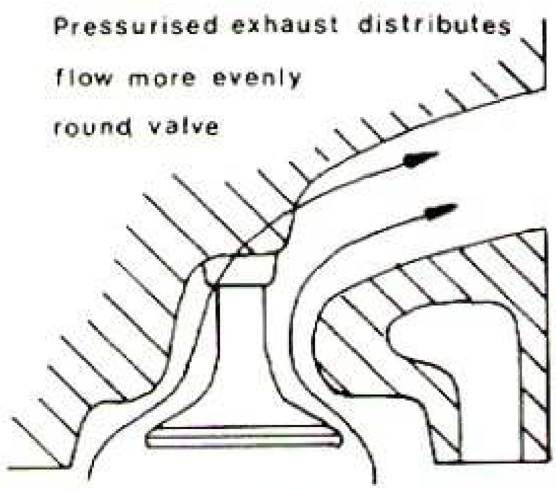
CHAPTER 4.
THE EXHAUST SYSTEM.

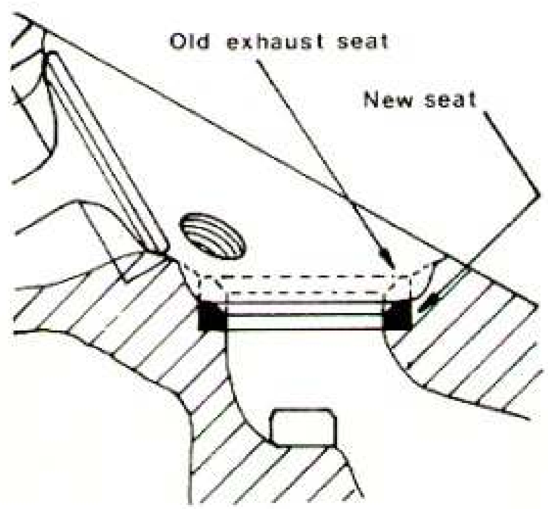
Four – stroke engines.

The four-stroke exhaust system consists of considerably more than the visible external manifold and pipe, and starts at the exhaust valve.
The valve itself needs to be much heavier in design at the back of the head than the inlet valve, in order to cope with the hot and scouring exhaust blast.
Do not be tempted to remove any of the protruding valve guide.
The valve is continuously struggling to shed its heat, and relies on contact with the guide and seat for this process.

For the same reason, the seat width must always be greater than for inlet valves and should be between 2.0 and 2.5 mm, depending on cylinder and valve size.
Modifications to the exhaust port should consist of increasing the bowl area around the stem and guide, and then a smooth blend to the chosen pipe size.



61 *Correct exhaust port shape.*



62 *Recessing of exhaust valve seat.*

As far as shape is concerned, a short straight section blended into an updraughted port gives the best flow/diameter ratio. (Fig. 61)

Just as with the inlet port, radiussing sharp corners will also improve flow.

Effect of the exhaust cam.

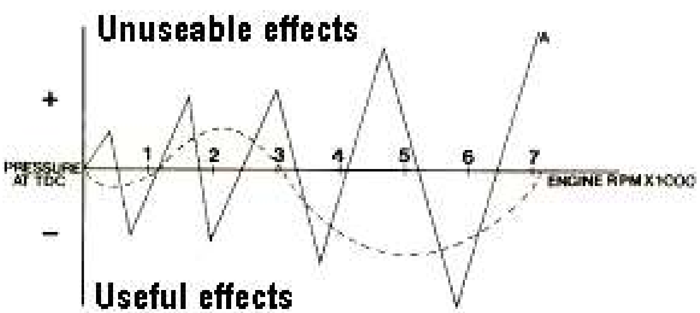
When fitting a full-race cam, the valve lift at T.D.C. is considerably increased.
A 400cc cylinder, in standard trim, will have about 1.0mm of lift at overlap T.D.C.
When a full-race cam is fitted, this will increase to around 5.0mm. This happens to both inlet and exhaust valves, so valve to piston clearance and valve to valve clearance is reduced considerably.

One way to alleviate this problem is to pocket the exhaust valve head. (Fig. 62).
Unlike the inlet, the flow capability of the exhaust actually improves with partial masking.

This phenomenon presents us with the convenient option of recessing the exhaust valve to improve valve-to-valve clearance.
However this procedure also results in the exhaust valve stem being too long for the valve spring fitted length, or too close to the cam lobe to allow suitable clearance adjustment.
This problem is dealt with further in Chapter 8.

Exhaust pipe length and size.

As the exhaust valve opens, a positive or pressure wave front is created which travels down the exhaust pipe at the speed of sound.



As this pressure wave reaches the end of the pipe, it expands and a negative or suction pulse travels back up the pipe towards the engine. As the negative wave front in turn reaches the cylinder, it reverses again and moves back towards the end of the pipe.
This fluctuating pressure pulse effect can be used to great advantage in tuning the engine.
If the system is designed in such a way that the negative or suction pulses return to the cylinder at overlap T.D.C., then they will assist in clearing the combustion chamber of exhaust gases.
In turn, this will cause a depression at the inlet valve, which will help draw in the inlet charge.

Coupling the pipes of multi-cylinder engines will also mean that the pulse effects from one cylinder can be used to assist the breathing of another.

The following formula can be used to calculate the ideal length for a given application:

$$L = \frac{129540 \times \text{E.T.}}{\text{R.P.M.} \times 6}$$

Where:
L = Primary pipe length in mms measured from the exhaust valve head.
E.T. = Exhaust valve duration in degrees from point of valve opening before B.D.C plus the full 180 degree stroke up to T.D.C.
R.P.M. = The estimated revs, at which max. power will be achieved minus five hundred.

Example:
Exhaust timing = 80 B.B.D.C. to 50 A.T.D.C. Estimated maximum power R.P.M. = 7200
E.T. = 80 + 180 = 260
R.P.M. will be 7200 - 500 = 6700
Therefore :

$$\text{Primary pipe length} = \frac{129540 \times 260}{6700 \times 6} = 837 \text{ mms. or } 32 \text{ ins.}$$

Having calculated the primary pipe length, we must now calculate the diameter as follows :

Divide "L" by 10 to bring it to cms. Call this "L2". (83.7)
Take the cylinder capacity in ccs and double it. (Say 400 x 2 = 800)
Divide by "L2" as previously calculated. (800 / 83.7 = 9.56)
Divide by 3.4 (9.56 / 3.4 = 2.8)
Find the square root ($\sqrt{2.8} = 1.67$)
Multiply by two and add 0.3 ((1.67 x 2) + 0.3 = 3.64)
Multiply by 10 to bring it back to mms. (10 x 3.64 = 36.4) 36.4mms = 1.43ins
This will give the O.D. of the tube in which at first sight will appear rather small.
This is because it assumes the use of a perfectly smooth straight pipe, which is impractical to use, so the following allowances must be made.
To allow for the viscous drag created in the bends used in an "average" primary pipe and also to allow for the slight pipe flattening that takes place at the bends, increase the internal cross-sectional area by 10-15%, depending on how tortuous the system is.
This will probably finish up as a pipe size that is non-standard, so go for the nearest available stock diameter above this figure.
Remember that "L" is from the exhaust valve head, so the exhaust port length will have to be deducted to get the actual manufacturing length.

This will then give the joining point of the primary pipes.
From this point, the secondary or tailpipe length can be "L" or any multiple of "L" and its diameter can be calculated using the method above, but by starting off with four times the cylinder capacity for a four cylinder engine, or three times for a "six".

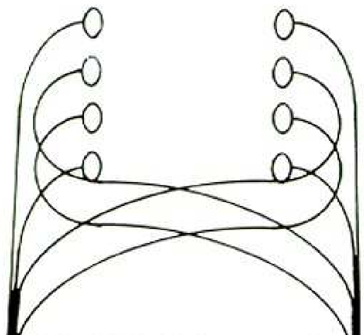
For maximum power development, "fours" should always finish up in a single tailpipe (Fig 64), while "sixes" should finish up with twin pipes, one of which couples cylinder numbers 1,2 and 3, the other coupling numbers 4, 5 and 6.

For street use, fours can also be designed with a secondary pipe set (Fig. 65) which, although not giving quite the same maximum power, gives a much broader spread of power.
The secondary pipes need to equal or be a multiple of “L”, with the next stock diameter up on the primary.

64 *4-into-1 exhaust system on 4-stroke motorcycle.*

65 *4-into-2-into-1 tuned 4-stroke exhaust manifold.*

Production V8s with 90 degree cranks should ideally have a crossover system (Fig. 66), but this is usually impractical, in which case they can be treated as two "fours", but will need a balance pipe linking the tailpipes at a tuned length, calculated as above.
Racing V8s with 180 degree cranks can be correctly treated as two "fours" with no balance pipe necessary (Fig. 67).



Ideal V8 90deg crank crossover system

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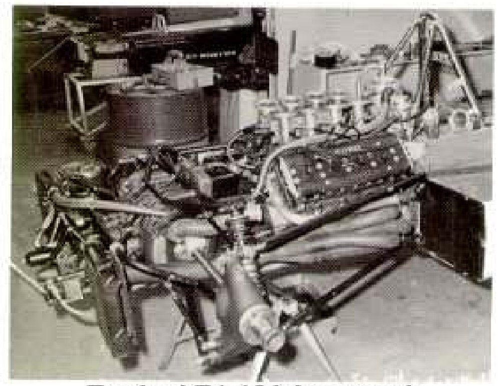
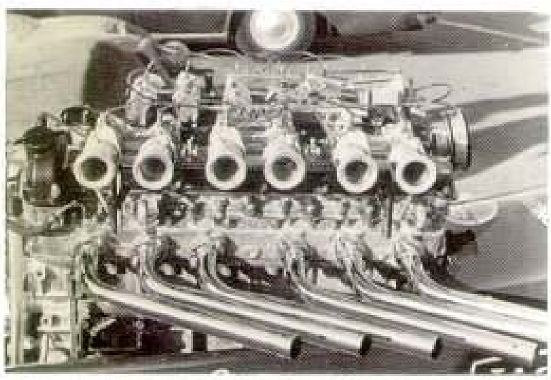


67 **Typical F1 180deg crank exhaust system**

If a silencer is used, expansion will take place at this point, so the start of the chamber should occur at a "tuned" tailpipe length, with additional tailpipe added to clear exhaust gas as necessary.



68 **Typical open stubbed exhaust system of drag racer**

Dragsters, using only wide-open throttle, requiring little or no progression, will typically use open stubs to give the highest, shortest band of power. (Fig 68)

The following chart is only intended as a guide to typical characteristics and requirements.

Individual engine applications will vary with engine characteristics and slight changes will be needed to achieve maximum performance.
Exhaust valve timings for turbocharged and supercharged engines are dealt with in the relevant section, later in the manual.

Exhaust timing selection chart

Application	Exhaust opens B.B.D.C.	Exhaust closes A.T.D.C.	Av.max. power R.P.M.	E.T.
Standard Engine	50	20	5500	230
Stage 1 Street	65	30	6200	245

Stage 2 Rally, M/Cross	75-80	40-50	6800	260
Stage 3 Advanced Rally and Motor Cross	82	54	7400	262
Stage 4 Full Circuit Race	88	56	8000	268
Dragster	90+	60+	8000+	270+

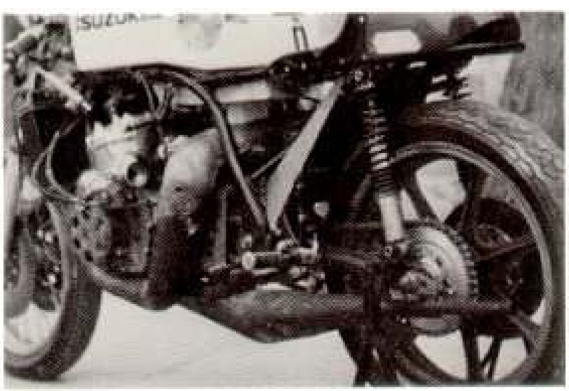
Two-stroke engines.

The pressure pulse reversal as described earlier also takes place in the two-stroke exhaust system and can be used to much greater effect if correctly manipulated.
This is due to the fact that the whole breathing process is dependant on a transfer of pressure from one area to another and is not positively valve controlled as it is in the four stroke.

The rules of good design are not nearly as easy to define, and best results are ultimately only achieved by exhaustive dynamometer testing.
Even then the results obtained will not apply to another engine if there is any slight variation in timing.

Multi-cylinder coupling is not mathematically feasible except with three cylinder configurations and even then, the improvement comes only in the mid-range, often very useful to spread the torque of highly tuned engines.

In order to obtain ultimate power from multi-cylinder two-stroke engines, it is necessary that they be treated as a group of single cylinders.



70 *Suzuki 2-stroke race bike with tuned exhaust system.*

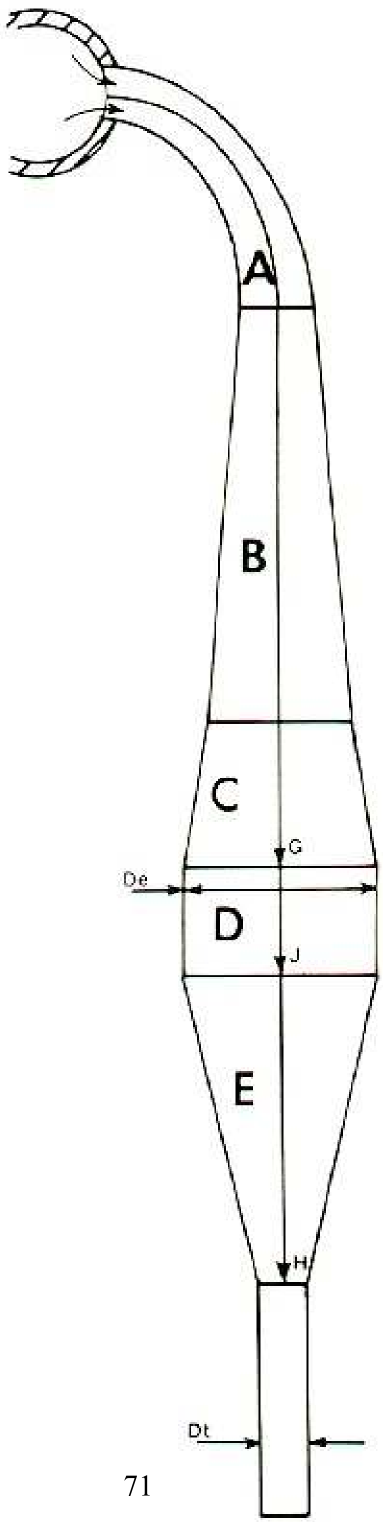
This often creates great difficulty in accommodating the mass of snake-like hardware. (Fig. 70)

The separate geometric components of the exhaust system are laid out in Fig. 71 and function in the following manner :

(A) Primary Pipe.

Often a parallel tube, particularly in cheaper production road machines, but ideally a tapered primary pipe should be used to control the expansion rate of the high speed gas slug ejected from the port and to convert its kinetic energy into pressure energy.

(B) Primary Divergent Cone.
Controls initial expansion of the pressure pulse and is often combined with :



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(C) Secondary Divergent Cone.
Which finally controls the pressure pulse expansion to induce the negative pulse, which travels back to the port to help scavenge the cylinder.

(D) Expansion Chamber.
Length acts as a time control before throttling of the gas slug which starts at :

(E) Convergent Cone.
Which throttles down the slug to the :

(F) Tail Pipe.
The size of which controls the high back pressure reverse "plug", which in turn pushes the overspill of intake charge back into the cylinder before the piston shuts the door.
The whole sequence, using correctly designed components, will result in a cylinder filling efficiency of more than 100% at the "tuned" engine speed.
True design formulae for these systems are highly complex and still not quite fully understood, but outlined below is a simplified starting point for those who want to have a go themselves :

$$\text{Length "G" in mms} = \frac{41910 \times TD}{R.P.M.}$$

Where TD is transfer port duration in degrees.
R.P.M. is desired "on pipe" R.P.M.

$$\text{Length "H" in mms} = \frac{41910 \times ED}{R.P.M.}$$

Where "ED" is exhaust port duration in degrees.

$$\text{Length "J" in mms} = \frac{41910 \times (TD + C)}{R.P.M.}$$

Where "C" will control the length of the parallel section and lengthen time before reverse plug starts.

Find this empirically by starting at 0 and increasing in intervals of 5.
"De" will be around 2.25 x Piston diameter for high revs and 2.0 x Piston diameter for torque.
"Dt" will be around 0.45 x Piston diameter for high revs and 0.5 x Piston diameter for torque.

Exhaust emissions.
Although not an important consideration in Europe when tuning for performance, exhaust emission control is becoming a major factor of engine design consideration.

When the air/fuel mixture burns in the combustion chamber, a number of chemical changes take place that result in the pungent and easily recognised exhaust fumes emitted from the tail pipe.
These consist of harmless gases like oxygen and carbon dioxide, and the deadly gases like nitrous oxides, carbon monoxide and hydrocarbons.
Nitrous oxides are usually the result of high combustion chamber temperatures, so these will often rise when tuning takes place.
Hydrocarbons are formed when small, trapped pockets of gas are left partially unburnt or mixtures are run too rich.
Well-tuned engines usually have low hydrocarbon outputs.

It is almost inevitable that, as Californian law already decrees, European exhaust emission legislation will eventually mean that tuning kits may not be sold unless they conform to the clean air laws.
Hopefully though, it will not apply to motor sport events.