Chapter 6.

Induction Fundamentals.

In a high performance racing engine the induction process is very dependent on the exhaust process but this is only true in a minor way if at all for more modest applications. The dynamics of the induction process are a consequence of the low pressure created by the descent of the piston but the dynamics of the exhaust process are generated by the egress of gas at high pressure without regard for the motion of the piston at the time. Ironically the exhaust process can ultimately make a profound contribution to the induction process but that is something that will be dealt with in the next chapter.

Induction Matters.

It seems that nearly every article or book ever written about the induction process can hardly get beyond the first paragraph without mentioning the word ‘flow’ (making this no exception). In particular the many millions of words written in go-faster books and magazines are predominantly about the business of flow and how to improve it. The problem is that it is not enough to think just in terms of flow because the induction process is actually driven by pressure changes and too much emphasis on flow can actually have a negative effect on the end result.

The only thing that induces air to flow into the cylinders of an engine, is pressure difference and, all things being equal, the bigger the pressure difference the higher the flow velocity will be, to the benefit of both filling the cylinder and inducing charge movement to aid combustion.

It seems appropriate therefore to start looking at the induction process by considering how the pressure changes inside the cylinder from the moment the inlet valve opens to a point just after it finally closes (Fig. 6.1). It is the situation at full throttle that matters here because any part load condition is always the result of a deliberate loss of efficiency by throttling to match the reduced power requirement.

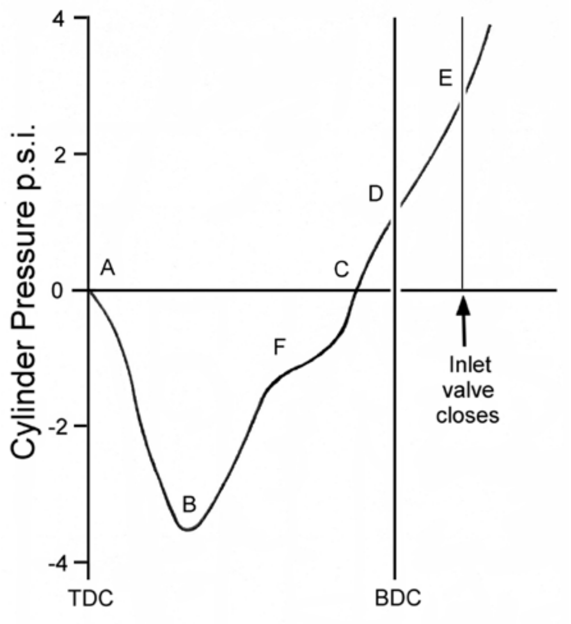

Fig. 6.1 Induction pressure curve.

Obviously the descent of the piston down the cylinder and the resulting increase in volume is what induces air to flow into it from the inlet valve and porting. At very low speeds there is plenty of time so there is not much change of pressure at all as air flows leisurely into the cylinder. With rising speed the pressure drop increases as does the speed of flow. The diagram shows how the cylinder pressure changes through the induction stroke of a typical engine running at moderate speed.

Assuming the pressure at TDC is very close to atmospheric (A) as the piston starts to descend the pressure drops fairly sharply until reaching a minimum peak (B) as the piston gets to about one third of the way down the stroke. Air will have been flowing in during this time or the pressure would be much lower still but the piston is now approaching its maximum velocity so the pressure starting to rise again is evidence that the inward flow must be accelerating rapidly as a result of the initial depression.

As the piston slows as it approaches the end of the downward stroke at BDC air is still flowing in at high velocity so pressure can actually start to rise beyond atmospheric (C,D). After BDC the piston starts to ascend which will cause the pressure to increase still more, assisted by the continuing in-flow due to inertia. Finally at (E) the inlet valve is closed and all flow ceases. The effectiveness of the entire sequence of events is determined by the pressure at (E) because if the rising cylinder pressure is more significant than the pressure developing in the port and the incoming inertia flow there will be backflow out of the cylinder so some of the charge will be lost. On the other hand if the valve closes early the inertia column will not have time to flow into the cylinder. Correctly timing this point is therefore very important for maximizing the air mass trapped in the cylinder at the commencement of the compression stroke but this ideal point gets later as engine speed rises.

The inlet valve closure point is determined by the camshaft and valve mechanism so for optimum performance over a wide speed range the mechanism must include some means of changing the cam timing for different speeds, i.e. delaying the closure point with rising speed. It was only during the 1990s that such mechanisms became widely used, driven also by other benefits with regard to fuel economy and control of exhaust emissions.

The other feature that is critical to the process is the deep suction peak (B) because this is where the pressure difference is created to generate the gas velocity that maintains flow after BDC. The size of the inlet valve(s) and port(s) and the valve lift are the main parameters that govern the velocity. Chapter 2 explained the significance of inlet gas velocity to the combustion process so it is important that the valve and port dimensions are chosen with care to suit the intended performance of the engine. A camshaft and valve mechanism for a very high speed racing engine will open the valves wider and for longer than one for a road going engine of the same cylinder size so the port and valve sizes may not have to be increased in size by as much as might be imagined.

This is partly because the inlet gas velocities in racing engines can be as high as Mach 0.5 which is consistent with the deep suction peak dropping to about 0.5 of an atmosphere, or around -8 p.s.i. Another factor is that pulse dynamics (to be discussed shortly) rely on changes of density so can achieve increased mass flow without increasing gas velocity.

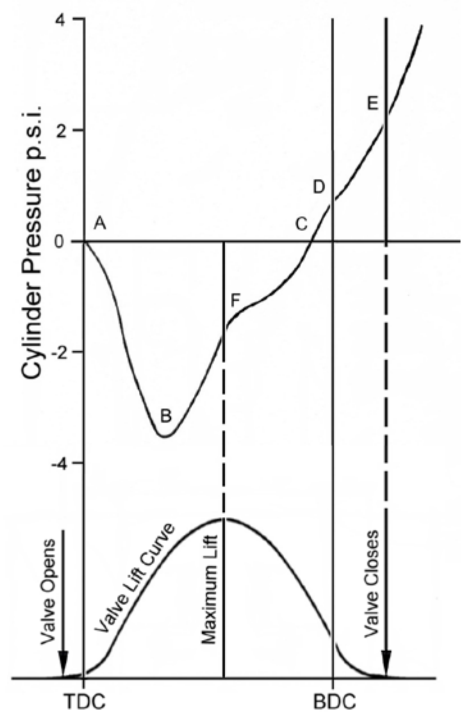

Fig. 6.2 Cylinder pressure against valve lift.

The pressure curve displays a rather curious kink (F) during the period where the pressure is recovering with increasing rate of flow when it might be assumed the rise would be steady. There are two reasons for this. Firstly, valve motion is generally timed so that maximum lift occurs at around 110 degrees after TDC which more or less coincides with point (F) after which the available valve area will start to reduce. This becomes clear when a curve plotting inlet valve lift is aligned with the pressure curve (Fig. 6.2).

The second reason is that the deep suction peak generates a negative pressure pulse that travels out through the inlet valve and the kink is partly the result of the pressure recovery after this phenomenon, which also can have other consequences of some importance as will be described presently. The kink often becomes a much more pronounced S bend showing a pressure reversal in engines with strong resonant dynamics.

Astute readers will hopefully now be realizing that if the port and valve can flow too easily too soon then it is likely that the deep suction peak will be less pronounced. The pressure difference then may not be sufficient to generate the high gas velocities necessary to complete the induction process to best effect. Of course increasing the engine speed will counter this effect but at the cost of losing performance at the original speed. Taken to extremes a high flowing port might only become efficient at engine speeds that are above the practical operating range so too much reliance on achieving maximum flow through the port and valve could actually lead to reduced volumetric efficiency and performance. It will also be obvious that any analysis of the port and valve must take into account the valve opening and closing cycle and the motion of the piston as defined by the stroke and connecting rod length.

INLET PORT SIZE READY RECKONER.

It has already been mentioned that if the inlet ports are made too big the gas velocity will not be sufficient for best volumetric efficiency but how can one quickly estimate the ideal size of port required without getting bogged down with data and complex software? One way is to simply divide the power output by the minimum port cross sectional area to derive a power/port area (PPA) index – i.e. b.h.p./sq inch. This is surprisingly useful because power is generally proportional to air mass flow and port area per b.h.p. is a reasonable indicator of how effectively an engine makes use of its available porting.

The important point about this simple index is that it takes into account not only volumetric efficiency but also mechanical and thermal losses. For instance reducing mechanical losses will improve the PPA index, as will an improvement of combustion, or an increase of compression, or an increase of gas flow through the same ports. On the other hand a poor PPA index will not be improved by increasing the port size alone because in such a case port area is unlikely to be the factor that is holding back performance. All things being equal a low PPA index indicates that there is probably good potential to increase the flow through the ports without making them bigger. Obviously a longer duration cam could improve the PPA but inevitably the engine performance would then be shifted to a higher engine speed.

Some Examples:-
1. Jaguar pre-HE V12 - 280 b.h.p. - total port area 13.5 sq”. PPA = 20.7
2. Jaguar Hydro V12 - 505 b.h.p - total port area 17.1 sq” PPA = 29.4
3. Cosworth DFV - 480 b.h.p. - total port area 15.2 sq”. PPA = 31.5
4. 2000 Ferrari F1 V10 - 817 b.h.p. - total port area 28.7 sq”(est). PPA = 28.4
5. 2004 F1 V10 - 900 b.h.p. - total port area 29.6 sq”(est). PPA = 30.4

Examples 1 and 2 illustrate the point quite well; 2 was a special hydroplane power unit that set world speed records but had to use standard valve sizes to comply with regulations. The ports could only be increased in area by 25% for a power gain of 80% which validates the prediction from the PPA figure for the standard engine that its port area is more than adequate. This engine formed the basis of later successful Group A racing engines. The high figure for the Cosworth DFV (3) is no surprise, and the slightly lower figures for later engines reflect the higher losses that result from running to very high speeds.

Go-faster magazines often feature photos of cylinder heads on flow benches with a ring of modeling clay around the edge of a port to provide a nicely radiused entry, yet how relevant can that be to how the same port behaves when connected to a complete induction system, particularly if the manifold tract has any sort of bend as it approaches the port?

The question also arises about what pressure to use in order to properly assess the flow efficiency of the port and valve. Measuring flow at some fixed pressure ratio (28“ water is the old SAE standard) does not seem to have much relevance to the rapid changes that take place in a real engine. One might reasonably argue that for a valid analysis, flow should be measured at a range of pressure ratios and valve lifts conforming to the curves shown above but of course that would not take into account the transient nature of the conditions for each test point. Even if useful data could be produced it then becomes logical to want to examine the effect of changes of valve lift and timing as well as port and valve geometry. A change that increases flow early in the opening phase may affect pressure at a later part of the cycle so the data must be updated across the whole cycle to ascertain the true effect.

Throw in the effects of unsteady flow dynamics (described shortly) in both the induction and exhaust systems and detailed analysis becomes a truly daunting task. Perhaps in desperation one falls back to relying on steady flow at 28” water pressure as an achievable compromise that is better than nothing, but the limitations are surely very obvious.

22 pages follow.