- Exhaust System Technology-

# The Sound and The Fury

THIS IS A MODIFIED VERSION OF AN ARTICLE
BY Jack Kane WHICH APPEARED IN ISSUE 036 of
RACE ENGINE TECHNOLOGY MAGAZINE

All too often the engine exhaust is an afterthought for the engine and chassis builders, yet its design and construction impacts significantly upon car performance. The exhaust system can be a vital tool for optimizing the performance of the engine, through the way in which its design manipulates the pressure waves that can crucially assist cylinder filling and scavenging. On the other side of the coin, the exhaust system presents many challenges. It is a major loss-path for thermal energy; and it can be a car packaging nightmare.

The environment which a competition exhaust system, and particularly engine headers, must survive, can only be described as a brutal combination of temperatures, stresses, corrosion and vibration. Contemporary exhaust technology can help reduce the problems and help to maximize the potential gains of the system.

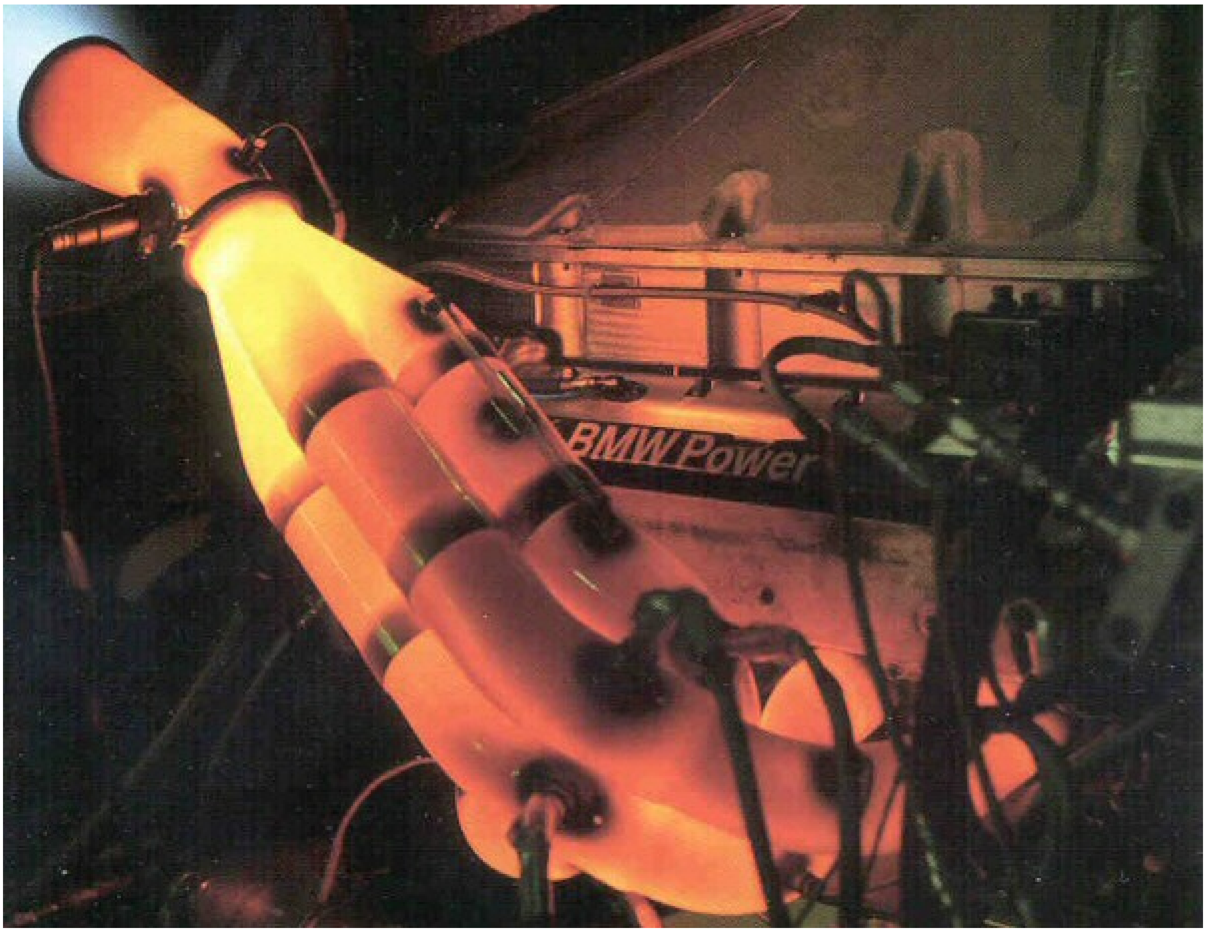

Figure 1
BMW Formula-One Engine at Full Power

It is interesting, from having spoken to several highly-placed and well recognized experts in this field, that while there is general agreement about what features cause improvements to happen, there are varying opinions about the reasons why those improvements occur.

## Basics

The computation of what actually goes on during an exhaust cycle is a highly complex problem in compressible fluid flow, the details of which are explained in detail in several texts, my favorite being Professor Gordon Blair's Design and Simulation of Four Stroke Engines. For the purposes of this article, the following overly-simplified explanation will serve to illustrate the principles.

There are two separate components to the exhaust event. The first is the removal of exhaust gasses from the cylinder, which occurs as a pulse of hot gas exiting the cylinder and flowing down the header primary tube. The second is the (much faster) travel of the pressure wave in the port caused by the pressure spike which occurs when the exhaust valve opens, and the various reflections of that wave. Taking proper advantage of these pressure waves (component two) can produce dramatic improvements in clearing the cylinder (component one) and can strongly assist the inflow of fresh charge.

Considering component one, when the exhaust valve first opens in a 4-stroke piston engine, the in-cylinder pressure is still well above atmospheric. In a normally-aspirated spark ignition engine burning gasoline and operating at high BMEP, the pressure can be 7 bar or more, and the pressure in the exhaust port at the valve is somewhere near 1 bar (atmospheric). As the valve opens, the pressure differential across the rapidly-changing valve aperture (pressure ratio of approximately 7) starts exhaust gas flowing through the opening, and the outrush causes the pressure in the port (behind the valve) to increase rapidly, or "spike".

The instantaneous velocity of the exhaust gas flow at any point is determined by the pressure gradient and the cross-sectional area at that point. In the header, a smaller tube diameter will increase the velocity at a given RPM, which might enhance the pressure wave tuning (the second component) and can be beneficial with regard to inertia effects. However, if the diameter is too small, there will be flow losses and consequent pressure gradient increases which can offset any tuning gains. So the selection of proper tubing diameters is an important part of the design.

In the early part of the exhaust cycle, the pressure difference across the valve is high, so the instantaneous gas particle velocity through the small exhaust valve aperture is very high. Sometime past mid-exhaust stroke, the majority of the exhaust gas has left the cylinder. At that time, the valve aperture area is quite large and the cylinder pressure is approaching atmospheric, which causes the instantaneous particle velocity across the valve to be much lower. It is at that phase of the exhaust cycle where the second component becomes important.

To help with the explanation of the second component, **Figure Two** shows traces of in-cylinder pressure (black), port pressure at the intake valve (light blue) and port pressure at the exhaust valve (red), taken from a simulation of a high BMEP engine operating near the optimum tuning point for both intake and exhaust.

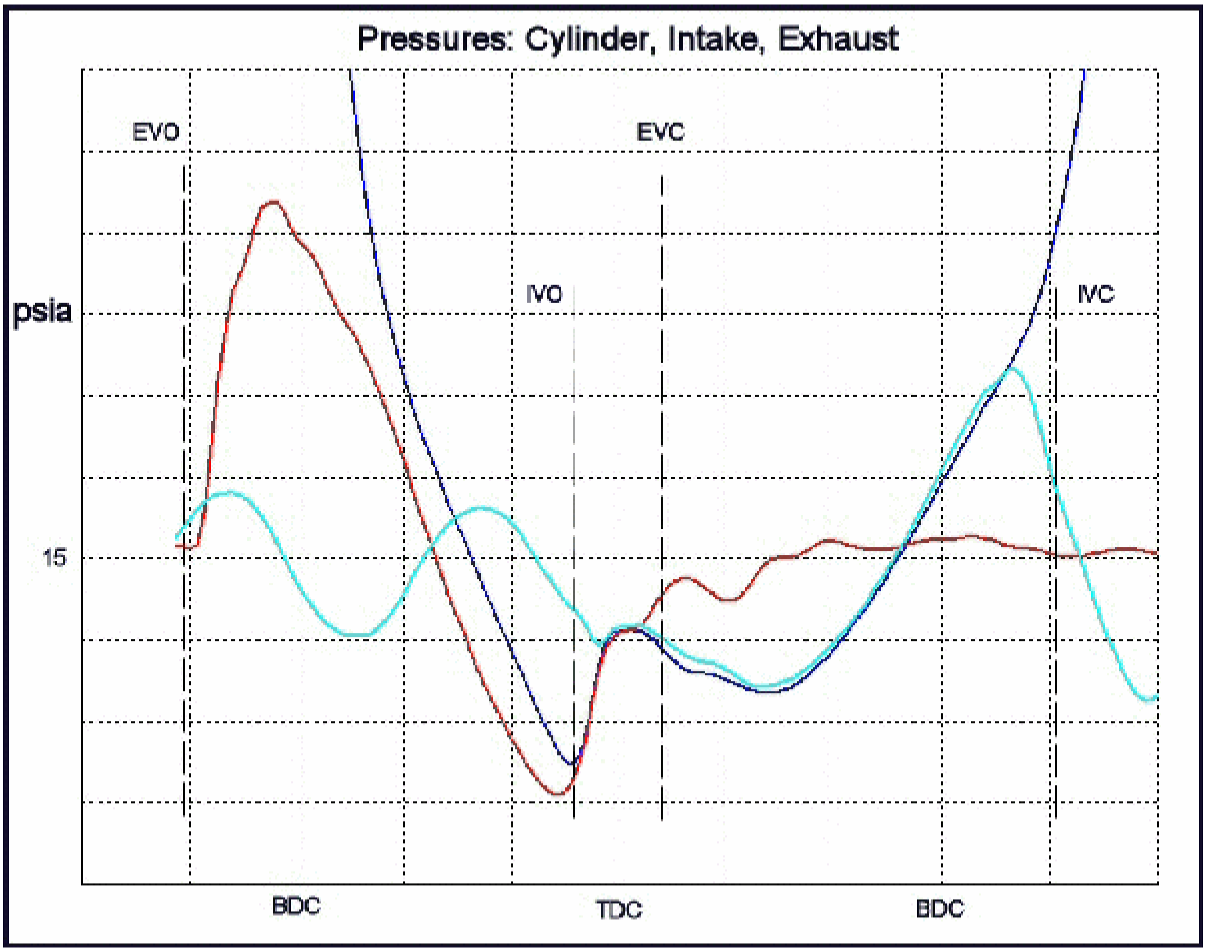


Figure 2
Intake Port, Exhaust Port and In-cylinder Pressures with Effective Tuning

The second component is the result of the pressure "spike" which occurs at EVO, shown by the peak in the red line in **Figure Two**, just after EVO. That pressure spike, or pressure wave, moves down the pipe at the sum of the local sonic velocity plus the particle velocity of the gas flow. Whenever the pressure wave encounters a change in cross-sectional area of the pipe, a reflected pressure wave is generated, which travels in the opposite direction. If the change in area is increasing (a step, collector, the atmosphere), the sense of the reflected pressure wave (compression or expansion) is inverted. If the change in area is decreasing (the end of another port having a closed valve, or a turbocharger nozzle, for example), the sense of the reflected wave is not inverted. The amplitude of the reflected wave is primarily determined by the proportionate change in cross-sectional area (area ratio), but the amplitude is diminished in any case. For purposes of approximation, the particle velocity can be ignored because its effect is self-canceling during the round-trip of the wave. However, highly-accurate simulations must take it into account. These waves are sometimes called finite difference waves, because of the finite difference numerical modeling techniques used to calculate their propagation characteristics.

In the case of the currently-flowing header primary, the EVO-initiated positive pressure (compression) wave is reflected back as a negative pressure (expansion) wave. If the arrival of the reflected negative pressure wave back at the exhaust valve can be arranged to occur during the latter part of the exhaust cycle, the resulting lower pressure in the port will enhance the removal of exhaust gas from the cylinder, and will reduce the pressure in the cylinder so that when the intake valve opens, the low pressure in the cylinder begins moving fresh charge into the cylinder while the piston is slowing to a stop at TDC.

Note in **Figure Two**, how the cylinder pressure (black) and exhaust port (red) pressures go strongly negative from approximately mid-exhaust stroke to TDC). Note also how the second-order reflected positive pressure wave in the intake tract (light blue) reaches the back of the intake valve just before IVO, and works together with properly-timed exhaust negative pressures to begin moving fresh charge into the cylinder.

If, on the other hand, the negative exhaust pressure wave arrives a non-optimal time, its effects can be detrimental to the clearing of the cylinder and ingestion of fresh charge. A reflected positive wave during overlap (from a turbocharger nozzle, for example) can push a large amount of exhaust gas back into the cylinder and the intake system.

**Figure Three** shows the same three pressure traces when the engine is operating well above the intake and exhaust tuning points. In addition to reduced breathing efficiency, note the additional pumping losses from the higher cylinder pressure in the latter portion of the exhaust cycle, caused in part by the late arrival of the reflected negative exhaust pulse.

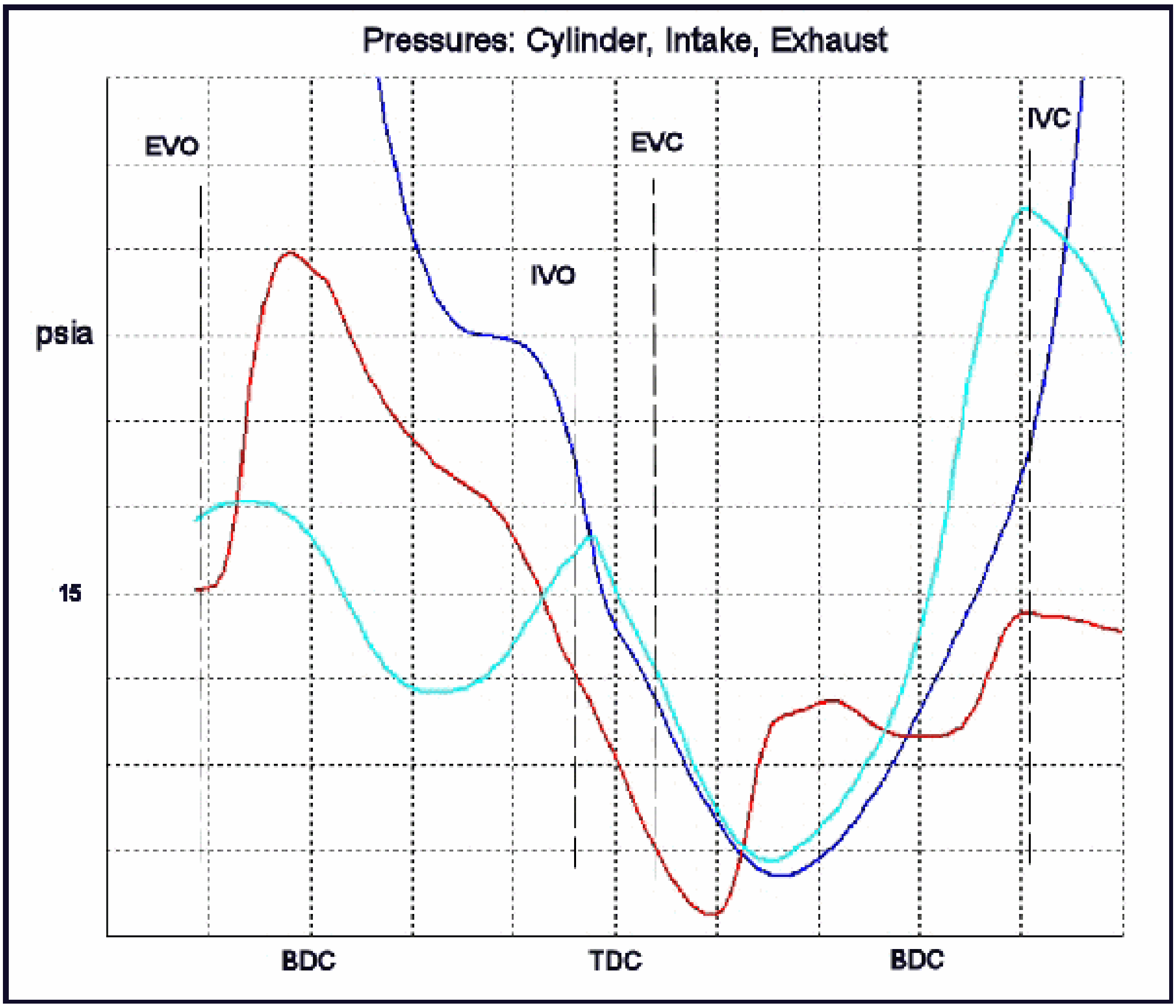


Figure 3
Intake Port, Exhaust Port and In-cylinder Pressures with Poor Tuning

The timing of the arrival of the negative wave at the back (port) side of the exhaust valve is determined by the engine RPM, the speed of sound in the pipe and the distance from the valve to the relevant change in area. Those three factors will cause the exhaust tuning to come in and out of tune over the engine operating speed range. Sophisticated designs can produce systems having more than one tuning point. The most significant example of exhaust pulse tuning is dramatically demonstrated by the operation of crankcase-scavenged, piston-ported two-stroke engines.

At the relevant tuning distance from the exhaust valves, the primary tubes from two or more cylinders are often joined together into a larger collector tube which provides the area increase to generate the reflected waves described above.

Using a 4-into-1 system as an example, the four primary tubes will ideally have the same centerline length and will sharply transition into an area having roughly three to four times the area of the primary.. The larger the cross-sectional area of the collector tube plus the area of all other tubes at the same junction compared to the area of the active primary tube (area ratio), the larger will be the amplitude of the reflected wave. However, the collector has an optimal size: too much area and the wave tuning in the collector will be diminished. The optimal length is related to the number of cylinders feeding into it.

The effect of a straight collector is generally a very peaky tuning, in which the lengths of the primaries can be varied to produce a 'rocking' effect of the torque curve around its peak. Lengthening the tubes raises the portion of the curve below peak and reduces the portion above peak torque; shortening them has the reverse effect. Various strategies have been devised to spread the effect of exhaust tune over a wider RPM band. These strategies typically involve generating additional waves of smaller amplitude (additional, smaller steps, for example) or attempts to increase the width (duration) of the pulse at the expense of pulse amplitude by using a tapered section to extend the area change over a longer period of time.

**Figure Four** shows one of these devices, known in the States as a 'merge collector'. The primaries converge into a nozzle area which is larger than the primary area but smaller than the final collector size. That keeps the gas velocity up for a bit longer, helping to scavenge neighboring pipes, and the smaller area ratio reduces the amplitude of the reflected wave. The section behind the nozzle tapers up to the final collector diameter, allowing the flow to decelerate with better pressure recovery than would occur with a sharp transition, and extends the width of the reflected wave. The characteristics of the reflected wave can be tuned with different nozzle areas, different final collector diameter and length, and the length of the tapered section. The net effect is usually aimed at boosting a particular portion of the torque curve and at extending the RPM band over which that boost is effective.

Figure 4
A "Merge Collector"

It is sometimes argued that the speed of sound is a function of pressure, density, temperature, and / or phase of the moon. Actually, the speed of sound in an ideal gas (which air emulates) is a function of the stiffness of the gas divided by the density. When one does the arithmetic necessary to create an equation which uses known parameters, the stiffness and density terms are replaced by equivalents from the ideal gas law, producing the equation: Va (acoustic velocity in meters per second) = square root ( S x R x T), where S is the ratio of specific heats (approximately 1.4 for air at 25°C, 1.35 for exhaust gas at 500°K), R is the gas constant (approximately 287 J/kg-°K for air, 291 for exhaust gas) and T is the absolute temperature (°Kelvin, which is °C + 273).

What that boils down to is that once one has the specific heats and gas constant value for a given gas (or mixture of gasses), the speed of sound varies only with the temperature. To add a bit of complexity, the instantaneous temperature of the exhaust gas varies along the exhaust path, perhaps as much as 150°C in a primary tube.

The next interesting basic is that as the pressure ratio increases across a smoothly-decreasing nozzle, the particle velocity at the smallest cross-sectional area increases with increasing pressure ratio until it reaches the local speed of sound. Once it has reached the speed of sound, no matter how much larger the pressure ratio becomes, the gas particle velocity remains at sonic ("choked"). An increase in the upstream pressure will increase the mass flow rate due to the increased density upstream of the nozzle, but the particle velocity through the nozzle remains sonic.

For air flowing in a smoothly-decreasing nozzle, the pressure ratio which just causes sonic flow (the 'critical pressure ratio') is slightly less than 2.0. For non-smooth and irregular nozzles (an exhaust valve, for instance) the critical pressure ratio is higher, but the effect is the same. That means that, for some period of time after EVO, the gas particle flow velocity across the exhaust valve is at the local speed of sound, which as shown later, is quite high at exhaust gas temperatures.

Again, it should be noted that these explanations are highly simplified. There are several very-high-end engine simulation software packages which are said to model engine performance, including exhaust system phenomena, quite accurately. These models are so sophisticated that they can take into account such esoterica as the local temperature gradients along the primary, secondary and collector tubes. For accuracy, these models rely on accurate engine data, including valve flow coefficients at various lifts. Apparently it is difficult to determine accurate flow coefficient data for the valves, particularly at high pressure ratios, which has a profound influence on the limits of computational accuracy.

That being said, several designers told me that the simulations tend to be less accurate in predicting the various effects of the collector, in terms of the real world effects of geometry, pipe angles, and the like. One approach to that problem has been to use a CFD simulation (a 3-D analysis) for the collectors, and couple those results with the 1-D simulations of the pipes.

## Exhaust Materials

Usually, header systems are fabricated from welded-up collections of cuts from pre-formed "U" bends and straight segments of tubing in the chosen material. There are several reasons for that, but the most persuasive is the fact that, in order to achieve the design configuration, there is not usually ample grip-space between bends to form the pipes from a single piece of tube. In some cases, where the bends are not too closely spaced, the pipes can be bent up in one piece using a mandrel bender which will retain the circular cross section of the tube throughout the bend and transition. The typical exhaust tube bender commonly found in automotive exhaust shops is not suitable for that duty since those benders distort the cross-section of the bends terribly and shrink the cross-sectional area.

Tubing bend radii (the radius of the plan-view centerline of the bend) are expressed in terms of multiples of the tubing diameter. For example, a "1.5-D bend" in 2-inch diameter tubing would have a bend radius of 3 inches. One fabricator described some specialized machinery he had devised for making high-quality exhaust tubing from sheet. The first machine rolls the sheets into straight tube sections of the required diameter. The second machine completes the straight section of tube with a continuous welded seam using a semiautomatic inert-gas-shielded process. A third machine does what had been thought to be impossible: bending 0.50-mm wall inconel tubes into less-than-1-D radius sections while retaining accurate cross-sectional geometry.

There are several materials commonly used in competition header and exhaust systems, depending on the requirements and operating temperatures.

For the most demanding applications, Inconel tubing is commonly used. Although the name "Inconel" is a registered trademark of Special Metals Corp., the term has become something of a generic reference to a family of austenitic nickel-chromium-based superalloys which have good strength at extreme temperatures and are resistant to oxidization and corrosion. Because of the excellent high-temperature properties, Inconel can offer increased reliability in header systems, and in certain applications, it is the only material which will do. The high-temperature strength properties can enable weight-reducing designs, since, for a given reliability requirement, Inconel allows the use of much thinner-wall tubing than could be used with other materials. The catch, as usual, is that Inconel tubing is quite expensive.

Certain Inconel alloys retain very high strength at elevated temperatures. One of the favorites for header applications is Inconel-625, a solid-solution alloy containing 58% Nickel, 22% Chromium, 9% Molybdenum, 5% Iron, 3.5% Niobium, 1% Cobalt. It has good weldability using inert-gas-shielded-arc processes, and good formability in the annealed condition, and has a lower thermal expansion rate than the stainless alloys commonly used in exhaust systems. Weldability and formability are both important because of the somewhat limited availability of Inconel tubing sizes, which often makes it necessary to form tubing sections from sheet. The yield strength of this alloy at 650 °C (1200°F) is 345 MPa (50 ksi), while at 870°C (1600°F) it is a remarkable 276 MPa (40 ksi). As with many metals, the high-temperature strength diminishes as the amount of time the parts are exposed to extreme temperatures increases.

Inconel tubing is nearly essential in high-output turbocharged applications, and I was told by several knowledgeable players that all the Formula-One cars and a few Cup teams use Inconel for their headers, both for reliability and for weight savings.

One builder told me that some teams are routinely using headers made from 0.50-mm (0.020 inch) wall Inconel tubing. He also told me that, in view of the immense heat load imposed by the exhaust gasses of contemporary Formula One engines, he seriously doubted that a set of stainless headers, even in 1.6-mm wall (0.065 inch), would survive. **Figure One**, a BMW F-1 engine at full power, graphically illustrates this demanding environment.

There are several austenitic stainless alloys which are commonly used in exhaust systems. In order of reducing temperature capabilities, they are 347, 321, 316 and 304. In addition, special variations in the basic alloy chemistry (carbon, nickel, titanium and niobium) are available to enhance the high temperature strength of these alloys.

Regarding the use of stainless, I was told by a knowledgeable source that in NASCAR Cup racing, the 304 and 321 stainless alloys were used more often than Inconel, depending on the preferences of the various teams. The manager of one prominent team told me that, in view of the facts that thinwall Inconel headers are (a) very fragile and readily damaged by inadvertent mishandling, (b) "grotesquely" expensive, and (c) provide almost immeasurable gains on a 3600 pound vehicle, his opinion is that the use of Inconel headers is not prudent stewardship of his resources. For a peek at the magnitude of the costs involved, one fabricator told that a single 1-D "U" bend of 2-inch diameter, 0.032-inch wall Inconel tubing would cost somewhere in the neighborhood of $200, whereas the same bend in 321-stainless would be in the $65 range.

Although titanium has been made to work quite well in exhaust valve applications, the practical temperature limits for titanium alloys suitable for tubing is quoted at about 300 °C (575 °F), which makes that material suitable for lightweight tailpipes in various applications and in certain motorcycle applications as well. My favorite supplier of titanium reports that grades 1 and 2 commercially-pure (CP) titanium have been used for the exhaust systems on competition 2-stroke motorcycles for decades. For lightness, many of these systems were made using 0.50 mm wall tubing, and treated as a consumable, being replaced after every meeting.

One might ponder why the same materials used for titanium exhaust valves are not used for exhaust tubing. Apparently, the simple reason is cost vs. benefit, since the estimated cost of thin sheets of Ti-6242 were estimated at over $150 per pound in large-quantity purchases. Add to that the fact that this material lacks the ductility to be readily formed into tubes, plus the fact that there would be problems welding the seams of a rolled tube, and yet more problems forming the welded straight tubes into bends, and it becomes evident that there are more suitable materials for exhaust tubing use.

## Formula One

Recently, I had the opportunity to hold in my tired, worn hands, a primary header tube which was alleged to have been for a nearly-contemporary F-1 application. Pictures of said hardware were not allowed, but the reproduction from memory, shown in **Figure Five**, illustrates the very interesting feature, the existence of a large-diameter step in the primary, quite close to the flange.

Figure 5
Formula-One Primary Header Tube

The illustration shows a single 10-mm step spaced approximately 125 mm from the flange. However, experts say that in 2008, two smaller steps (5 mm each) in the primary are more commonly seen, depending on the research and beliefs of the developers. The first step is typically between 100 and 200 mm from the flange. If there is a second step, it is typically another 100 to 150 mm beyond the first step, and in general, tubing sizes range from about 50 mm to 65 mm. (1.97" to 2.56"), although the specific designs seem to vary dramatically from team to team.

My first impression, which was shared by a number of experts with whom I spoke, was that, since these engines are operating up to 19,000 RPM, then the primary length required to achieve the negative pressure pulse during overlap was so short that, due to packaging constraints, the location of the collector would be too far away from the valves to initiate the properly-timed reflection. However, a bit more thought and a quick calculation revealed quite a different theory.

For purposes of approximation, assume that the mean temperature of the exhaust gas in the primary up near the head is 1500°F (815 °C). The speed-of-sound-in-air equation (close enough for approximations, according to Professor Blair) produces a sonic velocity of 661 m/s (2168 feet per second). At 18,000 RPM, (300 RPS) one crankshaft rotation takes 3.33 milliseconds (ms) or 3333 microseconds (µs). Therefore one degree of crank rotation takes 9.26 µs (3333 ÷ 360). If the first step in the primary is 200 mm from the back of the exhaust valves, then using the calculated speed of sound as an approximation of the propagation speed of the finite pressure wave, the 400 mm round trip from the valve to the step and back takes about 600 microseconds, or 65 degrees of crankshaft travel.

Assume that, in an 18,000 RPM engine, the establishment of enough exhaust valve opening to allow meaningful flow would occur in the neighborhood of 100° after TDC. Therefore, it is clear that this first reflection is timed to arrive back at the valves even before the piston reaches BDC. For what purpose? Recalling that during blowdown, there is sufficient pressure ratio in the cylinder to establish choked (sonic) flow through the exhaust valve orifice, then it would certainly be advantageous to maintain that gas velocity for as long as possible.

A noted engineer in the world of Formula-One confirmed that this is exactly the reason for the one-or-more large-magnitude steps in the primary: to place a negative pressure at the back of the exhaust valve timed so as to extend the duration of the critical pressure ratio.

## NASCAR Cup

The required Cup engine configuration (90° V8 with a two-plane crankshaft) provides an interesting challenge for exhaust system designers. Because of the firing order of this engine configuration, the exhaust pulses on each bank of the engine are unevenly-spaced. In the words of the technical director of one prominent team: *"The exhaust system design in Cup is an interesting tradeoff between minimizing flow losses while at the same time trying to optimize whatever tuning you can do with a non-equally-spaced system, which isn't a lot."*

With the GM cylinder-numbering system (1-3-5-7 on the left) and firing order {18436572; the 4-7 swap is not allowed in Cup} ), the exhaust pulse spacing on the left side (expressed in terms of degrees of crankshaft rotation) is 270°-180°-90°-180° while the spacing on the right side is 90°-180°-270°-180°. This uneven pulse spacing gravely impedes the achievement of a well-tuned exhaust system such as can be achieved with evenly-spaced pulses and a 4-into-1 collector.

That tuning difficulty led (more than a decade ago) to the re-introduction of the 4-into-2-into-1 (so-called "Tri-Y") configuration, which has been around since at least the 1960's. In the "Tri-Y", cylinders on each bank are paired so as to provide the maximum separation between pulses. Using the above numbering scheme, the primaries of cylinders 1 & 5 and 3 & 7 would be merged into slightly larger secondary pipes, which after the appropriate length, would be merged into the larger collector. On the right side, adjacent primaries are paired (2 & 4, 6 & 8). That provides a 450°-270° separation between pulses in each secondary. An example of this configuration is shown in **Figure Six**.

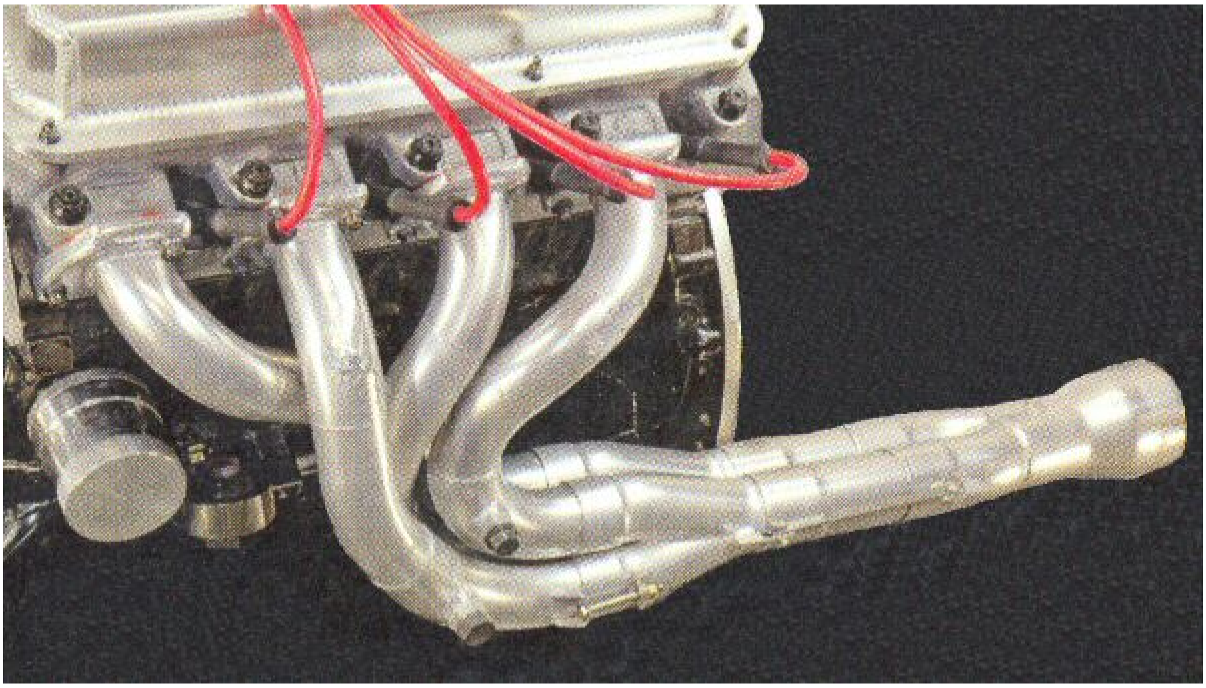

Figure 6
Example of a 4-2-1 Header System

The tuning of this type of system is not terribly intuitive. Several well-placed experts in Cup told me that their teams have consumed large amounts of modeling time using very sophisticated (and expensive) simulation software to arrive "in the ball park", and then fine tune the designs on the dyno. And, as would be expected, there are different header designs for long tracks, short tracks, and restrictor-plate tracks.

One expert mentioned that, while it is relatively straightforward to accurately model the behavior of the primaries, it is very difficult to accurately model the secondaries and collectors, because the theoretical reflections are meaningfully altered by specifics of geometry (bend radii, intersection angles, nozzle and diffuser angles, etc.) which cause destructive interference and pulse attenuation. That being said, several experts agreed that the rules-of-thumb still apply: better low end needs smaller and longer tubes; better high end needs bigger and shorter tubes.

There are additional challenges in Cup header design and tuning. The chassis teams often impose a major set of constraints on primary length and bend location so as to not interfere with critical items such as upper control arm pivot locations. The prevailing view is that, in terms of lap times, making the car turn better is a reasonable trade-off against a small amount of power increase. The straightforward header shown in **Figure Six** simply to illustrate the concept, is a dyno header, built almost without regard for any packaging constraint. Consider how difficult it might be to implement that concept within the very tight engine compartment of a Cup car, constrained by intruding frame tubes, suspension pickup points, a 230-mm long external oil pump, and the like.

Given the existing packaging constraints, it is indeed fortunate that the primary lengths in the 4-2-1 system are not nearly so critical as are the lengths of the secondaries. Several experts told me that the engines are very sensitive to changes in the length of secondary sections, and that most of the development effort is focused on secondary merge, length, diameter and step issues.

Moving rearward, the NASCAR Cup rulebook provides some interesting insight into additional exhaust system challenges. The rules include the requirements that the exhaust system for each bank of the V8 engine must be completely separate and may not connect in any location except for a single "X" or "H" pipe in a tightly-constrained region of the tailpipes, and must end with two tailpipes which exit under the frame rails within a tightly-constrained area on the right side of the car. Further, the pipes from the collector to the exit must be magnetic steel, no larger than 101.6 mm (4.0 inches) ID, and may have a circumference no greater than 336.5 mm (13.25").

The circumference restriction provides a subtle challenge. In order to fit beneath the COT frame and still provide ground clearance, the large diameter tailpipes are reshaped into a cross-sectional form having two long parallel walls (no closer to each other than 51 mm ) and a full radius at each end, such as illustrated in **Figure Seven**.

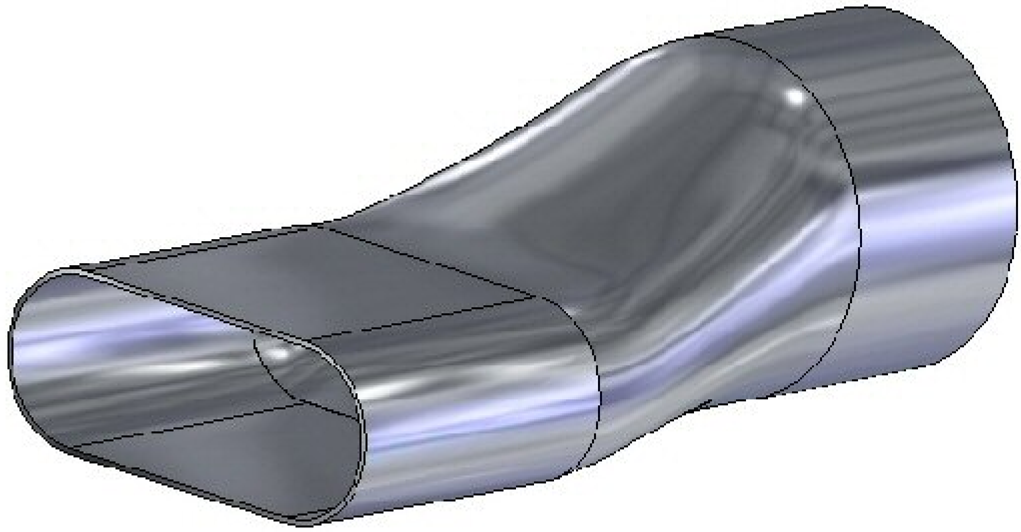

Figure 7
NASCAR Under-Frame Exhaust Pipe Exit

Because of the fact that a circular section provides the most cross-sectional area for a given circumference, the necessary ovalling of the pipe exit puts an orifice at the end of the tailpipe. If the exit section is the minimum height of 51 mm, the limiting circumference (assuming 1.6 mm-wall tube) yields a cross-sectional area which is only 77% of the 101.6 mm round tailpipe. That reduced area can be a flow restriction at high RPM.

## Top Fuel and Funny Car

At the top levels of drag racing, in particular Top Fuel and Funny-Car, the exhaust systems might seem very simple. The header systems, known as "zoomies," consist of a single pipe on each cylinder, dumping straight into the atmosphere, with each tube bent so that it faces upward, rearward, and often outward. The outward angle of bend in Funny Cars is typically larger than would be seen in an unbodied Top Fuel car, in order to eliminate bodywork damage from both temperatures and exhaust concussion forces.

In addition to the noise, a notable feature of these exhaust systems is the large volume of open, whitish flame standing just off the ends of these pipes, as shown in **Figure Eight**. That flame-front is the byproduct of two intersecting parameters.

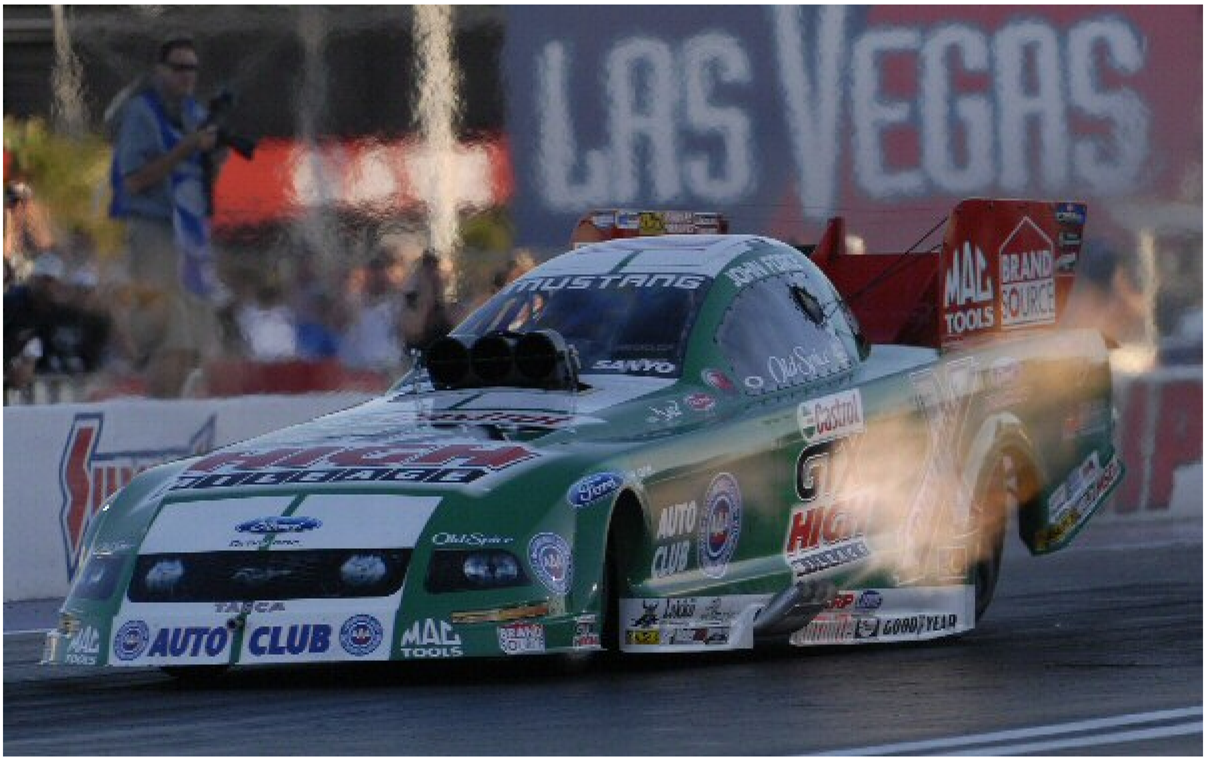


Figure 8
Secondary Combustion

First, these highly-supercharged, nitromethane-nourished engines have fuel flow rates stated to be in the 80 to 90 gallon-per-minute range. With that amount of fuel being delivered, it is clear that there will be a certain amount of fuel puddling behind the intake valve. When the intake opens, some portion of that collected fuel will be either in liquid form or in a mixture which is too rich to burn (insufficient oxygen molecules). Further, these engines apparently use a large amount of overlap in order to assist in cooling. The combination of the excess fuel and the long overlap assures that a non-trivial quantity of raw fuel and fuel mixture is short-circuited directly down the exhaust pipe, and heated during its journey. When it exits the primary, it finds an abundance of oxygen and initiates an energetic secondary combustion. The combination of the large momentum-change of the mass flow through the engine, plus this secondary combustion has been calculated by at least one aerospace engineer to generate normal reaction forces in excess of 2500 pounds (1130 kg).

Given that the pipes are angled in both the lateral and longitudinal planes, that exhaust reaction force can dramatically affect the vehicle stability. The vertical component obviously provides downforce to the chassis. The rearward component will add propulsive thrust. If everything is in balance, the sideward components generated by the left and right sets of pipes should counterbalance and net to near zero. However, I was told that the loss of one cylinder on a Funny-Car can cause the driver to have real difficulty controlling the car. That is because the loss of a single cylinder unbalances the sideward-thrust and adds a yaw-moment from the now-asymmetric rearward thrust. That same (highly-credible) source told me that the loss of two cylinders on the same bank will amost certainly render the car uncontrollable.

As might be expected, the length of the primaries plays a critical role in the engine tune. I was told by a lead engineer on a prominent Funny-Car team that there was a considerable amount of development effort required just to get the gasses out from underneath the Funny-Car bodywork.

That source also said that when they tried collector-systems, the result was that the engines ran "horribly". The theory is that the huge amount of exhaust gas flow into a relatively-confined space raised the collector pressure enough to create a destructive blockage in the collector pipe.

As far as the pipes themselves are concerned, it is well known that "too sharp a bend" in the primary or "too much length" dramatically reduces engine performance. Apparently, in supercharged nitromethane engines, any tuning on the exhaust side (cam, ports, headers) requires a substantial alteration in the fuel delivery curves. After experimenting with various exhaust system changes, then working to get the fuel system back into line with the engine changes, the net change in performance was typically considered to be not worth the time and effort. After having determined a working combination, experience has shown that development efforts in areas other than the exhaust system will be more productive.

I was told that currently, there is not a large amount of development effort on the Funny Car exhaust system, as the result of several practical and economic factors. It is hard to imagine the level of difficulty involved in doing engine development on a system which is not well suited to a dyno cell, and therefore must be tested on the track in 5-second test sessions. Without taking into account salaries, logistics, transportation, food, lodging, and other "overhead" expenses, the out-of-pocket cost to make "one more test run" is uncomfortably close to ten thousand dollars.

### Moto-GP

Neil Spalding, Race Engine Technology's in-house expert on motorcycles, provided me with a gallery of detailed photos showing the varied strategies employed in Moto-GP (the F-1 of motorcycle racing) to shape the engine power curves with exhaust tuning finesse, along with a wealth of information on these machines, including the fact that the use of Inconel tubing is fairly common.

In several RET articles, Neil has discussed the difficulty in getting the available power to the ground in Moto-GP, and the efforts which the manufacturers have taken to improve the available traction, including implementation of uneven firing orders so as to affect the tire contact patch in a beneficial way. The uneven spacing of exhaust pulses requires some out-of-the-box thinking to gain benefit from exhaust tuning. In order to linearize the engine power curve (flatten the torque curve) there has been widespread usage of the 4-2-1 design described above in the Cup section.

These systems use various techniques specific to the particular engine, including diverging tapers in the primary tubes just past the flange, steps in the primary tubes, converging-diverging collectors, straight collectors, diverging tapered collectors, and more.

**Figure Nine** shows the torturous 4-2-1 system developed for the 2005 Yamaha 990 cc irregular-fire inline 4. The picture shows the diverging taper in the primary just past the flanges. Neil told me that the current system for the 800-cc engine has substantially shorter primaries and secondaries due too the fact that the 800 cc engines turn up to 18,000 RPM, where the 990's were in the 16,000 RPM range.

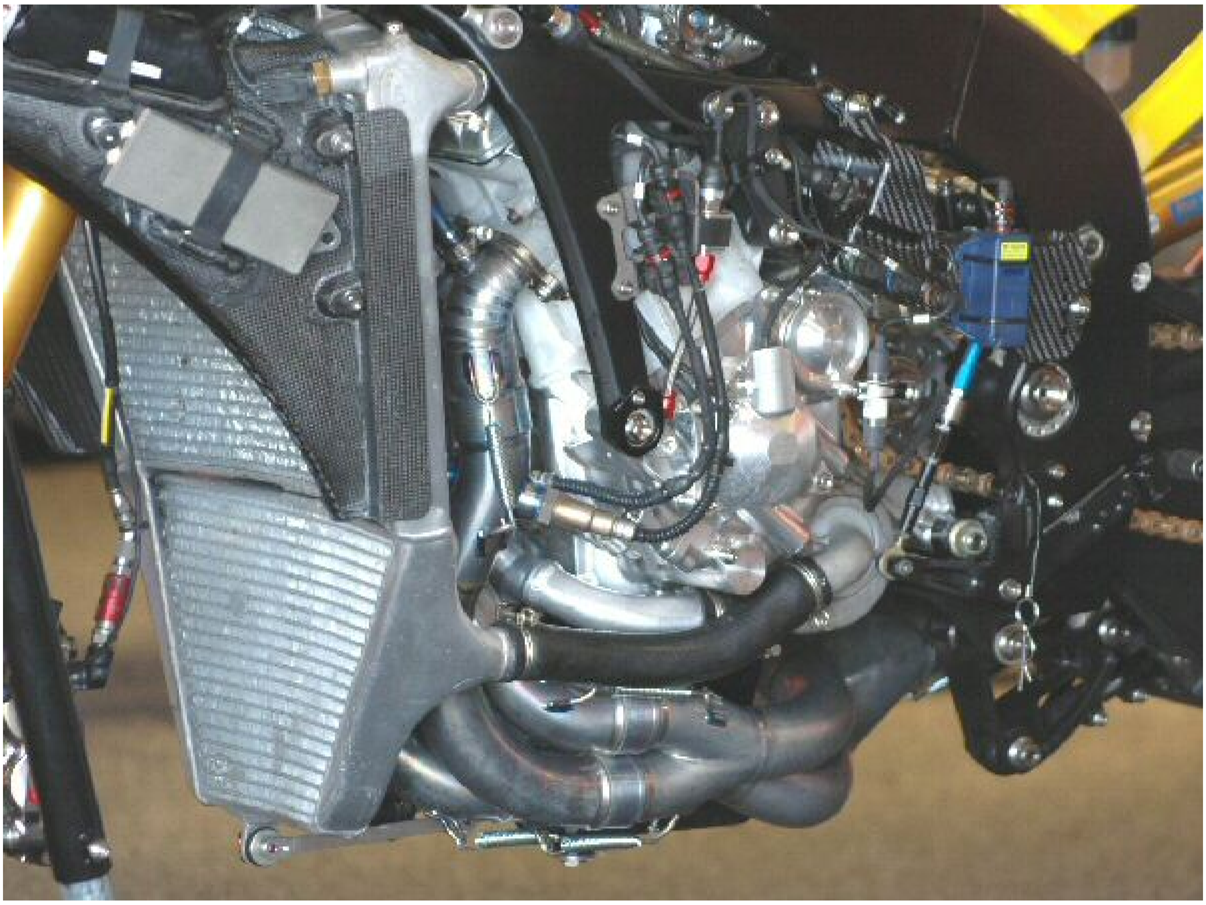

Figure 9
2005 Yamaha 990

**Figure Ten** shows the individual stacks used on an experimental Kawasaki 990 cc engine, which reportedly had a flat-plane crankshaft but which fired pairs of cylinders together. Note the very long tapered expansion pipes and reduced exit diameters, which will help reduce the extreme power-curve peakiness that occurs when a primary opens directly into the atmosphere (which constitutes an apparent infinite expansion area ratio). Note also how the lower tube has a longer centerline length and a longer tapered end. That too will help to spread the potentially very peaky tune of these pipes over a wider RPM band.

Figure 10
2005 Experimental Kawasaki 990

## Turbocharged Applications

According to the turbocharger engineers, the most important aspect of designing a good header system for a turbocharged application is to maximize the recovery of exhaust pulse energy. This energy recovery has at least two components.

The first is to provide evenly-spaced exhaust pulses to the turbine. To accomplish that, it is helpful first to be working with an engine (or bank of an engine) which has evenly-spaced firing intervals. In an application in which the cylinders feeding a given turbine or turbine section have even spacing, the lengths of the primary tubes should be as close to equal length as possible.

The second component is to maximize the recovery of pulse velocity energy. For that purpose, turbine housings are available in split housing, or "twin-scroll", configurations, in which there is a divider wall in the center of the turbine nozzle housing to separate the incoming flow into two separate streams. That allows the nearly ideal pulse separation of 240 crankshaft degrees to be achieved on an inline-6 engine by grouping the front 3 cylinders into one side of the housing and the rear three cylinders into the other side. The same effect can be achieved on a V6 engine by grouping each bank separately.

Although the split housing arrangement adds wetted area (hence boundary layer drag) to the gas flow, the advantages more than offset that drag increase. In instances where pulse energy recovery has been optimized, it is often possible, based on calculations using pressure and temperature losses across the turbine, to observe very high turbine efficiencies, which some experts say are in excess of 100%.

Pulses which are evenly-spaced but too close together will reduce the effectiveness of this pulse energy recovery. Apparently, that phenomenon is seen on even-fire inline 4-cylinder engines as well as on individual banks of flat-crank V8 engines, where the pulse separation is 180°. I was told that the ideal pulse separation was in the neighborhood of 240 crankshaft degrees, and that, on an even-fire (single-plane crankshaft) inline-4 (as opposed to the two-plane crankshafts used in some Moto-GP motorcycle engines) it is better to separate the end cylinders into one side and the center two into the other side of the turbine than to run all four together into an undivided scroll housing. The same reasoning applies to each bank of a V8 with a single-plane crankshaft.

With regard to the uneven pulse spacing of each bank of a two-plane crank V8, there is agreement that it is very difficult to organize the pulse spacing in a useful way. It has been demonstrated that where a small turbo is used on each bank, the use of a short-tube 4 into 2 system (same idea as the 4-2-1 discussed above) feeding a twin-scroll turbine could take some advantage of the resulting 450 - 270 separation in terms of pulse energy recovery. If a single, large turbo can be located in such a way that the tubing lengths from each bank can be fairly equal, then splitting the primaries to achieve 180° separation would be an advantage.

Whenever practical, reducing the heat (energy) losses before the exhaust gasses reach the turbine allows the turbine to be more effective. This has been done with double wall tubing, reflective coatings, and wraps. However, insulating the pipes to reduce heat loss will, of course, raise the operating temperature of the pipes themselves, which can require extreme-duty materials where more affordable materials would suffice in the un-insulated form.

Another important exhaust system consideration, in order to provide the most effective operation of the wastegate in controlling boost, is to position the wastegate inlet port so that it is subject to exhaust stream total pressure, rather than off to the side where it sees only static pressure.

**Source:**

**http://www.epi-eng.com/piston_engine_technology/exhaust_system_technology.htm**