas temperatures. These hydrocarbons will then burn in the exhaust system raising the exhaust gas temperatures.

The addition of exhaust gasses to the intake charge will reduce the already poor combustion quality. The end result is that the mixture is harder to ignite, and when it finally does light up it will burn at a slower rate further reducing power output. In a turbocharged engine excessive exhaust gas dilution will cause its own unique set of problems.

Detonation

We tend to think of combustion inside of the engine as a series of explosions, but in fact the combustion occurs at a very slow rate, at least compared to an explosion. In the absence of detonation, the mixture in the vicinity of the spark plugs is ignited first, and the "flame front" travels from that point, through the rest of the mixture in a fairly controlled manner. Detonation occurs after the combustion has initiated, and the pressure, and temperature in the chamber rises to the point that the remaining mixture literally explodes. Anyone who has ever experienced detonation understands that it certainly is an explosion! Detonation is caused by a combination of heat, and pressure, and so it stands to reason that excessive exhaust gas dilution, (remember these are hot gasses) will increase the likelyhood of detonation. As most of you know, detonation will destroy a turbocharged engine in a big hurry.

A "Perfect" Exhaust Cycle

Now that we have all of the pieces, it is time to put the puzzle together. I personally have a hard time understanding anything unless I can see it in front of me. For that reason I will refer once again to the illustration of the engine during its different phases.

As the exhaust port opens, (#13 in the illustration) the high pressure in the combustion chamber will force the gasses through the port and down the exhaust system at a high rate of speed. This, as you remember, is the blowdown period, and a large portion of the gasses will exit the chamber at this time. At the same time that the flow is initiated, a high pressure wave will travel towards the end of the exhaust system at the speed of sound. (Note that this high pressure wave will help to propel the slower moving exhaust gasses with it.)

Further into the cycle (#15) as the pressure differential between the chamber and the exhaust system has decreased, (ie., the chamber has "blown down") the velocity through the exhaust port will also decrease, and the remaining flow will be the result of the decreasing chamber volume. At this point, approximately half of the exhaust gasses will have exited the chamber.

At 135 degrees after bottom dead center, (between #15, and #16) the chamber will be at its maximum rate of decrease of volume. In other words, it is at this point in the cycle that the rotor will be travelling at maximum velocity. If the exhaust flow is insufficient, it will require a great deal of force to expel the gasses from the chamber. This is where the pumping losses during the exhaust stroke will be the greatest. Keep in mind that these losses cannot be eliminated, but they can certainly be lessened by providing sufficient exhaust flow.

Moving on to #17, and #18, the chamber volume is decreasing at a very slow rate, and the motor is doing very little to mechanically expel the gasses from the chamber. It is at this point in the cycle that pressure wave tuning comes into play. The high pressure wave that originated when the exhaust port first opened will have travelled to the collector, and been reflected back as a low pressure wave. (Remember last months section on pressure wave tuning?) If timed correctly, the wave will arrive at this point, just before the intake port opens. This low pressure wave, in conjunction with the "pull" created by the high speed gasses still in the exhaust system will lower the pressure in the chamber to sub atmospheric. When the intake port opens, this vacuum will help to initiate the flow of fresh mixture into the chamber, which will increase volumetric efficiency.

Looking back to December's port timing article, you can see that the intake port does not open until approximately 30 degrees after top dead center. That means that for the first 30 degrees after TDC, (The distance between #18, and #1 in the illustration) the chamber volume is increasing, but because only the exhaust port is open, the chamber will be filling with exhaust gasses by pulling them back out of the exhaust system. This is called exhaust gas reversion. If the exhaust gas velocity is low, (Such as at low rpm) the vacuum created by the increasing chamber volume can easily reverse the flow and pull the gasses back into the chamber. If, on the other hand, the exhaust gas velocity is high, it will take a great deal more energy to reverse their flow, and the result will be less exhaust gas dilution. This is why large exhaust ports, and large diameter exhaust tubing reduce low speed power.

Low RPM Operation

The above paragraphs describe a "perfect" exhaust stroke, and unfortunately this can only happen over a very narrow rpm range. Let's look at what happens when we halve the rpm. We will assume that the above example is at 8000 rpm. Now let's look at the same cycle at 4000 rpm. Since the exhaust cycle lasts twice as long at 4000 rpm, the chamber will have reached sub atmospheric pressure approximately half way through the cycle, assuming of course that we have sufficient exhaust flow. This sub atmospheric condition will send a low pressure wave travelling towards the end of the exhaust system at the speed of sound. (Remember that a pressure wave is intiated anytime pressure deviates from atmospheric.) This wave will reach the collector, and reflect back as a high pressure wave. Since we have halved the rpm, it is likely that this wave will arrive near the end of the exhaust stroke, (#18) and so the chamber pressure will be above atmospheric when the intake port opens. This will result in excessive exhaust gas dilution as compared to the 8000 rpm example. In addition to this, the exhaust gas velocity will be low, and during the period from TDC, to intake valve opening, the exhaust gas flow will reverse momentarily. This will also add to the amount of exhaust gas dilution.

If we wanted the exhaust stroke to be optimized for this lower rpm, several changes would be necessary.

1. Later exhaust port opening. Since we have more time to exhaust the chamber, the total exhaust duration can be lessend. The result of this will be that we can hold pressure in the chamber for a greater period of time. This will increase the amount of time that torque will be applied to the eccentric shaft.

2. Smaller cross sectional areas. Decreasing the cross sectional area of the port, and the exhaust tubing will increase the velocity of the exhaust gasses. This will result in less reverse flow, or exhaust gas reversion after top dead center, and will make the inertial scavenging towards the end of the cycle more effective.

3. Longer tuned lengths. Since the exhaust cycle occurs over a greater period of time at low rpm, the pressure wave must be further delayed if it is going to arrive at the appropriate time. In the case of optimizing the system for 4000, rather than 8000 rpm, the header lengths would need to be approximately twice as long. This is easiest to understand if you think of the headers as a delay source. What we are trying to do is delay the wave from the time it initiates to the end of the exhaust cycle. The further that the wave travels, later it will arrive at the exhaust port.

As you can see, we can only optimize the exhaust cycle over a fairly narrow rpm range. If at first this seems discouraging, it is important to consider that an optimized cycle over a narrow range is much better than a less than optimum cycle throughout the operating range. A "perfect" intake stroke can also only occur over a fairly narrow rpm range, and so it is important to consider the trade-offs when contemplating performace upgrades. If for instance you wish to "street port" your engine, you must understand that the increase in top end power will be accompanied by a decrease in low speed power.

The intent of these articles is not to make specific reccomendations, but to give you the knowledge to make informed decisions, and sort through the hype. For all of you racers, using the lessons learned from the exhaust system articles will allow you to make sense of exhaust tuning. If you apply these theories, and do some trial and error testing, you will likely unleash some hidden power. Now that you have the facts, you will understand why one system affects the engine differently than another, and this will make it much easier to arrive at the "correct" setup.

Until next month, thanks for reading, and stay tuned. There is much more to come. As for the topic of next months article...well, you'll just have to wait and see!

The Intake Cycle Part One

The last two articles explained the importance of an efficient exhaust cycle. As you now know, if the exhaust cycle is less than optimum, there is not much point in making other modifications. If, on the other hand, the exhaust cycle is optimized, improvements in the intake cycle will result in large power increases. Since the power output of the engine is directly proportional to the amount of air and fuel that it can ingest, the goal is to pack as much mixture as possible into the engine during the intake cycle.

Volumetric Efficiency and Charge Density

When considering intake efficiency, we normally think in terms of volume, or volumetric efficiency. Volumetric efficiency is stated as a percentage of the engines total airflow potential. (Total airflow potential, or pumping potential = displacement X rpm.) Volumetric efficiency is determind by measuring airflow into the engine while it is running. For instance, a 13B has a displacement of approximately 80 cubic inches. This means that it can potentially displace, or pump 80 cubic inches of air per revolution. At 6000 rpm, this equates to 480,000 cubic inches, or 277.8 cubic feet per minute. If measured airflow into the engine is also 277.8 cubic feet per minute (cfm) the engine is said to have 100% volumetric efficiency. (Volumetric efficiency = measured airflow/pumping potential.)

This is a convenient way to measure intake efficiency, but it is a bit misleading. What we are really concerned with is the mass of air and fuel that the engine can ingest. For instance, an engine operating at 100% volumetric efficiency at sea level will make much more power than an engine operating at 100% volumetric efficency at the top of Pikes Peak. In both instances, the engines are ingesting the same volume of air, but the engine operating at sea level will have taken in a greater mass of air. This is due to the greater air density at sea level. In more simple terms, one cubic foot of air at sea level weighs more than one cubic foot of air at Pikes Peak.

This is easy to understand if you consider the air in your tires. If one tire is inflated to 10 psi., and another tire is inflated to 50 psi., they both contain the same volume of air, but the tire with the higher pressure contains a greater mass of air. Air density, or mass, is directly proportional to pressure. At the higher elevation of Pikes Peak, there is less pressure, and therefore the air is less dense.

The Effect of Temperature

Temperature also has an affect on air density. The formula to determine air density in pounds per cubic foot is: Pressure in inches of mercury times 1.326, divided by absolute temperature in degrees farenheit. (Absolute temperature = temperature + 459.6) If, for instance your induction system draws intake air from under the hood, the intake air temperature on an 80 degree day can easily exceed 130 degrees. Using a barometric pressure of 29 Hg" for our example, the air density under the hood at 130 degrees is .0652 lbs. per cubic foot. Now if you change to a cold air setup which draws 80 degree intake air from outside of the car, the air density is .0713 lbs. per cubic foot This is an improvement of 9.3%! Using a 200 hp engine as an example, this is an improvement of 18.6 horsepower! In practice the horsepower improvement will be less than theoretical because the incoming air will be heated by the intake manifold, and the engine itself. Further improvement can be made by insulating the intake manifold so that it picks up less heat from the exhaust system, and radiator. It is probably not possible to achieve the theoretical density increase, but it should be clear that there is much to be gained by keeping the intake charge cool.

Air Fuel Ratio

With all of this discussion of airflow, and air density, it is important to understand why we need air in the first place. Quite simply, because without air, and the oxygen that it contains, the fuel will not burn. The optimum air fuel ratio varies slightly depending on the chemical makeup of the fuel, but generally, peak power occurs at a ratio of 13lbs. of air, to one pound of gasoline. Notice that this ratio is stated in lbs., and not volume. As you can see, we need much more air than fuel to develop peak horsepower.

Air weighs approximately .076 lbs. per cubic foot at sea level, and gasoline weighs approximately 46.7 lbs. per cubic foot. At an air fuel ratio of 13:1, you will need approximately 8,000 gallons of air for every gallon of fuel that you burn! When you look at it that way, it becomes obvious that the power output of an internal combustion engine is limited by the amount of air that it can ingest. If this doesn't convince you that airflow, more than anything else determines the success of a performance engine, nothing will!

Airflow

Total, or mass airflow into the engine is the single biggest factor in determining horsepower output. Put more simply, airflow is everything! The purpose of the intake stroke is to fill the engine with as much air as possible. There are many factors that contribute to the total airflow into the engine. The most important of these is the airflow capability of the induction system which includes everything from the air filter to the intake ports themselves. To optimize the airflow capability of the induction system, we must first be able to measure it. In more exact terms what we are measuring is the induction system's resistance to airflow.

Airflow measurement is done with a flowbench. The flowbench is the engine builders equivalent of a wind tunnel. The difference being that we are measuring flow restriction through a passage, rather than around a stationary object. A flowbench consists of a series of vacuum motors which create a pressure drop on one side of the passage being tested. The pressure drop causes air to flow through the passage, and that flow is then measured. In this way, the resistance to airflow can be determined, and the results of modifications can be quantified.

Pressure Differential

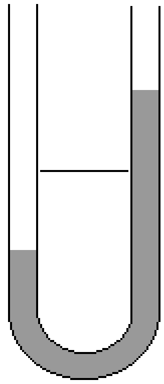
Airflow is initiated by a pressure differential. A pressure differential can be most simply described as a pressure difference between two points. Pressure, at least as it applies to the automotive field, is normally stated in psi. (Pounds per square inch), Hg" (Inches of mercury), or H2o" (Inches of water)

The earliest pressure measurement devices were simple manometers. A manometer is made up of a clear tube formed into the shape of a U, and partially filled with liquid. If there is no pressure differential between the two ends of the tube the liquid will fill both sides of the tube by the same amount If there is a pressure differential between the two tubes, the liquid will flow from the high pressure side of the tube to the low pressure side. The difference of the fluid height between the two sides of the tube is the pressure differential, stated in inches, millimeters, or whatever unit of measurement you prefer. If the tube is filled with water, the pressure drop is stated in inches of water, if it is filled with mercury...well you get it.

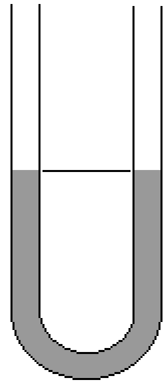
Atmospheric pressure at sea level, at 59 degrees farenheit is 1bar., 14.7 psi, 29.92 inches of mercury, or 406.9 inches of water. This is referred to as SPST, or standard pressure, standard temperature. From this you can see the realtionship of these different units of measure.

Intake Pressure Drop

Left side of tube above atmospheric pressure.



Both ends of tube at barometric pressure



During the intake stroke the chamber volume increases. Since the mass of the air in the chamber is the same, but its volume has increased, its density has decreased, and so then has its pressure. (Remember, air density is directly proportional to air pressure, and air pressure is directly proportional to air density.) The result is a pressure drop, or a pressure differential between the chamber, and the air at barometric pressure outside of the engine. This initiates air flow into the chamber

Airflow Restriction

As I mentioned earlier, when airflow is measured on the flowbench, it is really the restriction to airflow that is being measured. If for instance ther were no restriction to airflow, a pressure drop could not be realized. To be able to compare the airflow of one induction system to another, flowbench testing is usually done at a standard pressure differential. The most commonly used pressure drop for flowtesting is is 25 inches of water.

The pressure drop caused by the engine itself is not steady like that of the flowbench. In fact, the pressure varies a great deal throughout the intake cycle. There are many factors that determine the pressure throughout the cycle, (These factors will be discussed in the second part of this article.) but for the sake of simplicity, let's consider it as being steady.

Flowbench testing shows that the complete induction system of a second generation six port 13B flows approximately 125 cfm. at a pressure drop of 25 H2o" With a little bit of porting, this can be increased by approximately 10%, or 12.5 cfm. This 10% airflow increase will be true at any pressure drop. For instance, at 6 H2o", the stock ports flow 61.2 cfm, and the reshaped ports flow 67.4 cfm, for a gain of 6.2 cfm. In both cases, the result is a 10 percent increase in airflow. So as you can see, regardless of the actual pressure drop throughout the intake cycle, the increase of total airflow into the engine will equal 10%. In practice, the power increase is not always equal to the airflow increase, but it does follow quite closely. A few of the factors that affect the result of increased airflow are; heating of the intake air, pumping losses during the compression and exhaust stroke, and combustion efficiency.

Next months article will discuss the factors that affect the pressure differential throughout the intake cycle, and some tips on increasing the airflow potential of your induction system.

The Intake Cycle Part Two

OK, here we go with part two of the intake cycle. To all of you who regularly follow these monthly articles, I apologize for making you wait so long for the last half.

The previous article described the importance of increasing airflow into the engine. In this article I intend to discuss the many factors that affect total intake airflow, and horsepower output.

Total Intake Airflow

In the previous article I stated that the horsepower output of an engine is directly proportional to the amount of air and fuel that it can ingest. It stands to reason then, that if the goal is increased horsepower, we must increase the airflow potential of the induction system. The total airflow through any passage (Ports, intake manifold, etc.) is affected by three variables. These variables are: 1. The size of the passage. 2. The pressure differential between the inlet and the outlet. 3. The coefficient of discharge of the passage.

Coefficient of Discharge

Coefficient of discharge is normally stated as a percentage, and is a measure of how efficiently a passage will allow air to flow through it. A "perfect" venturi, having an inlet angle of 16°, an outlet angle of 7°, and an inlet and outlet area four times larger than the operating cross sectional area (Vena contracta, the smallest point of the venturi.) has a flow efficiency of 100%. A venturi such as this would flow 137.7 cubic feet per minute (CFM), per square inch, at a pressure differential of 25 inches of water. Using this as a baseline, we can determine the efficiency of a port, or intake manifold runner. 100% flow efficiency is not possible in most cases, but knowing the efficiency of a given combination is a large step towards being able to optimize it.

For example, consider two intake manifolds that flow the same amount of air at the same pressure differential, but one of these manifolds has a runner cross sectional area twice that of the other. The manifold with the larger runners will have a coefficient of discharge that is half that of the smaller manifold. In addition to being less efficient in technical terms, the larger manifold will also lower the horsepower output of the engine as compared to the smaller manifold. Why you ask?

Velocity

Hopefully all of you remember the previous discussion of velocity, and have a good understanding of its effects on unsteady airflow. If not, go back to part one of the exhaust cycle article for a freshen up. As I stated in that article, flow velocity through the exhaust system is not steady, and in many cases, the flow will reverse at some point in the cycle. This is also true of the intake system.

Pressure Wave Tuning

Since most rotary applications utilize a stock, or off the shelf aftermarket manifold, manipulating the pressure waves by changing the length of the induction tract is not as practical as with the exhaust system. For this reason I will not cover this in great detail.

The pressure wave theories that I discussed in the exhaust article apply to the intake system as well, but there are a few differences between the two. 1. The pressure waves will be much weaker, and so their effect will not be as great. 2. Since the intake manifold is typically much shorter than the exhaust system, the pressure waves will be reflected back and forth several times before they arrive at the intake port at the appropriate time in the cycle. Each time they reflect, they will lose some energy which reduces their usefulness. 3. In the case of the induction system, it is the positive, or high pressure waves, rather than the negative, or low pressure waves that are useful for increasing horsepower.

By timing the positive return wave to arrive at the intake port right before it closes, the pressure differential between the port, and the chamber will be increased. This will increase the flow into the chamber at the end of the cycle when it is typically at it lowest.

There are a few basic rules that apply to pressure wave tuning the induction system. A longer manifold will delay the waves for a greater period of time, and so tune the manifold for a lower rpm range, just as with the exhaust system. A longer manifold will also increase the peak torque output of the engine, in addition to the above mentioned effects. This is the result of the manifold containing a greater mass of air. (Remember, energy = mass times velocity squared.) At the end of the intake cycle, when the chamber pressure is increasing, this greater mass (Which is travelling at a high velocity) will better overcome the rising chamber pressure, resulting in greater airflow during that critical period. Additionally, a greater pressure drop will be created at the beginning of the cycle when the chamber begins to expand, because the engine will have to "pull" harder to get this greater mass of air moving. It is this initial low pressure condition which starts the pressure wave cycle, and the result is a pressure wave of greater intensity which if timed correctly, will increase volumetric efficiency.

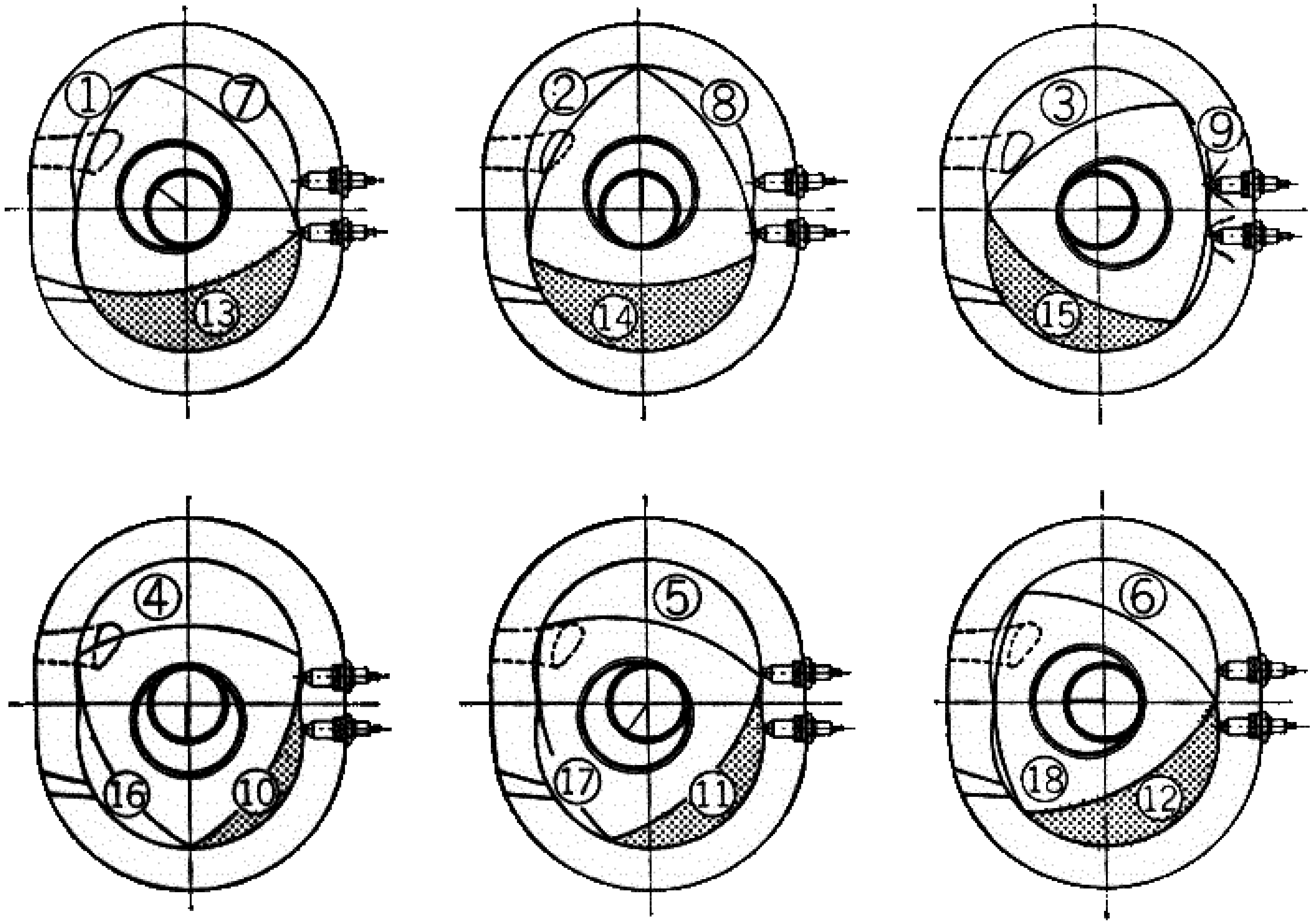
The Peak Horsepower Myth

If you are thinking to yourself that high rpm horsepower is all that matters for your application, consider this. Even with a close ratio racing gear box, you will need to make power over a range of at least 2,000 rpm. If the engine makes a staggering amount of horsepower at redline, but drops off quickly below that, acceleration will suffer. This is relatively common on race engines, and is the result of low velocity, or poor flow efficiency.

Peak horsepower is a measure of the absolute maximum horsepower that the engine can produce. It is a relative indication of an engines performance, but it only tells you what the engine is doing at one particular rpm. It tells very little about the actual performance, unless you will only run the engine at one rpm!

What is important is the average horsepower throughout a specified operating range. This operating range should be specified based on the gear ratios of the transmission.

I have presented quite a bit of information here, and to make all of this easier to visualize, I will once again refer you to the illustration of the rotary engine during its different phases.



1. 45° after TDC. The chamber is slowly expanding, and the air/fuel mixture is just starting to enter the chamber. This is the beginning of the intake cycle for a conventional side port engine, and the intake port has been open for approximately 20°.

A bridge port, or peripheral port engine will have had the intake port open for 150° to 200° at this point. If the exhaust system is working properly, the low pressure wave will have arrived, and initiated intake flow by TDC, or even sooner, replacing the exhaust gasses in the clearance volume with fresh intake charge.

2. 90° after TDC. The rate of expansion is now fairly rapid, and the low pressure in the chamber initiates a negative pressure wave in the induction system.

3. 180° after TDC. The intake port is completely open, and the point of maximum rate of expansion and flow has occured 45° earlier.

4. BDC At this point, the chamber is at its maximum volume, and past this point the chamber volume will decrease as the compression cycle begins.

5. and 6. 45°, and 90° after BDC. It is during this period that the intake port will close, and the effects of inertial supercharging become critical. The period between BDC, and intake port closing has the greatest effect on the volumetric efficiency of the engine. Velocity, velocity, velocity!

Well, I think that just about covers it. For a very thorough understanding of the gas exchange process, review the previous exhaust cycle articles, and consider the intake and exhaust as a complete cycle. Pay careful attention to the entire process, starting with the exhaust cycle, and the pressure condition that is left at TDC, which is where it all starts.

Paul Yaw
Yaw Power Products

Combustion and Ignition Part One

Now that we all understand what is involved in pumping air through the motor, it is time to delve into the portion of the cycle that actually produces power.

The combustion process is probably the least understood aspect of modern internal combustion engines. Rotary engines have not had the benefit of extensive combustion research as piston engines have.

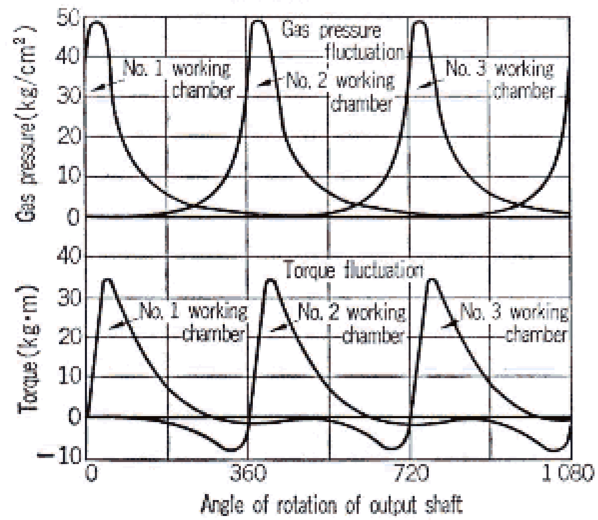
The point of filling the chamber with combustible mixture is of course to set it on fire, and convert this heat energy to torque at the eccentric shaft, as efficiently as possible. This seems simple enough on the surface, but it is every bit as complex as the other cycles, and just as important

The Big Bang!

The first step towards understanding the combustion process is the realization that the air fuel mixture does not explode when the spark plug fires. If this were to happen, the engine itself would explode within seconds. Normal combustion is characterized by the flame front travelling smoothly from the point of ignition to the outer edges of the combustion chamber. The best analogy I have heard, is that the flame travels much like dry grass burning in a field, with the starting point being equivalent to the spark plug location. (Please, no field burning experiments!)

Peak Cylinder Pressure, Maximum Torque Transfer, and Ignition Timing.

As the spark plug fires, initiating combustion, the fuel burns, and the pressure in the chamber increases. It is this pressure, reacted through the rotor, that spins the eccentric shaft creating horsepower. At approximately 45 degrees after top dead center, the rotor has the greatest mechanical advantage over the eccentric shaft. This is the point of maximum torque transfer to the eccentric shaft.



It is often stated that to maximize power output, peak cylinder pressure should occur at this point. This is a *gross* oversimplification, and does not consider the average torque transfer throughout the cycle.

This illustration (From "The Rotary Engine", by Kenichi Yamamoto) shows gas pressure and torque fluctuation throughout the expansion cycle. Note the torque reversal before TDC shown on the bottom chart. This illustrates the power required to compress the mixture as the chamber nears TDC. Maximum output will occur when the ratio of cylinder pressure after TDC, to cylinder pressure before top dead center is the greatest. If for instance we had the option of manipulating the cylinder pressure, we would want to place all of the pressure after TDC, centered at the point of maximum torque transfer, with none of the pressure ocuring before TDC.

In the real world we do not have this option because the fuel does not burn fast enough to complete the combustion process within such a narrow range. In order to achieve the greatest cylinder pressure during this period of high torque transfer, we must ignite the mixture well *before* TDC. At full throttle, ignition occurs at approximately 25 degrees before TDC. In that period between ignition, and TDC, the chamber volume is *decreasing,* compressing the mixture that has just been ignited.

Compressing this burning mixture (Which is expanding.) requires a certain amount of power, but this power loss if offset by the increased pressure "Under the curve" after TDC.

If you look closely at the torque fluctuation chart, you can see that the rotor *effectively* transfers power to the eccentric shaft over a very narrow range. If you give this a bit of thought, you will realize that the power output of the engine can be increased by increasing flame speed. Increasing the flame speed will release the energy from the fuel in a shorter period of time. This will allow more effective use of this energy by timing the combustion process to more closely match the torque transfer curve of the rotor/eccentric shaft combination.

3...2...1...Ignition!

Mazda's description of ignition goes something like this: "The air/fuel is ignited when the mixture in the spark gap, electrically energized sufficiently to resist the heat loss in its vicinity, is heated and ionized exceeding it's ignition point." A more simple explanation is that the spark heats the fuel until it is hot enough to burn. The important part of Mazda's explanation is that "Enough heat must be generated to resist the heat loss in its vicinity." As the plug fires, heat from the spark will be absorbed by the air/fuel charge in the gap, the spark plug itself, and the relatively cool metal of the rotor housing. Or in Mazda's words, "Everything in its vicinity."

The heat absorption into the rotor housing, and more importantly the spark plug, increases the spark energy required to ignite the fuel. While this energy is easily produced by a good ignition system, it is important to understand that heat absorption greatly affects the ability to ignite the fuel.

Flame Travel.

Once the mixture in the spark gap has been ignited, the flame will contine to spread by the same process of successively heating gasoline molecules to their ignition point. This heat transfer from one molecule to another is in effect, the definition of combustion.

Now that you understand what can be gained by increasing the rate of combustion, it is time to discuss the several factors that affect the combustion rate.

1-Air Fuel Ratio

Flame travel through a gasoline and air mixture will be fastest within a fairly narrow mixture range from 11.5:1 to 13.5:1. Flame speed will decrease rapidly outside of this range. This varies a bit from one fuel to the next, but the above stated range is a generally accepted estimation.

2-Exhaust Gas Dilution

The presence of exhaust gasses in the mixture will slow the flame travel considerably, because the exhaust gasses will not burn. The heat transfer that was described above will be slowed, because these gasses will absorb a portion of the heat needed to ignite the fuel molecules. This same situation occurs when a high percentage of water is present, such as on a very humid day.

At idle, the percentage of exhaust gasses in the mixture can exceed 70%. At high rpm, if the exhaust flow is insufficient, the result will be excessive exhaust gas dilution. In addition to slowing the flame travel, this exhaust gas dilution will also increase the occurence of misfires, and "weak fires." (Incomplete combustion.)

Note:Even at 100% volumetric efficiency, approximately 10% of the chamber volume will be made up of exhaust gasses from the previous cycle. (Rememebr the clearance volume discussed in Part Two of the exhaust cycle article?)

3-Charge Density

The density of the charge also has a substantial affect on the flame speed. Quite simply, a denser (Greater mass per unit volume.) charge will burn at a faster rate. Higher density will increase the speed of heat transfer from one molecule to the next because there will be less dead space between these molecules.

Higher charge density can be achieved by increasing volumetric efficiency, decreasing charge temperature, or increasing the compression ratio. All three of the above conditions will increase the power output, and thermal efficiency of the engine.

4-Mixture Distribution.

It is easy to assume that the air and fuel molecules are thoroughly mixed, but in the real world, the distribution is less than optimum. As was stated earlier, the fastest flame travel occurs within a fairly narrow range of mixture ratios. If the air/fuel ratio is excessively rich in some areas, and excessively lean in others, the flame will spread in an irregular manner. This condition slows the combustion process.

Poor mixture distribution causes problems in addition to slowing combustion. It is not only possible, but normal to have a mixture in the spark gap that is too rich or lean to fire efficiently. This condition is most likely at low engine speeds when there is insufficient velocity to keep the fuel droplets in suspension, but it happens at higher speeds as well. The result is at best, incomplete combustion, and at worse, a complete misfire.

Complete misfires happen more often than you may think. At 6,000 rpm, each chamber will complete 100 cycles per second! At this rate, you could have several misfires in a row, and not interpret it as such. At best you would sense that engine was running rough.

5-Atomization

What is commonly referred to as atomization, is actually large globs of fuel dispersed through the air. It is *far* from being atomized. Still, it is the commonly accepted term, and has come to refer to the size of the fuel droplets. Since the fuel must be in the presence of oxygen to burn, you can see that only the molecules on the outer "skin" of these fuel droplets will be able to burn unless something is done to break them into smaller particles.

As the flame travels through the mixture, the heat will vaporize the fuel droplets, and allow the greater percentage of fuel to come in contact with oxygen so that it will burn. The problem here, is that this vaporization is the result of heat absortion. The result is decreased flame travel speed, as much of the heat energy is being used to vaporize the fuel. If the fuel is not sufficiently vaporized when it enters the chamber, the ignition system will not be able to generate enough energy to initiate combustion!

This may seem unlikely to some of you because gasoline is considered by most to be extremely volatile. The fact is that until it mixes with air in the correct ratio, (Approximately 10:1 to 20:1) it will not ignite.

The biggest contributor to poor atomization is low inlet velocity. While this is much more of a problem with carbureted engines, fuel injected engines will also suffer from poor atomization if inlet velocity is excessively low. In the case of modern emission controlled engines, the injectors are placed as close to the combustion chamber as possible to reduce fuel "drop out" as the mixture travels from the injector to the combustion chamber.

To help illustrate this point, consider the way that a fuel injected car runs when the injectors are dirty the atomization is poor!

6-Heat Absorption

The final consideration is the relatively cool metal parts that make up the combustion chamber. The combustion chamber is made up of the face of the rotor, and the inner surface of the rotor housing. Both of these surfaces are kept relatively cool by oil and water. These cool metal surfaces will absorb heat during the combustion process, slowing the flame travel. In addition to decreased flame speed, the air fuel mixture immediately adjacent to these surfaces *will not burn!*

These cool surfaces will absorb enough heat from the air fuel mixture to keep it from reaching its ignition temperature. The rotary has a tremendous amount of surface area as compared to a piston engine of the same displacement. This is referred to as the surface to volume ratio. The rotary engine has a *very* high surface to volume ratio, and its poor thermal efficiency is due to this.

Cyclic Variations

The mixture distribution, and so the combustion efficiency, will vary a great deal from one cycle to the next. This is due to the random nature of both fluid flow, and combustion. It has often been stated in automotive literature that the spark energy required to ignite homogenous air fuel mixtures between 11.5:1, and 13.5:1 is very low. This statement is normally taken one step further to suggest that once this minimum spark energy requirement has been met, there is nothing to be gained by using high current ignition systems.

This may be true in simple laboratory tests, with homogenous mixtures, but it ignores the dynamic nature of internal combustion engines. The fact is, that not only does the mixture quality in the spark gap change from one cycle to another, it is also changing *during* the cycle. For this reason, long duration, or multiple sparks are very beneficial to combustion efficiency. This is especially true in rotary engines where the air fuel mixture is traveling past the spark plug at a high rate of speed when it fires.

Conclusion

With several articles under your belt, it should now be obvious that the rotary engine (Or any internal combustion engine for that matter.) is a very complex device. Nothing illustrates this more clearly than the fact that exhaust efficiency, which seems completely unrelated, has a considerable effect on combustion efficiency.

A successful performance engine is the result of optimizing many different variables from header lengths to spark plug selection, and something as simple as intake velocity will affect not only low speed power, but high speed power as well through its effect on mixture quality.

The key is to **understand** why one variable affects another.

Paul Yaw
Yaw Power Products

Timing

Distributor Advance Mechanisms

If your car has a distributor, it has two mechanisms that advance the timing based on manifold vacuum, and rpm. Let's start with the vacuum advance. There are two vacuum pots on the side of your distributor. One for leading, and one for trailing. Each of the pots is referenced to manifold vacuum through a vacuum line or hose.

When the throttle plates are closed, or just partially open, there is high vacuum in the manifold. The reason is that the engine, which is simply a pump, is spinning away trying to draw in air, but since the throttle is blocking airflow, a great deal of suction or vacuum is present in the manifold. This is the equivalent of plugging the hose of your shop vac with the palm of your hand. The motor is spinning, trying to move some air, but since you are resisting the airflow, a vacuum, or more specifically, a low pressure is created.

This vacuum, actuated through the vacuum advance pot will advance the timing based on the intensity of the vacuum. As you open the throttle, the vacuum lessens, and the timing advance decreases. At full throttle (Butterflies all the way open.) the vacuum advance has no effect on the timing, because there is no manifold vacuum.

Centrifugal, or "Mechanical" Advance

In case you were wondering, centrifugal advance, and mechanical advance are two different terms for the same mechanism. The centrifugal advance will advance the timing based on engine rpm. It consists of two weights held in place by springs which are attached to the distributor shaft. At low rpm, when the shaft is spinning slowly, the weights are held in place by the springs. As the rpm increases, the centrifugal force of the spinning weights overcomes the spring tension, and allows the weights to move outward. These weights are attached to the advance mechanism in such a manner that as they move outward, the timing of both leading and trailing will be advanced. This advance will begin at about 1,500 rpm, reacing maximum or "total" advance of approximately 20 degrees by 4,000 rpm, at which point no further advance will result from increasing rpm.

Why All the Complexity?

The timing requirements of the engine vary, based on charge density, rpm, and..here's the biggy...emissions requirements! It is common to eliminate both advance mechanisms so that the engine will be at full advance at all times. You will have a few less things that can fail, and setting the timing will be less of a hassle. It is manily for these reasons that race cars run "locked" timing.

Setting Your Timing For Best Power

Now for the fun stuff. Most stock or mildly ported 12A's will make best power with total timing of 24 degrees before top dead center leading, and 16 degrees before top dead center trailing. 13B's (GSL-SE.) Will normally run best at 26 leading, and 16 trailing. The timing for best power will vary slightly from one engine to another, but a leading/trailing split of 8 degrees works best in almost all 12A applications, and 10 degrees for a 13B. Since it is the total advance that we are concerned with, this creates a problem because your stock pulley is marked at top dead center, and twenty degrees after top dead center which is the factory recommended setting at idle.

$60 Pulley?

You don't have to buy an expensive aftermarket pulley just to have the appropriate timing marks. It is quite simple to make additional marks on the stock pulley.

The first step is to remove the stock pulley. This comes off easily by removing the four 10mm. screws that hold it in place. Some of the stock pulleys are indexed, and will only go on one way. Others can be placed in any of four positions, (90 degrees apart.) so I recommend spinning the engine to top dead center before removing the pulley so that you don't put it on wrong.

You will need a pair of vernier calipers, a small file or hack saw blade, a roll of masking tape, a calculator, and some brightly colored paint. If you do not own a pair of calipers, this is just the excuse you need to buy a set. Most retail tool supply stores carry them. They range in price from $25 to over $200. Luckily, the $25 calipers are more than accurate enough for this application, and once you have a set, you will be amazed at how often you use them.

Do not buy a set of plastic calipers from Sears or Home Depot. These are absolute crap, and most of them are colored to look like stainless steel so you will buy them, only to find that they are pieces of shit once you get them out of the package. A good set will have a small dial gauge, and come in a protective plastic box. Note: Amtos is a good inexpensive brand.

Step by Step

1. Measure the diameter (Distance across.) of the pulley. This should be approximately 4.5 inches.

2. Take the diameter, and multiply it by 3.14. This will give you the circumference (Distance around.) of the pulley

3. A full circle is divided into 360 degrees, so if you divide the circumference by 360, the answer will be the distance around that equals one degree. For this application, 5 degree increments will be fine, so divide the circumference by 72. The answer will be the distance around the pulley that is equal to 5 degrees.

Example: If the pulley is 4.5" in diamter, 4.5 times 3.14=14.13. 14.13 divided by 72=.196. so .196 = 5 degrees.

4. Tear off a piece of making tape approximately 3 inches long, and stick it to a smooth flat surface. Make a mark at the far left side on the edge nearest you, and label it TDC, for top dead center. Using your shiny new calipers, set them at 5 degrees, which using the above example is .196 inches. Make a mark .196 inches to the right of the TDC mark. This is your 5 degree mark. Now multiply .196 by two (.393). Set your calipers to this number, and make another mark .393 inches to the right of the top dead center mark. This is your 10 degree mark. Now simply continue to multiply the 5 degree number by 3,4,5, and 6 to get your 15, 20, 25, and 30 degree marks.

You are probably thinking screw that, I'll just set the calipers at .196 (5 degrees) and measure from my last mark. Doing it that way will multiply your errors as you get further from TDC. If for instance your mark is off by .005", and you make the same error every time, by the time you get to 30 degrees, you could be off by several degrees! Take your time and make it accurate!!!

5. Stick the tape to the pulley lining up your TDC mark **exactly** with the TDC mark on the pulley. Looking at the pulley from the front, with TDC on top, your tape marks should be to the right of TDC, and the factory 20 degrees after TDC mark (Trailing.) should be to the left.

6. Now take a small square, or triangular file and make a notch on the pulley at each of your marks. Make sure that your notch is exactly centered on your pencil mark.

7. Dab some brightly colored paint into each of the notches. If you are using a spray can, you can spray the notches, and wipe off the excess with some solvent and a rag. If you lightly wipe the excess, the paint will remain in the low spots that you filed into the pulley.

8. Put the pulley back on the engine and you are done.

Setting the Timing to Total Advance.

Now that you have marks on the pulley, rev the engine above 4,000 rpm, and set the timing at the appropriate spot. I mentioned 24, and 16 which of course is not marked on the pulley. It is easy to eyeball this and arrive at any number you wish. If you are concerned about accuracy, you can add as many marks as you want to the pulley, but I have found that marks every five degrees are sufficient.

Final note: Turning the distributor changes the timing of the leading and tailing by equal amounts. You should set the leading first, and then adjust the vacuum pot to set the trailing.

Don't be afraid to try different timing settings. Every combination is a bit different, and detonation is not a concern in a naturally aspirated engine. If you do experience detonation, and the leading is less than thirty degrees advanced, you probably have excessive carbon buildup inside of the engine. Redline injector cleaner will do a good job of reducing this buildup, although it may take several tanks to completely clean the engine.

Flow Testing – What is it, and Why is it Important?

Flow testing has become a hotly debated subject in the rotary world. The mailing lists, and forums are full of discussions, arguments, and full out attacks, mostly debated by individuals who have never even built an engine, let alone, had to prove its performance on a dynamometer, or a race track. Unfortunately, these discussions are fueled by a few uninformed "Engine Builders" who feel the need to justify their back yard approach to engine building.

The fact is that flowbench testing is a very important part of engine building, and anyone who is not using a flowbench to develop their ports/manifolds is missing out on some very valuable information.

Let's consider a few obvious points.

Everyone interested in increasing horsepower will discuss, or at least consider flow.

Everyone talks about it, even the guys who claim that flowtesting is a waste of time. "I need bigger venturis because they flow better", or "A K&N; filter flows better than stock." We all agree that increasing flow is a good thing, but, **if you can't measure it, how do you know it actually flows better?**

Even though you can't see, feel, or hear how well a port flows, there are those who claim that **measuring** the flow is not valid.

If taking an actual measurement of the flow is not valid, I don't know what is. Imagine taking the same approach to the rest of the engine build.

How would you set apex seal clearance without measuring the seal and the rotor housing with a micrometer? You cannot see the difference between a seal that is 3.1436" long, and one that is 3.1446" long. You also cannot see the difference between an exhaust port that flows 270 cfm, and one that flows 240 cfm.

Anyone who tells you that they can do either of the above is either lying, or delusional.

To take this concept to its most ridiculous extreme, imagine that you ask your engine builder for a quote, and he holds up two fingers and says "It will take a stack of bills about this high..." Ones? Twenties? Euro's? This high... WTF?

As ridiculous as that seems, it is no different than randomly grinding an intake runner, and then proclaiming "It flows more now!"

In many cases, it is quite possible to enlarge a passage and make it flow worse!

Let me give you a few examples.

1. Port matching the intermediate runners on a stock 12A intake manifold. This absolutely ruins the flow. Additionally, the velocity is reduced, and so both high, and low rpm power is reduced.

2. Removing the injector boss in a 6-port intake manifold. Reshaping, and reducing the height of this slightly will improve the flow a considerable amount, but if you go too far, the flow will decrease, and you will end up with a port that flows less than when you started.

3. Cutting a large radius on the roof of an exhaust port. (12A, or 13B) A generous radius is helpful if the exhaust port is close to stock size, but if it has been widened considerably, a large radius will absolutely kill the flow. This is one of the most sensitive areas of any of the ports in a rotary engine, and you can lose 10% to 15% flow in a hurry. The difference between a .050" radius, and a .070" radius can easily cost you 25 cfm.

So how do I know this? I **measured** it! There is no other way that I could know. I found out that these things didn't work because I did what **looked** right, and then measured the results, only to find that I could not have been more wrong.

The first time that I tested a stock 12A intake manifold with port matched intermediate runners, I couldn't believe the results. I actually spent the next half hour looking for leaks in the flowbench! Over the years, I have checked for leaks a few more times after seeing results that I couldn't believe. The point is that you just can't guess where the air wants to go, and just to make sure we are all on the same page, **if you are not measuring, you are guessing.**

This is the nature of airflow, and this is the reason that Boeing, NASA, F1 teams, and auto manufacturers still use a wind tunnel to develop their designs. These companies have entire teams of engineers with more airflow knowledge, and experience than we can imagine, and they still go to the wind tunnel to prove, test, and optimize their designs.

If that doesn't convince you that measuring is far superior to guessing, nothing will change your mind, and you should not read any further.

Enough of the Why, What about the How?

This is our flowbench. We designed and built this specifically for our own needs , rather than adapting a unit designed for testing piston engine heads. With this unit, we can test the flow of intake ports, exhaust ports, carburetors, throttle bodies, and even complete exhaust systems.

We have many different fixtures for testing a variety of pieces. This picture shows the unit with the exhaust port fixture attached. (The flanged tube in blue.)

The intake fixture is much more complex because we flow through a complete chamber which consists of a modified rotor, and rotor housing. This allows us to accurately simulate the flow patterns that exist inside the motor. It also allows us to add an induction system so that we can see how much an intake manifold, carburetor, or even an air filter will affect the total intake flow.

To learn more about how a flowbench works, check out the **Superflow** website.

For all the excitement surrounding flowtesting, a flow bench is actually a very simple device, and the "technology" used for measuring airflow has been around since the 1800's.

When we flow test a part, we have to have a way to quantify the results, just as if we were measuring the length of a 2X4 with a measuring tape. The flow potential of a port, carb etc. is described as the amount of air, in cubic feet per minute, that will flow through it with a given pressure differential. (Normally referred to as the test pressure.)

Pressure differential is just another way of saying that there is greater pressure at one end than the other. This is as simple as sucking Pepsi through a straw. When you suck on the straw, you are creating a partial vacuum, the result being that the pressure is greater at one end as compared to the other. Once you create a pressure differential, you have flow.

The first thing that is needed to test airflow is a pressure differential, or more simply, a way to move a lot of air. A flowbench normally has several small vacuum motors, much like what is used in a standard shop vac. (My first flow bench was actually built from a large shop vac.)

This vacuum source can be used to either suck through, or blow through the piece being tested. Either approach is valid, as it is the pressure differential that determines the flow, not the absolute pressure.

The airflow is measured by passing the air though a calibrated orifice, and then measuring the pressure differential across the orifice. This "calibrated orifice" is normally just a thin metal plate, with a round hole in it (Pretty high tech huh?) and most flow benches have several orifices of varying diameters so that different flow ranges can be accurately measured.

Since the pressures that we are dealing with are normally quite small, they are measured with a manometer, rather than a gauge. A manometer is a very simple device which uses the force of gravity, and a fluid of known mass to measure pressure.

For a better understanding, have a look at our Intake cycle article.

What follows is a pictorial guide to porting, and flow testing an exhaust port.

And Finally, How This Applies to Your Motor!

The first thing to consider is that the horsepower output of any internal combustion engine is determined **more than any other factor** by the airflow potential of that engine. To describe it simply, all the power comes from the heat energy of the combustion of air and fuel. Getting enough fuel into an engine is not a problem, because fuel is quite compact, but along with that fuel, we need an appropriate amount of air to go with it. Filling the chamber with air at 9000 rpm is quite a task, because we only have **5 thousandths of one second** to do so! Even at idle, the intake cycle only lasts 6 hundredths of a second!

As you can see, we must have very efficient flow passages if we are to get a reasonable amount of air into the engine. By now it should be obvious that filling the engines chambers with air and fuel is not a simple thing, and it is somewhat complicated by the fact that we cannot see, feel, smell, or even guess how efficient our flow path is. Luckily, we have the flowbench which will tell us **exactly** how efficient our flow path is. Without this tool, you are left to guess, and you will spend a great deal of time chasing your tail.

Enough Words, How About Some Pics.

A picture is worth a thousand words right? We recently disassembled a 12A for a local customer who felt that it was in need of a freshen up. We were asked to do what we could to improve the power while we had it apart, and so we decided to lighten the rotating assembly, and spend some time on the ports.

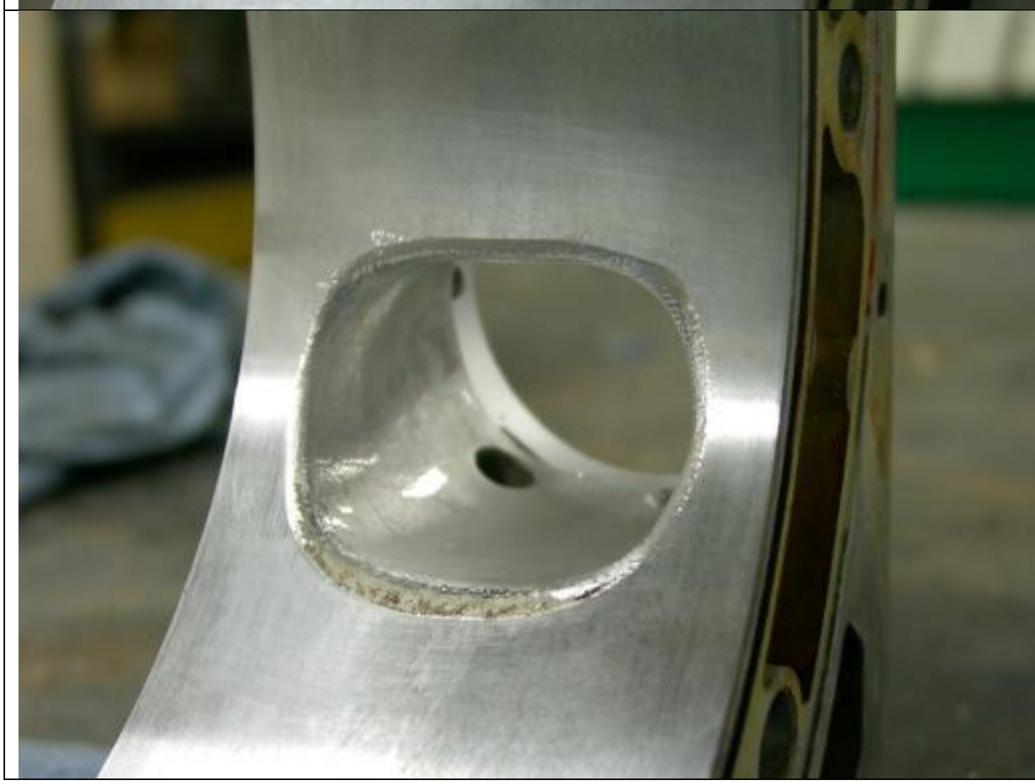
We never know what to expect when we get inside of a competitors piece, but this motor gave us a pleasant surprise. The ports looked a bit rough, but flowed fairly well. This seemed like a great opportunity to give a "guided tour" of the porting and flowtesting process.

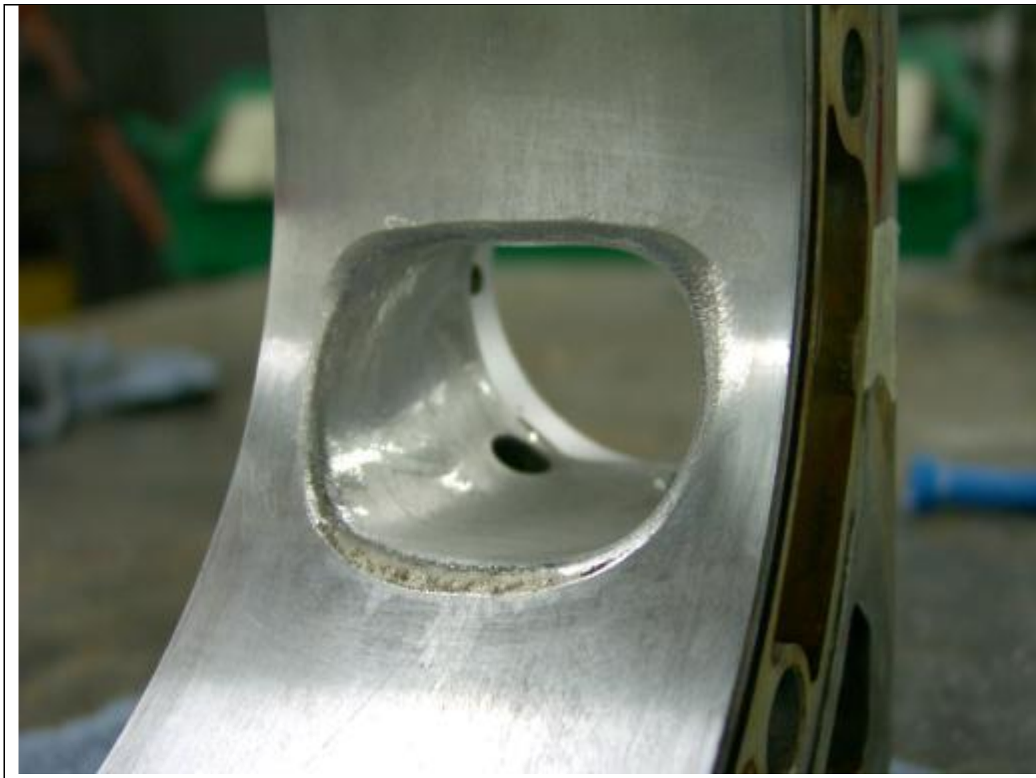
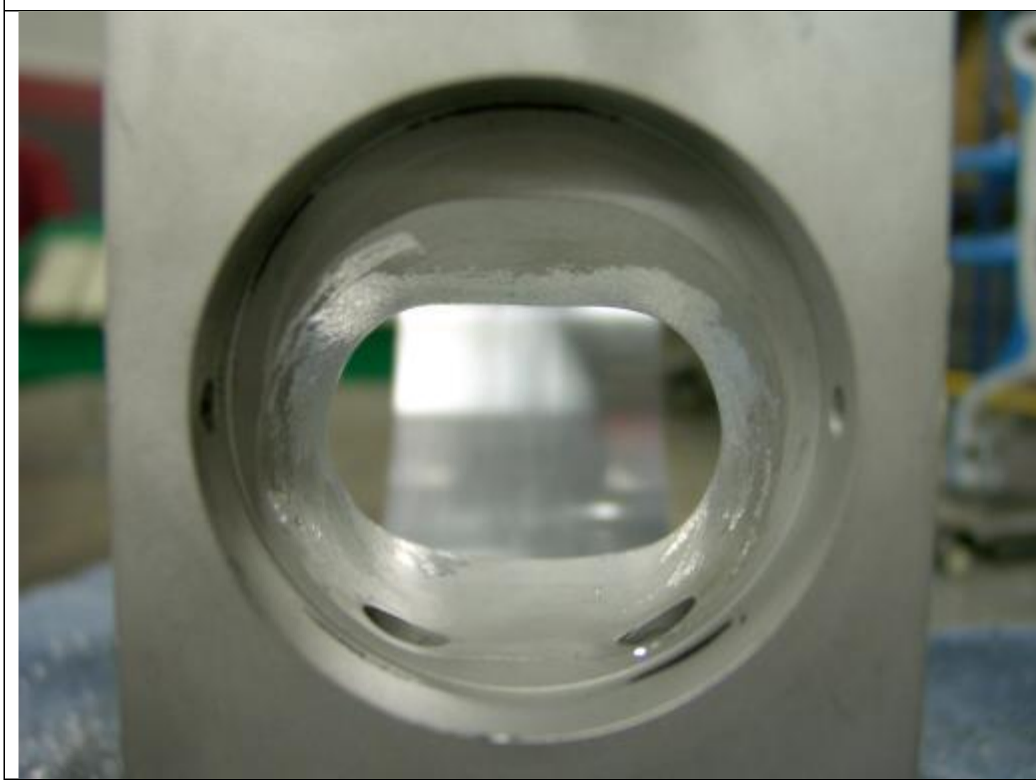
Just to make things more interesting, I had a rookie do the portwork. Having cut, and flowed countless exhaust ports myself, I normally get to the completed port without much testing, which wouldn't make for an interesting article. Nathan on the other hand did his first set of complete exhaust ports just a few weeks ago.

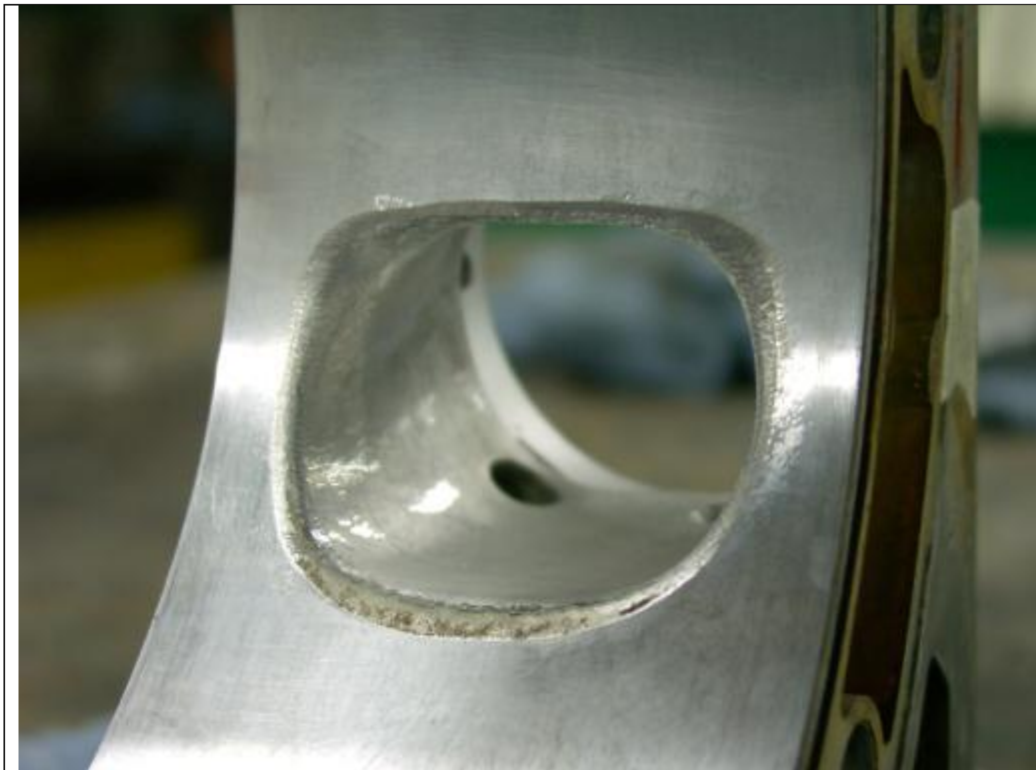
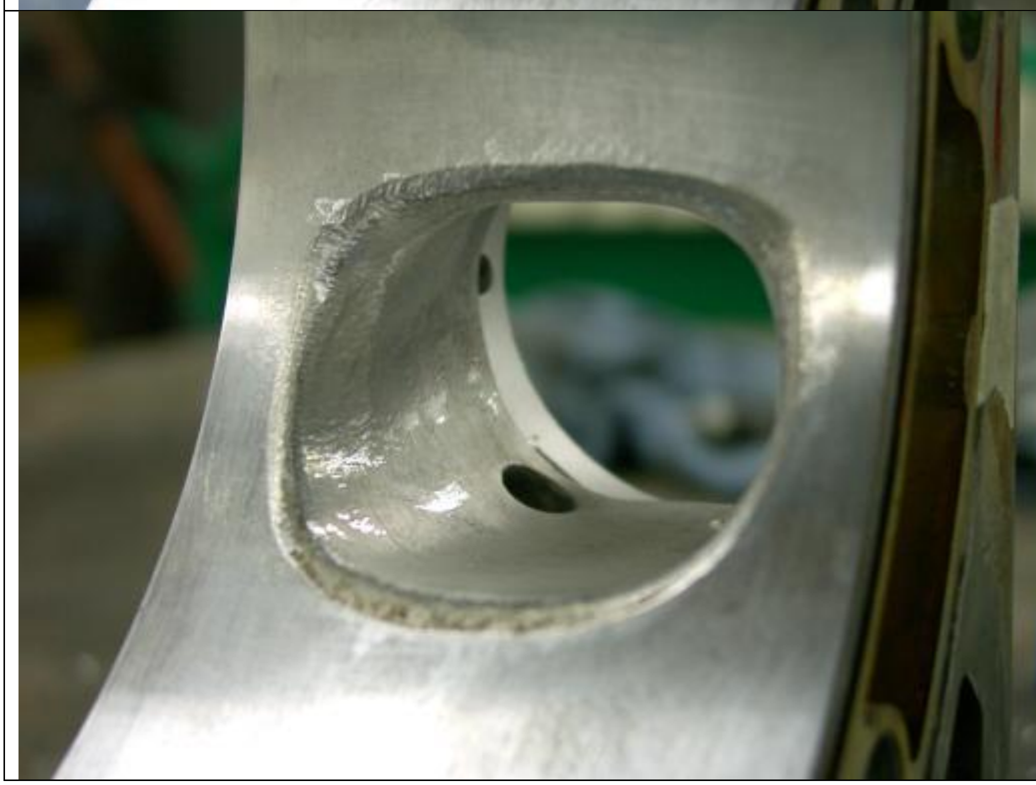
Note: Nathan has been working here for about 8 months while going to school. He currently has a bachelors in mechanical engineering, and is taking further schooling while working here building carburetors, machining and prepping engine parts, and double checking my math.

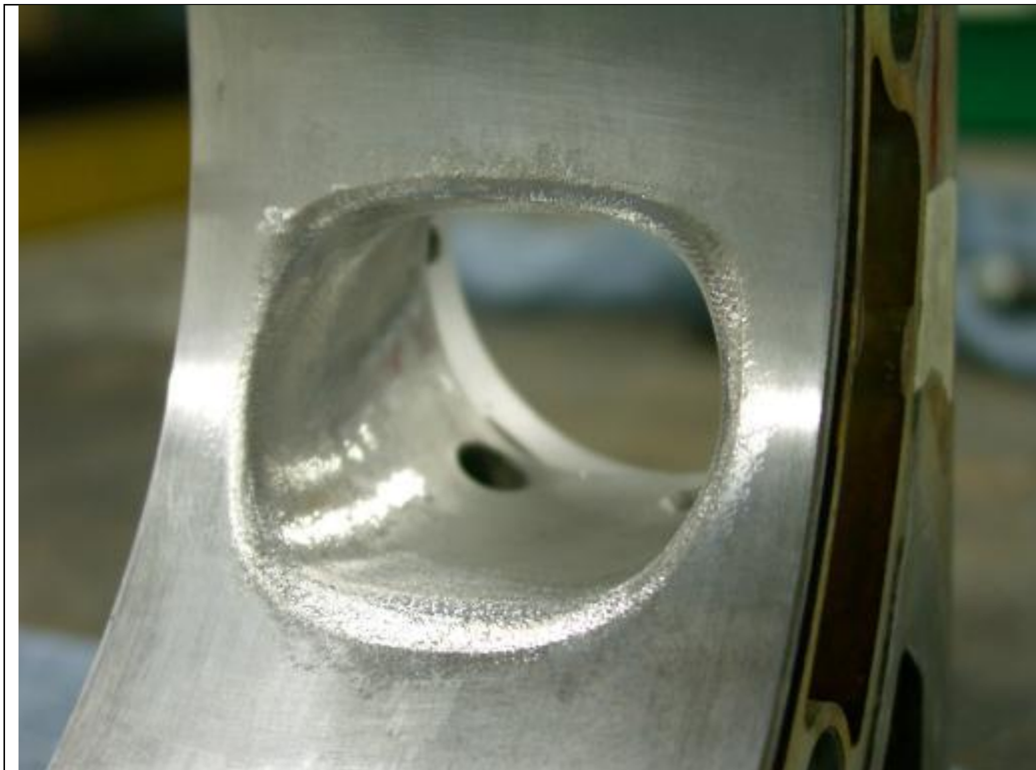
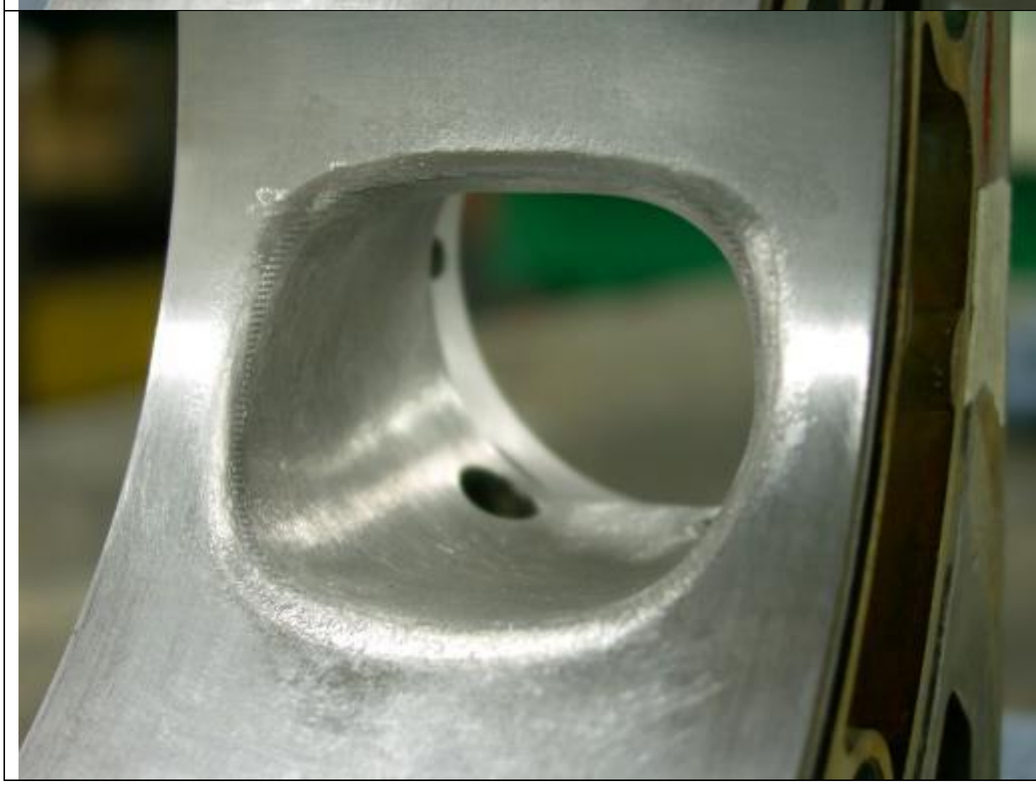
Not knowing exactly which areas require attention, he actually lost flow three times on the way to completing the port. This helps illustrate the importance of measuring the flow rather than guessing. The only guidance I gave was letting him know that the port roof would need to be widened, and to make sure that the inlet radius on the roof was not too large.

So, with very little experience, a digital camera, and a die grinder, I turned him loose.

	This shows the port before "tweaking" It looks a bit rough, but flows 230.6 CFM which isn't bad for a 12A exhaust port. This port was definetely cut by an experienced engine builder.
	This shows the port after widening the roof by "squaring" the corners. Going on looks alone, one would think that it would flow worse because of the square top, but in fact the flow jumped up to 249.6 CFM for an increase of 8.2% Take a good look at the two pictures. What you see is a very small increase in port size that resulted in a substantial increase in flow.

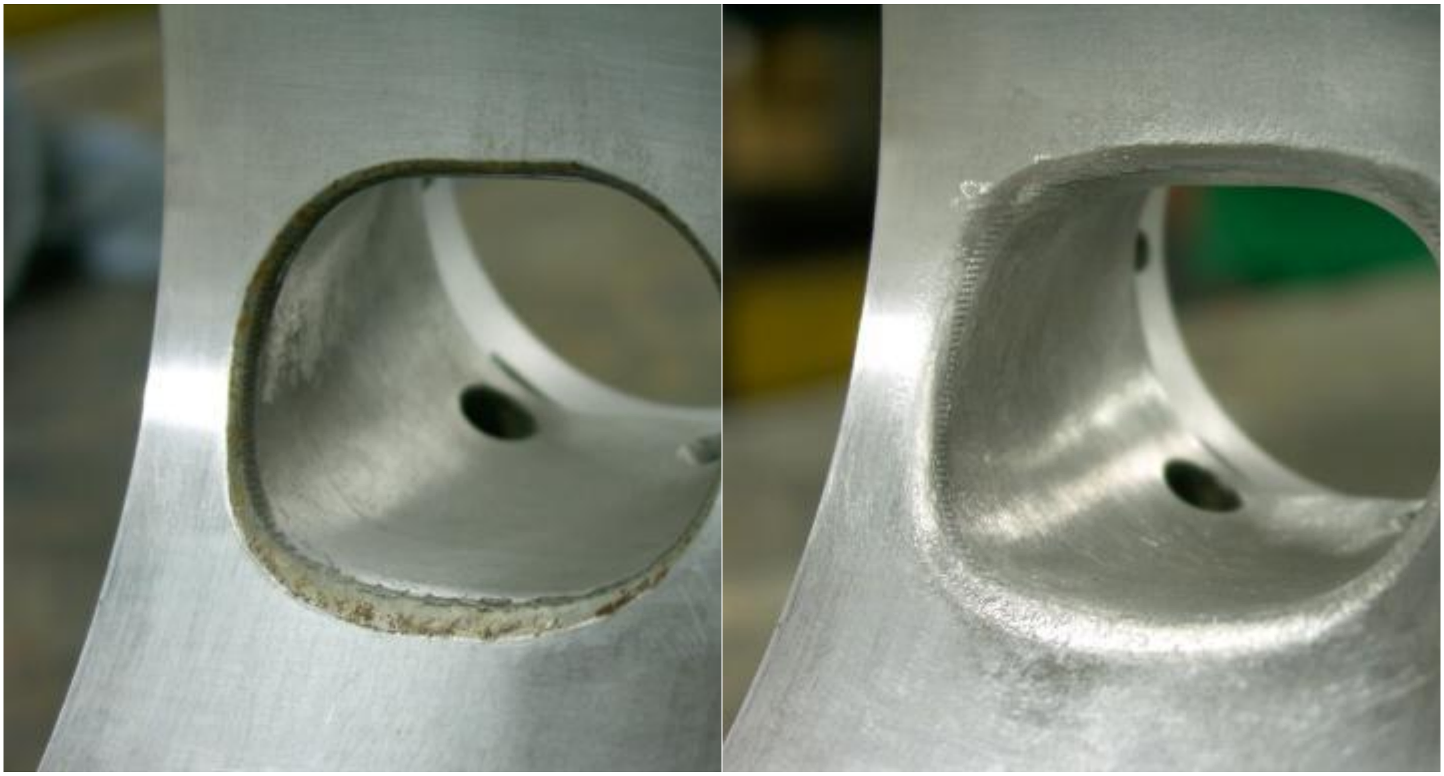
	This shows the port after further squaring and widening of the roof. (If a little is good, a lot must be better!) The widening was done only at the port window, and the flow dropped to 238.3 CFM which is a 4.7% decrease. Again, a very small change in port shape resulted in a substantial change in flow. From experience, I can see that the radii at the corners of the roof are too large, but going on looks alone, one would think that a large radius is helpful. The flowbench tells us otherwise!
	This shows the outlet side of the port after extending the port window shape into the runner. This reduced the corner radii, and brought the flow back up to the same 249.6 CFM that we had before.

	Knowing that having too large a radius on the roof of the port would hurt flow, Nathan removed some material from the flat portion of the roof, and guess what...We lost flow again!!! This time the flow dropped to 240.1 CFM. Having the flowbench to test the ports can be quite frustrating, **but** at least we know that we lost flow, and must do something to correct it. Note how slight the difference is between the two ports.
	This shows the port after laying the top back slightly to increase the radius on the roof. BINGO! The flow has now increased to 260.9 CFM which is an increase of 13.1% over our original port.

	At this point, the existing port window shape was maintained, but was smoothed up for another flow test. The flow increased by 1 CFM. Even though it looks much better, the increase was only 4 tenths of 1 percent.
	And finally, the arc of the port roof was flattened slightly, and everything was blended in. The end result is 263.8 CFM which is an increase of 14.4% over our original port. The port will now get a bit more clean up just for the sake of cosmetics, and it is ready to go.

Below is the original port, and the modified port shown side by side for comparison.

Note that only the top half of the port was changed. Any work done on the port floor would have shown little or no improvement, and substantial widening of the port floor would have hurt the flow badly.



So what's left? Just one simple question.

Do you want your engine builder to be measuring, or guessing?

Thanks for reading.

PY

Most Ridiculous Item of the Day

Have any of you ever watched Bill O'Rielly on the Fox news channel? You know, the guy with the huge ego, stating his opinion as fact to "save the country" from its own evils. Bill has a section of his daily show devoted ridiculous happenings in the world of politics.

If I were to do my own "Most Ridiculous" item in the world of racing it would be based on the following statement. "Horsepower sells motor cars, but torque wins motor races." This couldn't be further from the truth.

Like it or not, everything that goes on around us is governed by the laws of physics, and these laws are non-negotiable. The good news is that we don't have to be Einstein to apply the basic laws of physics to racing. The fact that too few do is the reason that such ridiculous statements are common in racing.

Let's start with a few definitions. Webster's dictionary describes torque as "A turning or twisting force." Note that the definition does not imply motion. As applied to an engine, it is simply a measure of the twisting force at the crank/eccentric shaft. Torque is normally rated in Lbs.-Ft. Since pounds feet doesn't exactly roll off the tongue, most of us refer to it as foot pounds.

Notice that there are two terms. Force (In lbs.) and distance (In ft.). At first it may seem strange to describe a "Turning or twisting force." in terms of distance, but a more detailed description makes it clear. If I were to put a shaft in a bench vice, attach a 1-ft. long lever to the end, perpendicular to the shaft, and then hang a one pound weight off the end of the lever, I would be applying one ft.-lb. to that shaft. Notice that the shaft is not rotating even though a torque is applied to the shaft.

If I were to replace the one-foot lever with a 100-foot lever, I would now be applying 100 ft.-lbs. to the shaft with the same one lb. weight. As you can see, the amount of twisting force on the shaft will vary depending on the length of the arm, and that requires that we specify a measure of distance to properly describe the force seen at the center of the shaft.

Let's say for instance that I pull the shaft from the vice, and ask you to hold it in your hand. If I do this with a one pound weight hanging from the end of the one foot lever, I will be applying a force of one ft.-lb. to your hand, and you will have no problem holding on to it. If I replace the lever with one that is 10 ft. long, with the same 1lb. weight on the end, (For all these scenarios, we assume that the lever itself is weightless.) You will now have a force of 10 ft.-lbs. applied to your hand, and it will be much harder to keep the shaft from rotating, even though you are still only resisting the one pound weight.

This would have exactly the same effect as setting a torque wrench to 10 ft.-lbs., attaching it to the end of the shaft, and applying force until the wrench clicks. Ten ft.-lbs. is ten ft.-lbs. whether it is applied with a one-foot lever and a ten-pound weight, or a ten-foot lever and a one-pound weight. Torque is equal to the weight, or force, times the length of the lever. It's that simple.

If a particular engine has a peak torque rating of 200 ft.-lbs., that force is equivalent to attaching a one foot lever to the shaft, and hanging a 200 lb. Weight from the end of it. Or…any other combination of weight and lever length which has the product 200.

Notice that I can take you from easily holding on to the one pound weight, to not even having a chance of holding it just by changing the length of the lever. (Like a one-lb. weight, and a 100-ft. lever.) Of course you say, that's just leverage! Well…you're right! Keep that in mind, because it is that leverage that makes all the difference, and a gear is in fact just a clever way to apply leverage between two or more rotating devices.

Let's say that the shaft used in our example is the input shaft of the transmission from a 1993 RX-7 with the following ratios.

1st 3.483 to 1

2nd 2.015 to 1

3rd 1.391 to 1

4th 1.0 to 1

5th .719 to 1

If the transmission is in 4th gear, one complete revolution of the input shaft will result in one complete revolution of the output shaft, just as if there were a solid shaft running all the way through. If we attach a 1-ft. lever to the input shaft with a 10-lb. weight on the end, the torque at the input shaft will equal 10 ft.-lbs. as we have already determined. Since we have a 1 to 1 ratio from input to output, we will also have 10 ft.-lbs. at the output shaft.

If we were to keep the same weight and lever on the input shaft, but switch the transmission to third gear, we would still have 10 ft.-lbs. at the input shaft, but we would now have 13.91 ft.-lbs. at the output shaft. This value is the product of the input torque and the gear ratio. (10-ft.-lbs. times 1.391 gear ratio equals 13.91 ft.-lbs.) If we were to switch the transmission into 1st gear, the result would be 34.83 ft.-lbs. at the output shaft.

As you can see, a gearbox gives us a simple way to vary the torque through leverage, and it is equivalent to changing the length of the lever. Thanks to gears, we can have any amount of torque that we want! In fact, a bone stock 12A making only 100 ft.-lbs. of torque could be geared to pull an 18-wheeler up a steep hill, as long as we are not in any big hurry to get the job done.

Let’s say that it takes 10,000 lbs of force to pull a heavy weight up a hill. No problem! We could even do it with our stock 1980-GS in 4th gear if we are willing to build a custom ring and pinion gear with a ratio of 100 to 1. (100-ft.-lbs. times transmission gear ratio of 1:1 times ring and pinion gear ratio of 100:1 equals 10,000 ft.-lbs.)

If we are using a tire with a diameter of 24", the distance from the axle center to the ground is exactly one foot, and so the force is equal to 10,000 lbs. Remember, the torque is equal to the lever length times the force. If we re-write that formula to solve for force, force is equal to torque divided by the lever length, and so that 10,000 ft.-lbs. at the rear axle results in 10,000 lbs. of force at the tire contact patch.

With this same information, we can also calculate the acceleration rate of the vehicle, but first we need to consider Newton’s second law of motion, which states that "Acceleration is proportional to force." and "Acceleration is inversely proportional to Mass." This law is normally stated more simply as "Force equals mass times acceleration." or F=MA. If we rewrite this to solve for acceleration, we get A=F/M. To find the rate of acceleration for a vehicle, we simply divide the force (In lbs. at the tire contact patch.) by the mass (Total weight of the vehicle in lbs.)

Let’s calculate the acceleration rate of a 1st. gen. RX-7. The engine has a torque peak of 100-ft.-lbs. In fourth gear, the ratio is 1 to 1, and so the torque at the output shaft is also 100-ft.-lbs. The ring and pinion ratio is 3.909 to 1, and so the torque at the rear axle will be (100 times 3.909) 390.9-ft.-lbs. The tire diameter is 24 inches, and so the lever length (Distance from the **center of the axle** to the ground.) is 12 inches, or one foot. The resulting force at the tire contact patch will be (390.9-ft.-lbs. of torque divided by lever length of one foot.) 390.9-lbs. of force. The total vehicle weight with a driver is 2600 lbs., and so the acceleration rate in G’s (The force of gravity.) will be force (390.9) divided by mass (2600) which equals .15 G’s.

If we do the same calculations for first gear acceleration, we find that the force at the contact patch is 1,436 lbs., and the acceleration rate is .55 G’s. It’s clear that we have used the gears for leverage, with the result being a greater rate of acceleration in 1st gear. Of course you knew that already, but now you know why.

By now it should be clear that the acceleration rate of a vehicle is determined by the weight, and the force at the contact patch, which is the result of the torque output of the engine, and all the levers/gears between it and the ground.

You’re probably thinking that we have just determined the acceleration rate of the vehicle, and even changed it with gearing, with no mention of horsepower. So torque really is the determining factor right? Wrong! We haven’t considered speed.

We can gain acceleration by changing the gear ratios, but we can’t go very fast in first gear, so what’s the point? We have effectively changed the amount of torque available to accelerate the vehicle, but our top speed is limited to about 25 mph.

Confused yet?

Read on.

OK, so we all know what torque is, now let’s get to horsepower.

Referring once again to Webster’s dictionary, horsepower is defined as "A foot/pound/second unit of power, equivalent to 550 foot/pounds per second."

Put more simply, horsepower is a measure of work done over time, or the rate at which work is done.

So now we have another term to confuse things. As if force and distance weren’t enough, we now have time involved, and the shaft must actually be spinning. Why… Well, if you are just standing there holding on to a shaft with a lever and a weight, you are doing no work. If you stand there long enough, you will feel like you are working, but in fact you are doing no such thing. Don’t believe me? Clamp the shaft back in the vice, and you can leave it there indefinitely without having to feed it, add gas, and any other means of supplying it with energy.

Torque, all by itself does nothing useful. In fact, the definition of torque does not even require that the shaft be moving. I am sure that all of you want your car to do something useful, like take you to the movies, or get you around the track before the other guy. In other words, you need your car to do some work, and you want it to do that work in the least amount of time possible.

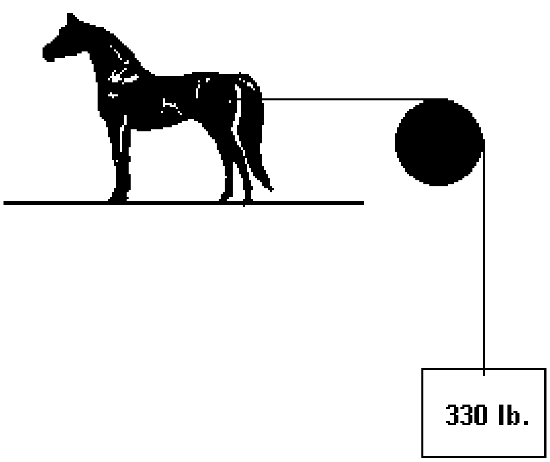
If I give you a wagon full of cement blocks, and ask you to pull it one mile up a hill, you would agree that I am asking you to do work. If I ask you and your buddy to each pull a wagon full of cement blocks up the hill, you will both be doing the same amount of work. But…If it takes you an hour, and your buddy does it in 30 minutes we have a very different situation. You have both done the same amount of work, but your buddy, by completing the task in half the time, has proven that he can develop twice the horsepower that you can. You both traveled the same distance, and exerted the same amount of force, but the third term in the definition of horsepower, time, was different.

For those of you who are sticklers for details, the force required to pull the wagon up the hill at a steady speed is equal to the weight of the wagon, times the sine of the angle of the hill. If the wagon weighed 100-lbs., and the hill was at a 45-degree angle, the required force would be (Sine 45 Times 100 lbs.) equals 70.7 lbs.

If we were interested in moving the wagon by driving the wheels rather than pulling it by the handle, we could convert the force to torque by dividing the required force by the radius of the driven wheels. Let's say that we have 6" diameter wheels. That would give a lever length (Distance from the center of the axle to the ground.) of 3 inches, or .25 feet. The required torque would then be (70.7 lbs. times .25 ft.) which equals 17.675 ft.-lbs.

James Watt, who spent the majority of his life perfecting the steam engine, created the term horsepower. He was looking for a way to measure the rate of work done by a horse so that he could make valid comparisons between horses which did most of the work in those days, and his steam engines which he hoped would do most of the work in the future.

Watt found that on average, a horse could lift 330-lbs of coal 100-ft in one minute. He then stated that the power available from one horse was equal to (330-lbs. times 100-ft.) or 33,000-lbs./ft./min. If you divide that by 60 to convert to lbs./ft./sec. you get 550-lbs./ft./sec. Watt called this one horsepower, which leaves most of us wondering why he didn't call it one watt. I don't have the answer to that, but I do know that 746.6 watts equals one horsepower. If you ever see an engine rated in watts, (This is still popular in some countries.) you can divide by 746.6 to determine the horsepower. Or, you can tell you pals that your bone stock 3rd. gen. RX-7 puts out One hundred ninety thousand, three hundred and eighty three watts.



So if one horsepower is equal to 33,000 lbs.-ft. per minute, we can rearrange that to say that horsepower equals torque times rpm, divided by 5252. How do we get there?

In the above formula, force and distance are stated in ft.-lbs., and time is stated in RPM, so we need to convert our terms. First we need to express that 33,000 lbs. of force as 33,000 ft-lbs. As you now know, that is equivalent to a 33,000-lb. weight hanging from a 1-ft. lever. Then we need to express the one-foot per minute as RPM.

The circumference of a circle is defined as the diameter of the circle times Pi, which is 3.14159. We have a one-foot lever, so if we were to spin the shaft, the outer edge of the lever would scribe a 2-foot diameter circle. The circumference of a 2 foot circle is (3.14159 times 2) 6.282 feet. If we divide 1 foot by the distance traveled in a complete revolution (1 divided by 6.282) we get .159 revolutions per minute, which is equal to one foot per minute.

So now we have: One horsepower equals 33,000 lbs.-ft. of torque per .159 RPM.

That's still kind of ugly dealing with just a fraction of an rpm, so we divide both terms by .159 and we get: one horsepower equals 5252 lbs.-ft. of torque per 1 rpm.

This can be rewritten a few different ways that are valuable to us.

Horsepower equals torque times rpm divided by 5252. Horsepower = (Torque X RPM) / 5252

Torque equals horsepower times 5252 divided by rpm. Torque = (Horsepower X 5252) / RPM

RPM equals horsepower times 5252 divided by torque. RPM = (Horsepower X 5252) / Torque

If you know any two of the terms, you can calculate the third. You might also notice that torque and horsepower will always be equal at 5,252 rpm, horsepower will be greater than torque above 5252 RPM, and torque will be greater than horsepower below 5252 RPM. ALWAYS…NO EXCEPTION! Just look at any dyno sheet, and you will see what I mean. If you see a dyno sheet where this is not true, you can be sure that someone fudged the numbers to help sell a product.

Back to the issue at hand, I'm sure that the coal tugging horse and a wagon full of cement blocks probably doesn't seem all that relevant to your racecar. So let's look at things another way.

The definition of horsepower includes three terms. Force, distance, AND time, where torque is simply a force applied over a distance. In the case of Watt's experiment, the force was exerted by the weight of the coal, which was being lifted from the mine. In a car, we are interested in acceleration, not the ability to lift an object. In our case, the force is exerted by the inertia of the vehicle, which resists acceleration.

So now I need to bore you with another definition. Back to Webster's dictionary, inertia is defined as "A property of matter that causes it to resist changes in velocity." In more simple terms, your car would rather not be accelerated from 30 to 70 mph, and so an external force is required to make this happen. This force comes from your engine.

To accurately describe the acceleration capability of your vehicle, we must consider time. If we just considered force, and distance, we wouldn't really be saying much about the car. If I tell you that my car can pull a 3,000-lb. weight 100-ft. up a hill, would you be impressed? Certainly not, because I haven't really told you much. If I told you that I could do it in 10 seconds, while your car needed 15 seconds to do the same job, you might be impressed.

After all, what we are really interested in is the ability to cover distance in a period of time. The distance from the exit of one corner to the entry of the next, or the quarter mile, or maybe even the distance from one stoplight to the next.

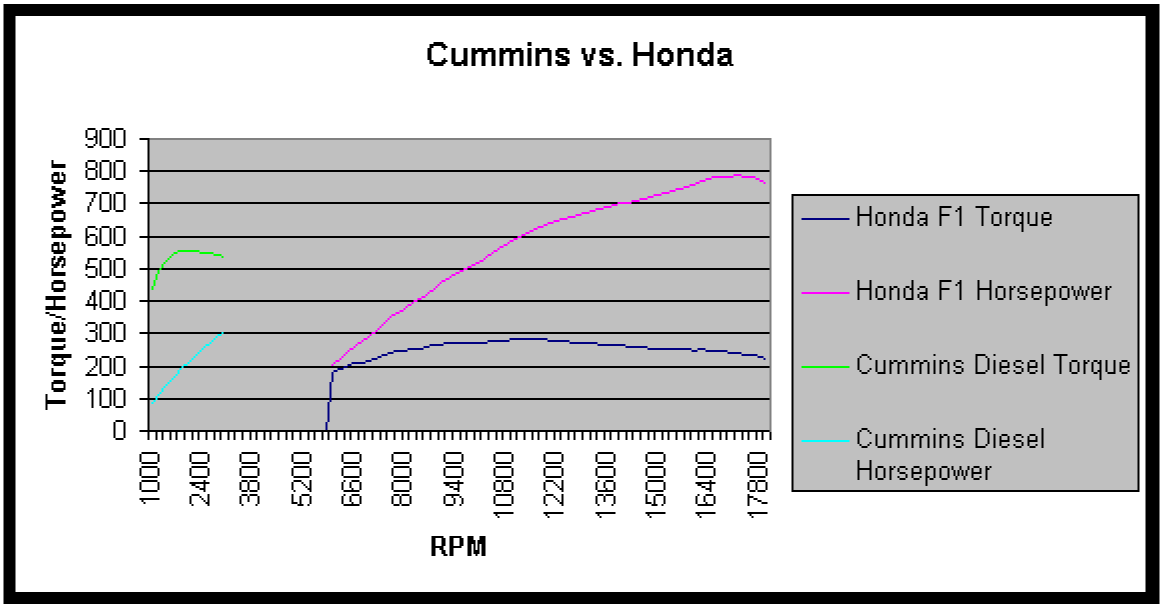
If we consider the rate of acceleration, AND miles per hour, we have all three terms included in the definition of horsepower. Time, distance, and force. Force is the rate of acceleration, or the force of inertia. Time is in hours, and distance is in miles.

So now instead of just considering the rate of acceleration arbitrarily, let's include miles per hour.

And while we're at it, let's consider the acceleration rate of two very different motors to illustrate the importance of horsepower, and the absolute irrelevance of torque.

At one extreme we have a Honda F1 motor which revs to 18,000 rpm, makes nearly 800 horsepower, but a measly 281 ft.-lbs. of torque. At the other end of the spectrum we have the Cummins turbo diesel available in the 2003 Dodge Ram which makes a whopping 555 ft.-lbs. of torque, but only 305 horsepower. So which one do you think will accelerate faster?

Everyone that you ask will answer that the Honda F1 engine will accelerate faster. Even your neighbor with the big block who claims that torque is the key to going fast. If torque were the determining factor, the Cummins diesel would win hands down. So what gives?



Let's calculate the acceleration rate for both engines in a hypothetical 2500 lb. car using the transmission from the 1995 RX-7 with a two-foot diameter tire. Since we know that an F1 car will go 200 MPH, we will gear the car for that speed with both motors.

Starting with the F1 engine which redlines at 18,000 rpm, we need to calculate the required ring and pinion ratio to achieve 200 mph at redline in 5th gear.

First we convert miles per hour to miles per minute by dividing by 60.

200/60=3.333 miles per minute

The tire diameter is rated in feet, so we must convert this 3.333 miles per minute into feet per minute. There are 5280 feet in a mile, so:

3.333 X 5280 = 17,600 feet per minute

If our tire is 2 feet in diameter, the circumference is 2 feet times Pi

2 X 3.14159 = 6.28318 feet per revolution.

Now we divide the feet per minute, by the feet per revolution and we get:

17,600 / 6.28318 = 2801.13 tire revolutions per minute to achieve 200 MPH.

The engine redlines at 18,000 RPM, and in 5th gear the transmission ratio is .719 to 1. To determine the RPM of the output shaft at redline, we take the engine RPM divided by the gear ratio to get:

18,000 / .719 = 25,034 RPM

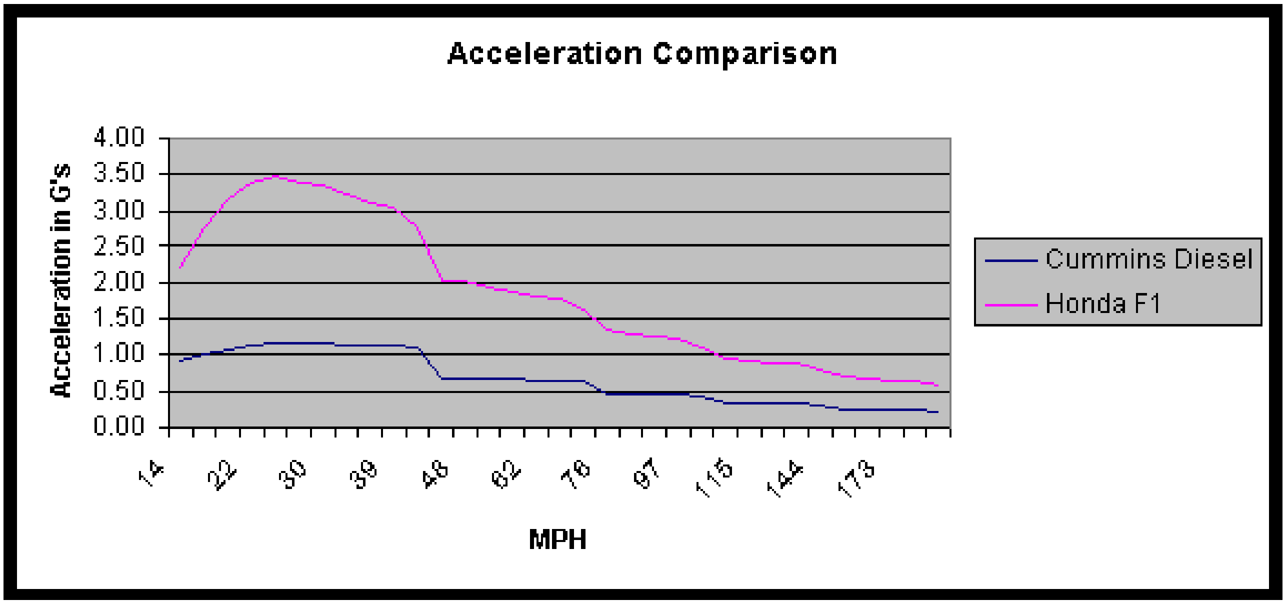
The output shaft is spinning 25,034 RPM, and we need the wheel to spin 2801.13 RPM to go 200 MPH. To find the correct ring and pinion ratio, we divide the output shaft RPM by the required tire RPM and we get:

25,034 / 2801.13 = 8.937 to 1

Going through all the same boring math for the Cummins diesel which is only spinning 3000 RPM at redline, we get a required ring and pinion gear of 1.489 to 1 to go 200 MPH at redline in 5th gear.

(Note that the F1 engine is spinning 6 times faster than the Cummins, and so the required ring and pinion ratio is exactly 6 times higher.)

With the transmission in first gear, both vehicles will be traveling at 13.76 miles per hour at the bottom of their powerband. (1,000 RPM for the Cummins, and 6,000 rpm for the Honda.) The following chart shows the acceleration rate of both engines in our hypothetical vehicle from that point to 200 MPH.



Note that at any point on the chart, the percent difference in the rate of acceleration is EXACTLY the difference in horsepower. For instance, at 200 mph, the Honda F1 engine is accelerating at a rate of .572 G's, while the Cummins diesel is accelerating at a rate of .228 G's. If we divide .228 into .572 we get 2.5, and so the acceleration rate of the Honda is 2.5 times greater than that of the Cummins.

The Cummins, at 3,000 rpm is making 305 horsepower, while the Honda is making 763 horsepower. The Honda is making 2.5 times the power of the Cummins, which is exactly the difference in the rate of acceleration. You can work this out at **any** point on the chart, and you will find that this direct relationship between horsepower and rate of acceleration always holds true.

I'm sure there is someone out there that still thinks I'm off my rocker, but as I stated earlier, the laws of physics are non-negotiable. After showing this article to a few people for proofreading, one person stated that the results aren't valid because torque motors are for low rpm grunt, and aren't meant to run at 200 MPH. Ignoring the fact that this is a ridiculous statement, let's consider what would happen if we geared both combinations for a top speed of 100 MPH. It should occur to you that this would be a simple matter of doubling the ring and pinion ratio, and that would be correct.

The end result is that the acceleration figures would simply double across the board for both combinations. The difference in acceleration would still match the difference in horsepower, and ultimately the difference in performance would be the same.

So there it is. Horsepower is the determining factor in the rate of acceleration of any vehicle. The next article will go into more detail, and show you how these simple calculations can be used to choose appropriate gearing for any track.

Below are some useful definitions and formulas.

TORQUE IN LBS./FT. = (WEIGHT IN LBS. X LEVER ARM LENGTH IN FEET.)

1 HP = 550 LBS./FT./SEC.

1 HP = 33,000 LBS./FT./MIN.

HP = (TORQUE X RPM) / 5252

TORQUE = (HP X 5252) / RPM

TORQUE AT THE REAR WHEELS = (ENGINE TORQUE X TRANSMISSION GEAR RATIO X RING AND PINION RATIO)

ACCELERATIVE THRUST = (TORQUE AT THE REAR WHEELS / TIRE RADIUS IN FT.)

RATE OF ACCELERATION = (ACCELERATIVE THRUST / TOTAL VEHICLE WEIGHT.)