”Computer Power”

Introduction

The content of this article is, as you might guess, not about computer performance but rather how engine power can be predicted through the use of engine simulation tools.

Little of detailed information is available on power curves from later Velocette engines like Viper, Venom, Thruxton etc. True, some curves are published like the one in figure 1. However it looks like manually smoothed curves, rather than the rougher curves usually obtained from a dyno.

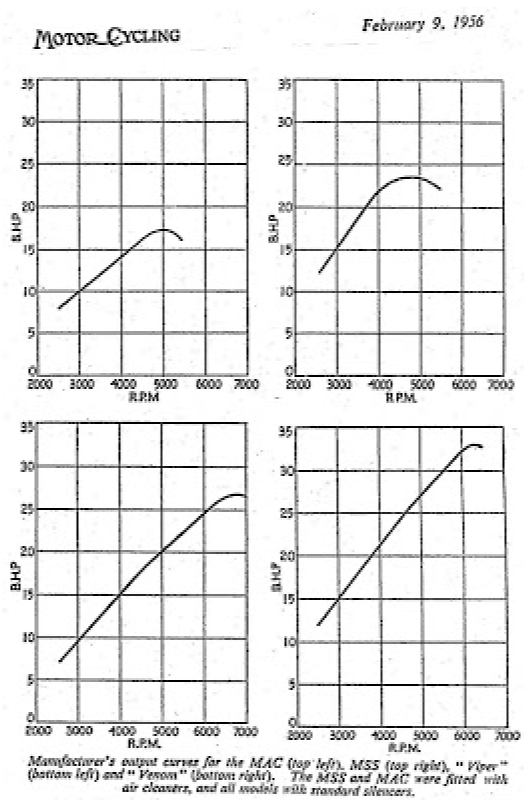

Figure 1 Power-curves from Viper and Venom bottom left and right. (Source: D Quinlan)

The claimed power output from the Venom engine varies from about 32 to 38 HP depending on year model, bike version (clubman- scrambler), where the scrambler is rated highest. The Thruxton with all its mods develops around 41 hp. (Ref: 1,2and3)

But what was changed in the Venom engine to account for the variation in power? Different compression ratios, different carb model different exhaust system. Was this sufficient to create a 6 HP increase over the years/models?

The Viper engine is claimed to develop 27 HP in all versions, even though early engines had a smaller carburetor and different camshaft (17/7) instead of 17/8 used on the later versions. These two very different cams should produce quite different power characteristic.

Having access to simulation tools, the idea came up to run power simulations on Velocette engines. In the following it is explained how simulated power curves are obtained and the inputs which are needed. Limitations in the engine modeling will also be covered.

Simulation tool – GT-Power

GT Power may sound a bit fancy, but GT is actually the initials of the company Gamma Technologies that has developed the simulation software. The software contains modules for various tasks like crankshaft, camshaft, cooling, emissions, exhaust noise (or sound) etc. (GT Suite). The main group of customers is likely to be car engine manufacturers, but the software is being picked up by motorcycle manufacturers, large diesel-engine manufacturers and research/development institutions. In this case the author belongs to the last.

So, what can we expect regarding accuracy compared to measured power. Quite high accuracy is claimed by the software company. With accurate input data regarding intake and exhaust system geometry, valve size and lift profiles, intake and exhaust-port flow coefficients, compression ratio, fuel type and air fuel ratio the predicted power should fall I the region of +-5 % of the measured power on a dynamometer (of course the old rule is still valid, garbage-in means garbage-out).

Having tested the simulation tool on different engines with verified power curves, I feel confident that the simulation results on Velocette engines should be close to actual values. In this "study" I have used the original engine specifications to find the performance curves of some engines. Some additional info will also be discussed like: torque- characteristic Volumetric efficiency, influence of exhaust system etc. The following engines models have been subject to modeling and "testing" in GT power.

1. Venom 500 cc, using my own engine to find the necessary input parameters.
2. Viper 350 cc

These models are chosen while the needed input is readily available

Creating an engine model

How do we create an engine model? A good starting point is to draw a sketch of the engine's cylinder head and attach the intake and exhaust system. This sketch will be the basis of building the engine simulation model. It should include all details that are considered important to the flow of air and exhaust. Sketch in Figure 2.

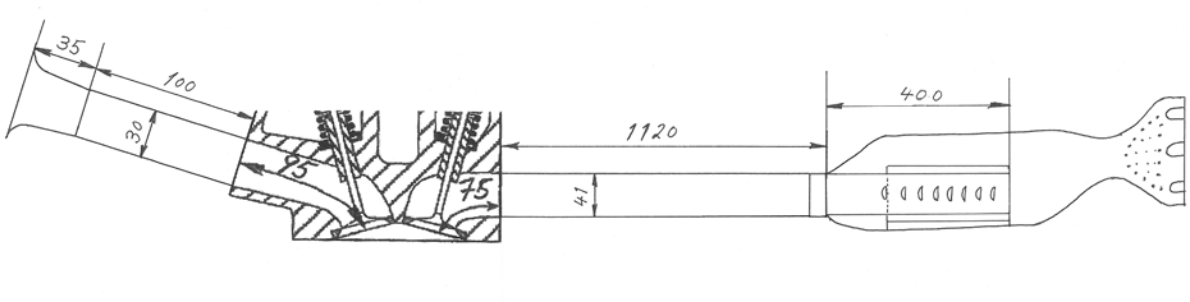

Figure 2. Sketch of 1-cylinder engine model showing actual diameters and lengths

Next - the modeling procedure.
But first, a short introduction of the simulation- tool and the user interface. The engine modeling is object based. All actual engine components (objects) are predefined in library recognizable as small icons. Components like pipes, valves, injectors /carburetors, ports, cylinder, crank etc. are placed on a worksheet by click and drag operations. The components are then linked together by another click-and-drag operation. When this is done we have made the engine layout, but the actual engine /system dimensions still needs to be configured. A double- click on each component gives access to input tables. In the case of a pipe we need to type in pipe dimensions like flow diameter and length, together with a number of other info that I will not go into here. To complete the modeling, this requires input which are also more challenging to obtain, as would be detailed data on valve lift curves, port flow data, cylinder unit geometry, combustion rate etc. The GT-power engine scheme figure 3 below.

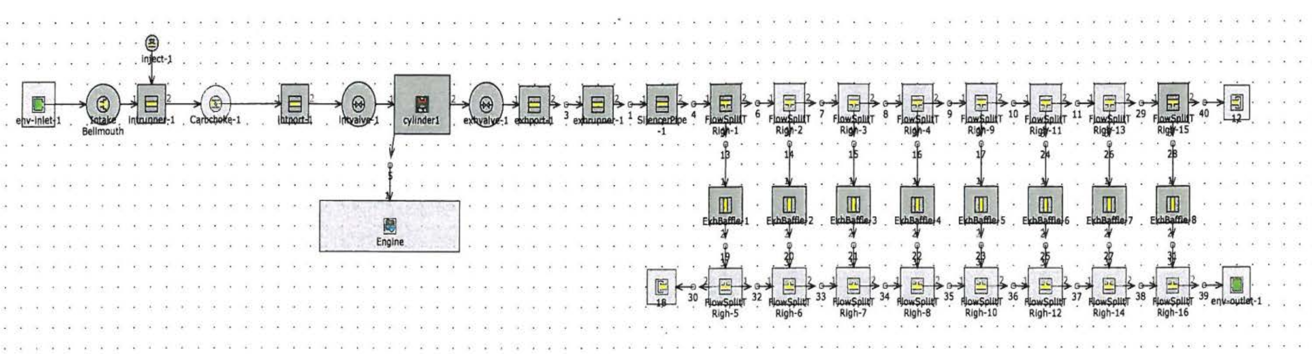

Figure 3 Showing Venom engine model assembly. Air intake to the left, right side boxes represents silencer with its baffle system.

Input data to simulation model

In this chapter we will look at some essential input data that is needed. Table 1 contains the more common specs. of Venom and Viper engine.

Engine specifications	**Venom**	**Viper**
Cylinder Bore [mm]	86	72
Stroke [mm]	86	86
Connecting rod length [mm]	175	175
Compression ratio	8.0	8.5
Intake valve diameter [mm]	46	42.8
Exhaust valve diameter [mm]	42.8	39.6
Carburetor bore [mm]	30 (1-3/16")	27 (1-1/16")
Exhaust pipe internal [mm]	41.3 (OD 1.75" L=44")	35 (OD 1.5"L=44")
Silencer	Fishtail w/baffles	Fishtail w/baffles
Fuel	Petrol	Petrol
Air/Fuel ratio	12.5	12.5

Table 1.

Valve lift curves.

The valve lift curves are recorded by using a micrometer dial indicator on top of the pushrods. Crank angle readings from degree disk on the output main-shaft (with reference to picture - Dai Gibbison "camshaft analysis" in Fishtail). In this case the valve lift curves are recorded every 2.5 degrees.

When putting the lift data into an Excel sheet for plotting, I noticed that the valve lift events curves did not match the factory specifications.

Valve events M 17/8	Factory spec at 0.05" pushrod lift [^{o}CA]	Actual at 0.05" pushrod lift [^{o}CA]	Comments
Exhaust open	65	68	3 degrees advanced
Exhaust close	35	32	"
Intake open	45	48	"
Intake close	55	52	"

Table 2. Recorded valve events are advanced compared to std. specification. Such deviation can be expected, according to Velocette technical literature.

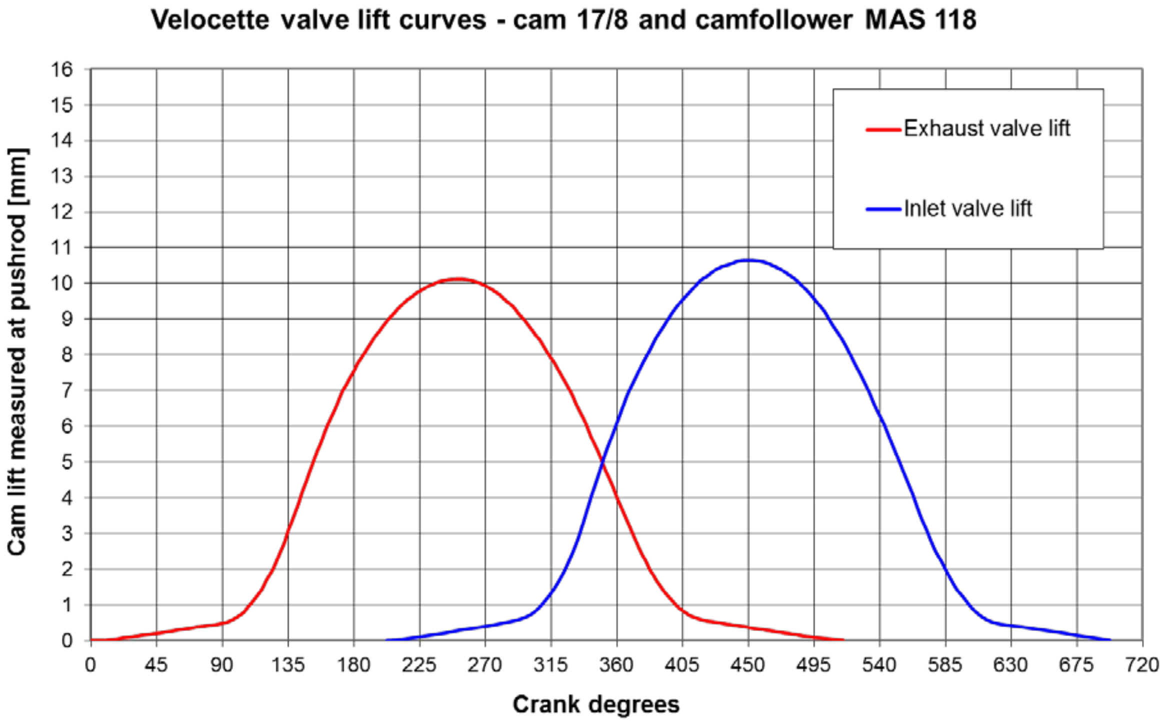

Figure 4. Pushrod lift curves. Actual valve-lift is reduced by actual valve lash. The valve lift events are adjusted according to factory specifications. Note intake valve lift of about 6mm at TDC, which explains the need for deep valve recesses in the piston top.

Port flow

Flow capacity of intake and exhaust ports are another set of important input data. These data were produced by performing flow tests on a standard Venom head. Actual flow was recorded at specific valve lift from 0 to 12mm lift, in step of 2 mm. The flow coefficient is then established by comparing measured flow to the theoretically calculated flow in the valve opening. Much theory and definitions are produced on this subject. A relatively easy to understand definition of flow coefficient, is to compare the flow related to the valve curtain area. Curtain area shown in figure 5.

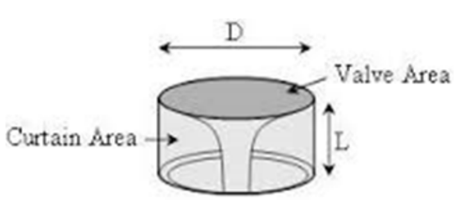

Figure 5. Curtain area represents the theoretical flow area, defined by valve reference diameter and valve lift. This Curtain area definition is used as input to the simulation tool.

The curtain area increases with valve lift, and so do the measured flow. Using this method, the flow coefficient is close to theoretical max. value at lower lift, but drops off as curtain area increases. The drop-off depends on flow restrictions upstream and sometimes downstream of the valve opening.   At lower lift the flow restriction is located at the valve seat, and the measured flow and calculated flow is quite similar given a well-designed valve and seat without flow interfering recesses. At higher lift the upstream port design/capacity starts to make an impact in limiting the flow (combination of small port and big valve - protruding valve guide). The port flow coefficient is thus a combination of port and valve flow coefficient. The measured port flow coefficient is shown in figure 7.

An alternative curtain area definition is shown in figure 6 where area is also a function of valve seat angle.

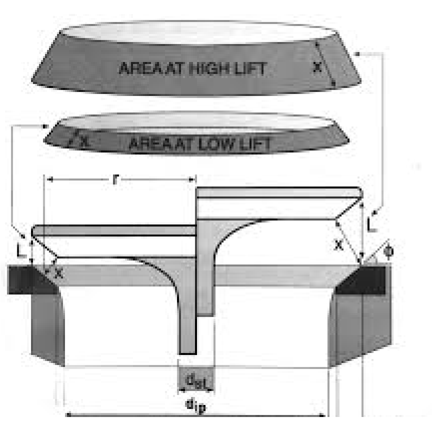

Figure 6. Curtain area defined by valve seat diameter, valve lift and valve seat angle

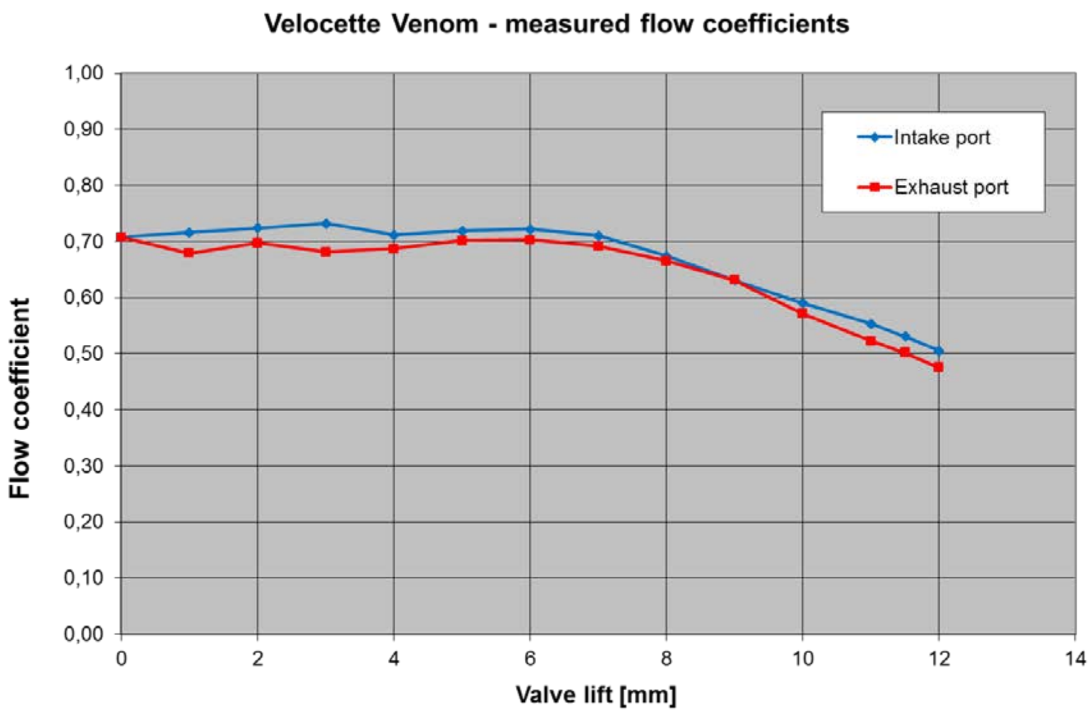

Figure 7. Measured flow is close to theoretical flow at lower lift (Theoretical max. about 0.71 in this case). At higher lift the measured flow is significantly lower that the theoretical value.

Silencer - Fishtail

It must be said that modeling the silencer is not a straightforward job. The Velocette baffle system is a perforated inserted pipe that "leaks" exhaust into a concentric tube with diameter of about 57mm fixed inside the silencer can. This outer tube has a small-hole perforation pattern at the front end and a hole in the rear end of about 10mm. The perforations in the insert pipe (baffle) are spread over a length of about 10 "(figure 8). The outlet of the silencer can is quite big which means that there is no significant backpressure inside the can, even at full load. This means that we can model the silencer as a perforated pipe, without making a significant error regarding the backpressure characteristics of the fishtail. The function of the baffle is to give the engine a better torque curve, compared to more common silencer designs. The baffle outlet flow area is difficult to determine exactly. When the exhaust flow enters the baffles, flow is deflected causing a flow-loss which cannot easily be quantified. To get a feeling for the impact on baffle area, some variation is tested in simulations to arrive at a "reasonable" flow-area which reduces the peak power output about 4 % (1.5 HP).

Figure 8. Velocette fishtail baffle (Source: Grove Classic)

The model is now fully defined and the simulations can be started. The usual run set-up will produce engine performance data at specified engine rpm. To get a power curve the program will calculate engine performance at user defined rpms/rpm steps to cover the speed range of the engine.

So where are the equations, math and solver routines etc.? Luckily for me this is already taken care of by the software designer, and as a user, I have no access to this "department".

After run completion, output files are generated containing all the interesting information on engine performance etc. The results are available for printing plotting etc. by dedicated software.

Results - Venom 500cc

The Venoms power curve peaks at about 6200rpm with 33.9 HP from the engine output shaft. Max. torque 44.3 Nm at 4500 rpm, which corresponds to 32.6 lb-ft.

Compare to power curve in fig 1 showing about 33hp at 6200 rpm.

At 2500 – 3000 rpm the torque is relatively low due to low volumetric efficiency. This is caused by reflecting pressure pulses with flow reversion and inferior scavenging. Due to the valve overlap, the pressure pulses travels through the combustion chamber into the intake port. This backflow is also called "fuel standoff" while fuel-fog is visible outside carburetor air intake. Engines with small overlap, is less affected by reflected pressure waves.

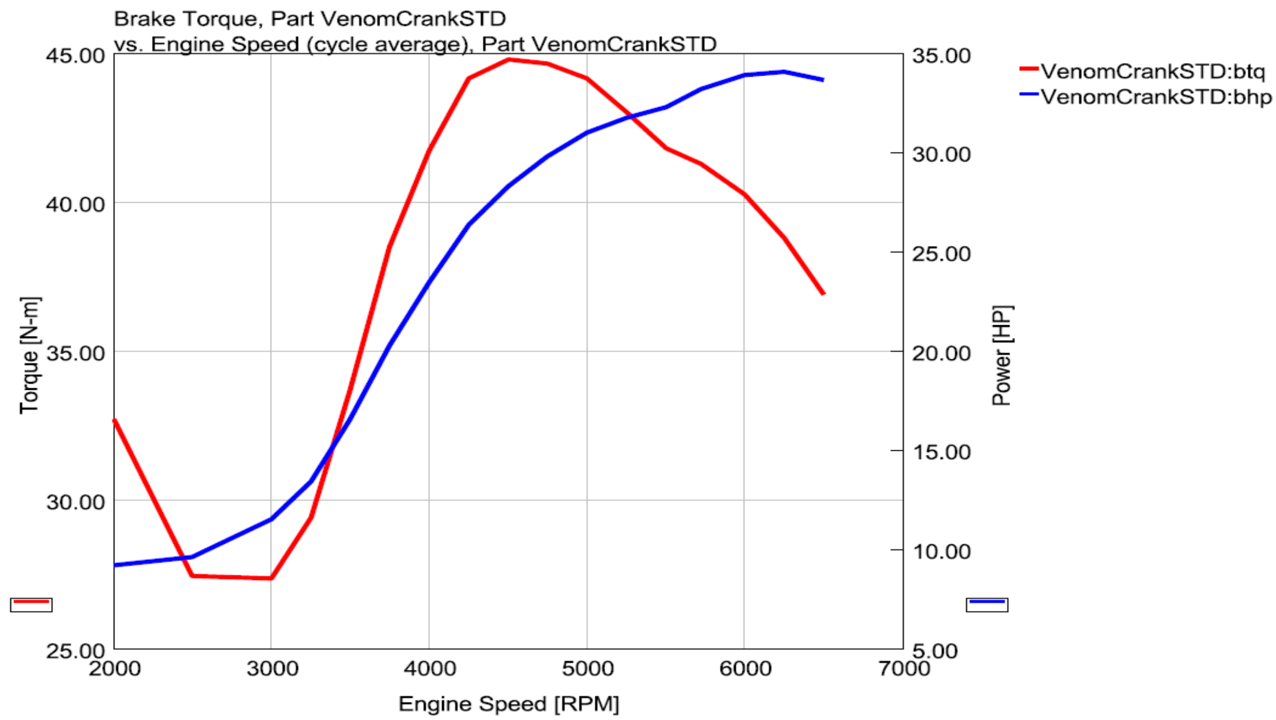

Figure 9. Venom power and torque curve. Note that peak power rpm corresponds with graph in figure 1.

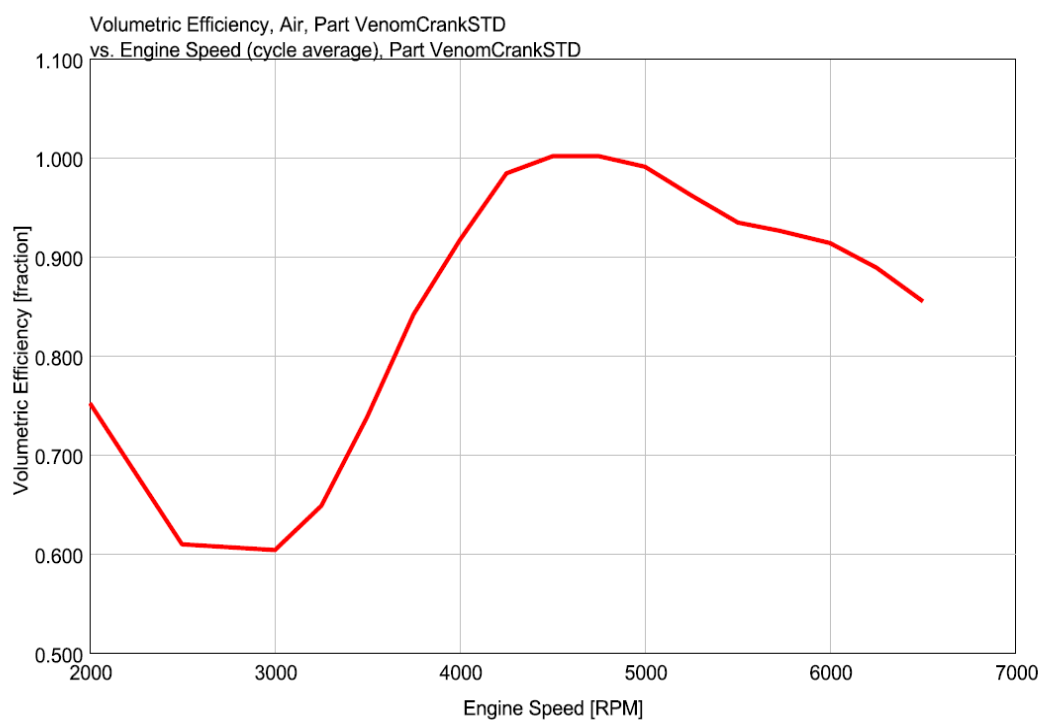

Figure 10 showing volumetric efficiency. Note how the volometric efficiency corresponds with the torque curve.

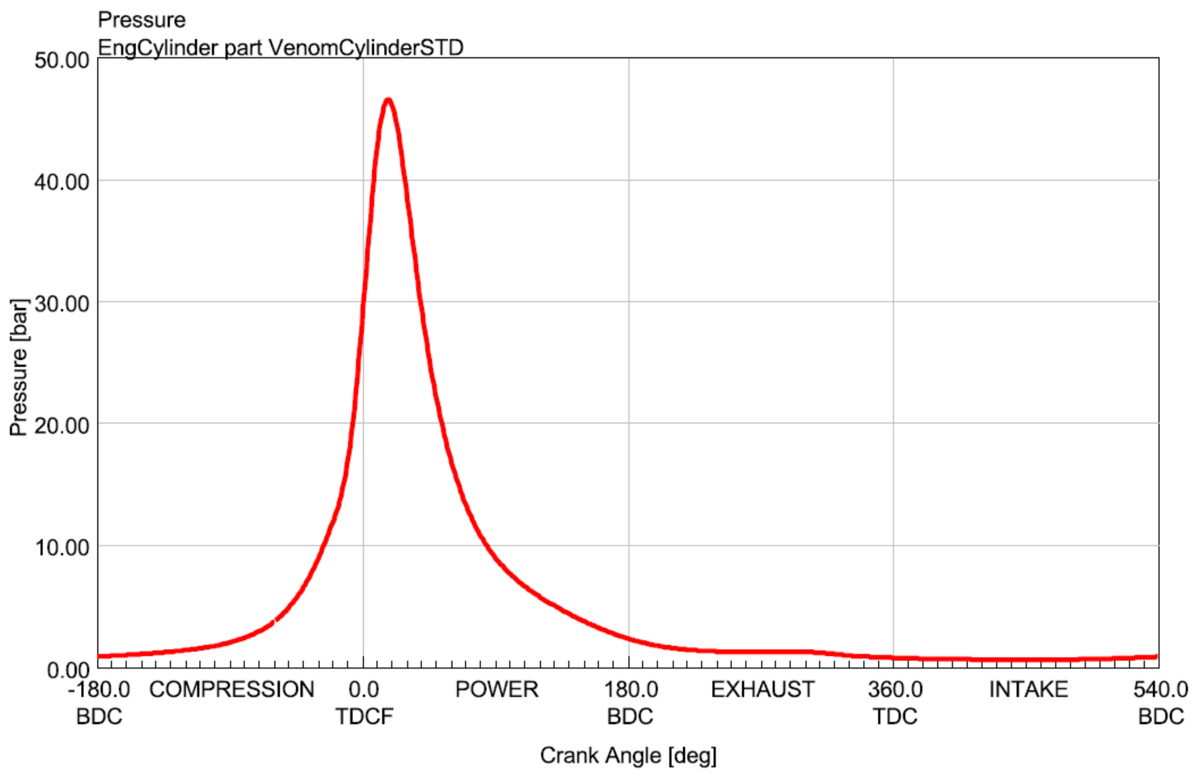

Figure 12. Combustion pressure curve at 4500 rpm

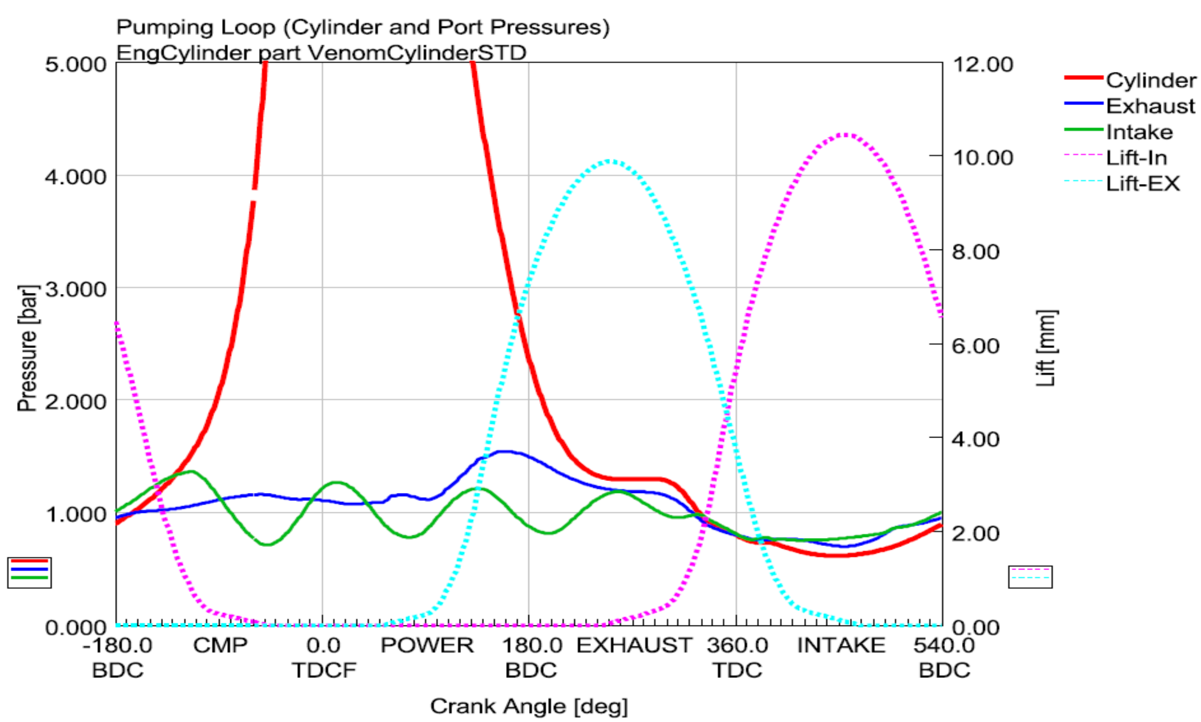

Figure 14. Plot of scavenging loop at 4500rpm.

Note the exhaust pressure (blue line) at valve drops below ambient pressure during valve overlap, which starts induction before the piston begins the intake stroke. This helps increasing volumetric efficiency.

Results - Viper 350cc

The Viper's power curve peak at about 7200 rpm with 26.4 HP. Max torque is 30.8 Nm at 5000 - 5250 rpm, which corresponds to 22.7 lb-ft. Compare to power curve in figure 1 with about 27 HP at about 7000rpm.

At 2500 – 3500 rpm the torque is relatively low due to low volumetric efficiency. This again is caused by reflecting pressure pulses, like in the case of the Venom engine.

Using the same silencer as the Venom, the Viper engine has probably an advantage due to lower flow and thus lower back pressure.

OBS. Viper head is not flow tested but instead Venom flow coefficient is used. This is a relative good assumption but differences might cause some power deviation at high rpm.

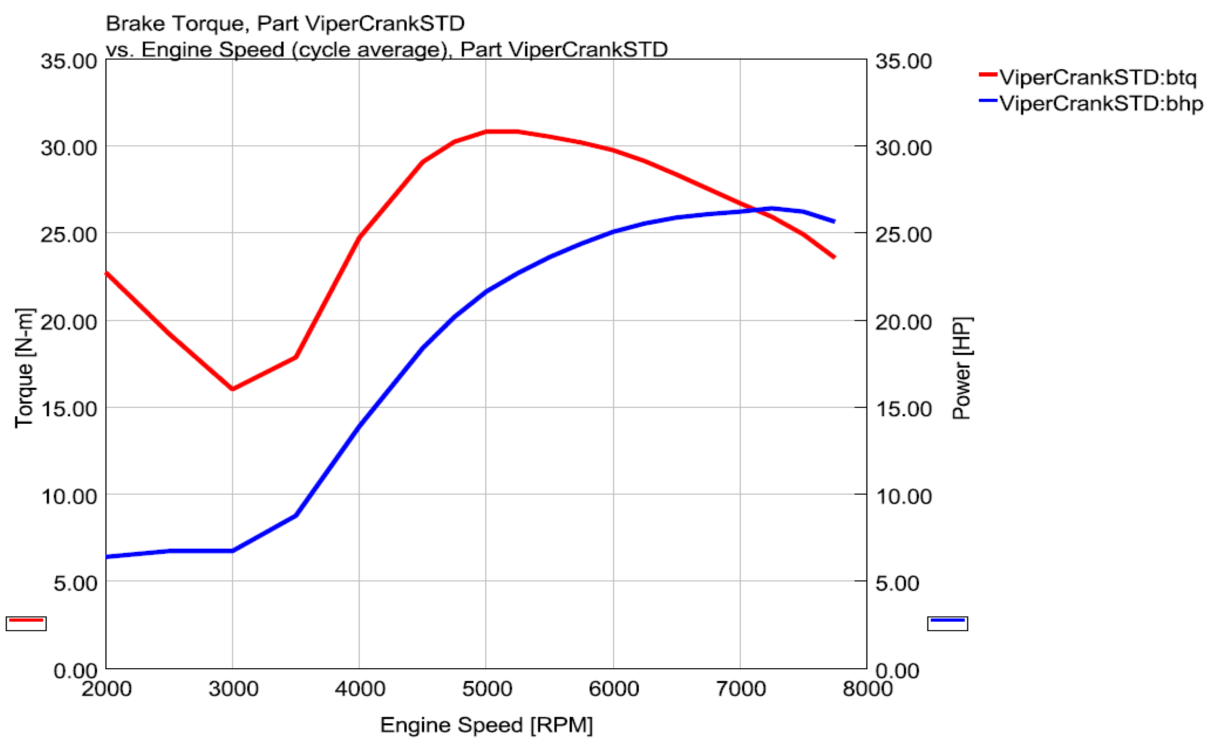

Figure 15. Peak power at about 7200 rpm. The torque-peak comes at about 5250 rpm, 500 rpm higher that the Venom engine.

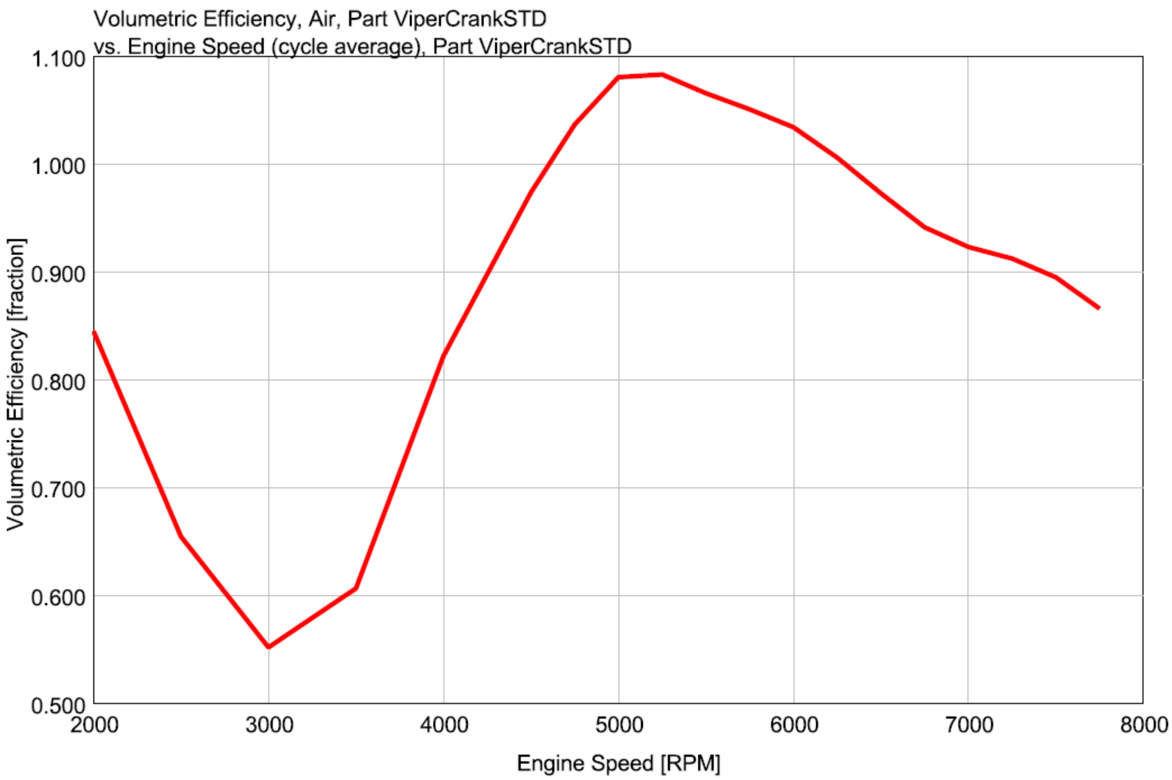

Figure 16. Relatively low volumetric efficiency around 3000 rpm. At revs above 4250 rpm it is higher compared to the venom engine.

Summary -Discussion

The simulation tool show good correlation to measured/published data regarding power output.

The torque curve shows where the engine produces good pulling power. In particular the Viper engine is low on torque at low revs.

Literature recommends not- to over-rev the Venom engine due to risk of valve float. At 7200 rpm the Viper engine is certainly in region of valve float, at least with aged valve springs.

The intention of doing this modeling work is not only to show what we already know about performance output. The established model can now be used to change the engines characteristics, or to optimize the engine using software options.

Task like improving the Viper engines torque curve or increasing the power of the Venom engine, are interesting tasks that I am planning to look into in future work.

Limitations in the simulation model

Some assumptions must be used as input to the simulation. These are typically:

1. Burn-rate
2. Heat losses
3. Friction losses

Some explanation is necessary to get an understanding of these factors.

Burn-rate means the combustion propagation during each combustion cycle. The used burn-rate is realistic, considering the hemispherical combustion chamber design. A short burn rate is beneficial to engine efficiency and knocking margin.

A simplified heat loss model is used in these calculations. A more detailed model might produce slightly different heat loss and thus engine power.

Engine friction is an important parameter. The basis friction model is taken from GT-Power examples. A slight modification is implemented due to difference between sliding bearings and roller/ball bearings in actual engines with less friction.

Dynamic valve lift might be different from static valve lift reading. At higher revs the dynamic valve lift will be different from the recorded lift curves, which has an impact on simulation results, in particular when valve float occurs. Dynamic lift curves are possible to implement, but is not within the scope of this study.

References

1. The Velocette SAGA, Titch Allen
2. Technical Excellence Exemplified, Ivan Rhodes
3. Velocette motorcycles - MSS to Thruxton, Rod Burris
4. Velocette Technical Site, David Gibbison