



University of Wales Trinity Saint David

School of Engineering

Exhibition Paper

*The Effects on Engine Emissions and Performance through the Changes of
Cylinder Head Port Design- An Experimental Investigation Using CAD
Software.*

Christopher Robbins

P146163

B.Sc. Automotive Engineering

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1. Introduction

This major project report considers the design and efficiency of inlet ports in automotive combustion engines (ICE). The effects that port design has on volumetric efficiency, pressure changes and velocity will be evaluated to see the limiting factors

Using Solidworks, a standard port is tested, evaluated and redesigned to increase the volumetric efficiency. The inlet window is a short period with valves only opening to a maximum of 12% of an engine bore diameter. The changes need to benefit the engine at specific increments.

The cylinder head is from the automotive realm that has been designed to meet emission regulation as well as supplying high power outputs at lower revolutions per minute (RPM) to achieve better driveability. The inlet port will be modified slightly to achieve a smoother port with a lower degree of roughness. Research will show port design from the perspective of performance, emissions and driveability.

A flow bench is used to find out the flow of the port. This gives a starting point when comparing simulation data and calculated data. A mold of the inlet port is taken so that it can be replicated in Solidworks for simulation testing and improvements.

1.1 The importance to industry.

Euro emission regulations are becoming stricter and manufactures must find ways to reduce engine emissions.

One alternative is to reduce engine size but a reduction in engine size often gives a reduction in performance.

Ford 1.0-litre EcoBoost petrol engine named 2017 International Engine of the Year “Best Engine Under 1.0-litre” for the sixth year in a row.
Fuel-efficient, compact and powerful engine has won 10 International Engine of Year awards since launch in 2012; powers one in five Ford vehicles sold in Europe
Ford to offer 1.0-litre EcoBoost with innovative cylinder deactivation technology for further reduced CO2 emissions and improved fuel-efficiency from early 2018.

Figure 1: A list of facts about the 1.0-Litre EcoBoost Engine (Media.ford.com, 2018).

1.2 The aims of the project.

- To increase the volumetric efficiency of a cylinder head (Ford Zetec 1.6, 16 valve) by modification of inlet port design.
- To not effect engine emissions, driveability and performance in a negative way when the inlet port is modified, through mathematical calculations, and live and simulated data.

2. Literature review

2.1 Background

A cylinder head ports has changed over the years of the internal combustion engine. This includes materials used such as cast iron and aluminium, and port shapes/design. Different port designs effect the way the air travels through it to the combustion chamber as seen in the diagram (Smith and Wenner, 1977).

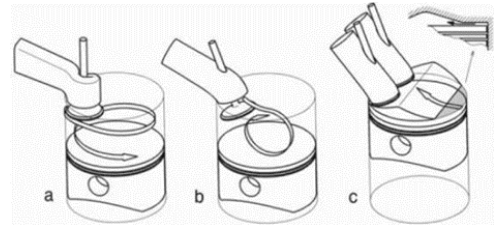


Figure 2: Different ports (Sch\argl, O., Edtmayer, H., Schmidt, S., Leiber, S).

An inlet ports size has a direct correlation to the size of an engine.

This is to maintain flow across the RPM range and to avoid choking (ports too small for engine size) and poor driveability/ low torque (ports too big for engine size) (Vizard, D, 2012).

2.2 Fundamentals of an Inlet Port

An inlet ports task is to guide air through the port to behind the valve or combustion chamber. More prominence has been placed on port design by the recurrence of the direct injection petrol engines (Mohammadebrahim, A., Hannani, S.K. and Shafii, B., 2014).

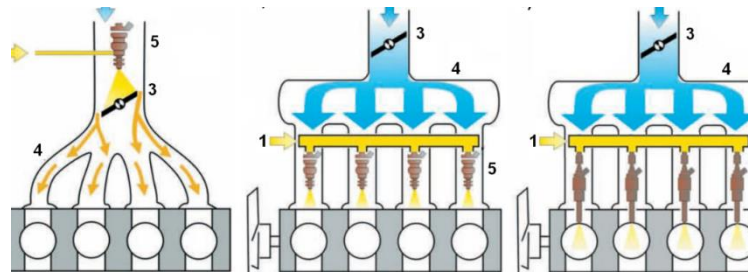


Figure 3: Single point/indirect, multipoint/indirect and direct injection applications (Sendyka and Nog, 2018).

On indirect and carburation type applications the fuel must travel through the inlet port. In the past, with carburation units, inlet ports needed to be rough to create a turbulent air flow which led to better emulsifying of the air and fuel to achieve as close to a stoichiometric ratio. The same principle applies to indirect fuel injection engines dependant on where in the inlet system the injector is positioned i.e. injector position by throttle butterfly or behind the inlet valve (Vizard, D, 2012).

An injector that is placed right behind the intake valve is still indirect but the fuel doesn't travel the length of the port to arrive at its destination. This gives more freedom to port design and to create a smoother surface for high air velocity and less risk of carbon build up.

2.3 Volumetric Efficiency

Volumetric efficiency is the ratio of air that is theoretically capable of being consumed into the cylinders combustion chamber (potential) against the actual air that makes it into the combustion chamber (Smith, L., Fickenscher, T., and Osborne, R- 1999). Restrictions such as valve size, port size, carburettor and throttle body size etc. affect the ratio (Baechtel.J, 2011).

$$VE \% = (\text{Measured CFM} \div \text{Potential CFM}) \times 100$$

(Baechtel.J, 2011)

Most automotive engines achieve a volumetric efficiency is around 70 to 80%, but on higher performance vehicles 80% to the mid 90's is achievable. Highly tuned race engines, for example drag racing engines, regularly surpass 100% volumetric efficiency because of the use of heavily modified intake systems, exhaust scavenging and cam timing (e.g. 130% efficient = air is 30% more dense than that of the air entering the system), (Burgess and Gollan, 2010).

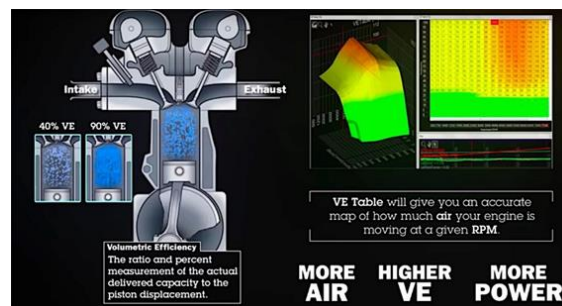


Figure 4: Air and fuel particles (EngineLabs, 2017).

2.4 Methods to increase volumetric efficiency within a cylinder head

2.4.1 Turbocharging

A turbocharger uses the combustion gases to power a turbine on one section of the turbocharger unit (Morotta, J, 2010). The turbine is connected to the compressor wheel via a shaft, sealed at each end, which draws the ambient air in and compresses it (Davis and Perkins-Davis, 2001). As the air is compressed the temperature increases and so the density of the air decreases which is not desired for an engine so the compressed air is directed through an intercooler (Stone, 1989).

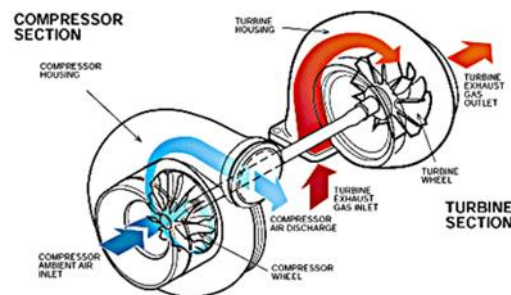


Figure 5: Turbocharger diagram (Brighthub Engineering, 2017).

The air exiting the turbocharger doesn't reach back to ambient temperature but does decrease significantly, thus increasing the density of the inlet air and reducing the temperature. This increases the volumetric efficiency (Charge air after cooling, 2017).

2.4.2 Supercharging

A supercharger requires a direct connection to the engine to power it usually in the form of a belt which takes power from the engine to achieve more power. A supercharger increases the velocity of air travelling to the combustion chamber but as no need for exhaust gases the inlet air is much cooler and thus the use of an intercooler is not needed (Servati, H. and DeLosh, R).

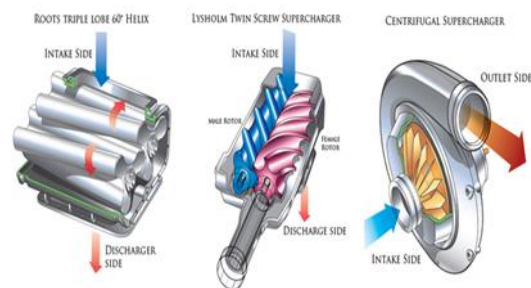


Figure 6: Types of supercharger (Mechanical Booster, 2017).

There are three main types of superchargers from the roots triple lobe helix, twin screw and the centrifugal type. The roots type like the twin screw rotates so that the air is forced into smaller spaces where it is then compressed and exited at a higher velocity. A centrifugal is the most efficient out of the 3 because of the size and load taken on the engine (Dietsche, K. (2011).

2.4.3 Valve timing/overlap

Valve overlap is to assist the exiting process of the exhaust gases by opening the inlet valve early when the exhaust valve is still open. By the upwards movement of the piston and the valve overlap acts as a vacuum which draws more air into the combustion chamber which in turn drives the spent gases out into the exhaust manifold (Baechtel, J, 2011). This process can also help raise the volumetric efficiency by pulling in more air into the combustion chamber. An engine with high lift cams and long cam durations will exaggerate this degree of overlap allowing for more time for the inlet air to enter the combustion chamber (Vizard, D, 2012).

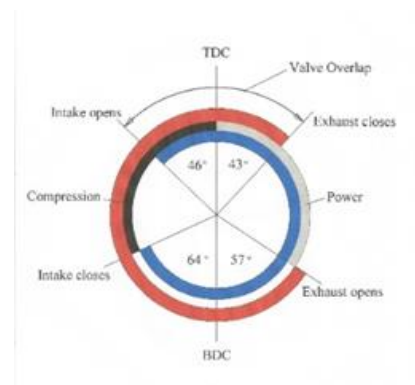


Figure 7: Valve duration (Anon, 2017).

2.4.4 Port design and modification

The use of sequential injection and the placement of an injector closer to the valve eliminates puddling and therefore the rise in emissions (Malkhede, D. and Khalane, H- 2015).

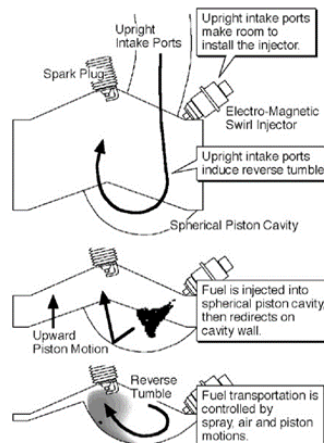


Figure 8: Vertically placed inlet port (Kuwahara, K., Watanabe, T., Tanada, H., Ueda, K-2000).

A technical paper by Mitsubishi shows an actual change in port location instead of design, where the inlet port is orientated vertically instead of horizontally apposed as on most internal combustion engines.

The air entering doesn't need a rough port surface to create turbulence for the air and fuel to mix as the injector is placed internally. This allows a better atomization of the fuel within the air and a better coverage of the combustion chamber (Kuwahara, K., Watanabe, T., Tanada, H., Ueda, K-2000).

A study by Renault found that intake entry, in-line with the cylinder axis, had better performance in relation to the total mass flow rate compared to horizontally opposed ports. As a percentage, the flow was found to be approximately 7 to 9 % and 7.7 to 9.6 % higher on inlet entry (Vashishtha, A., Rathinam, B., Delahaye, L., Ravet-2012).

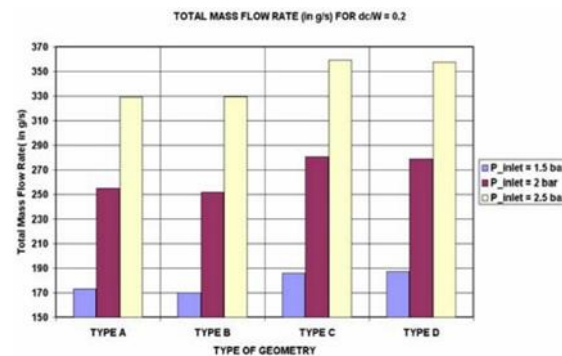


Figure 9: Table of port location (vertical vs horizontal) with relation to clockwise and/or anti-clockwise flow. (Vashishtha, A., Rathinam, B., Delahaye, L., Ravet-2012).

Figure 9 shows that horizontally apposed ports (Type A & B) have a lower mass flow rate than that of vertically positioned ports (Type C & D). The direct of flow i.e. clockwise or anti-clockwise makes no significant difference to the flow. Testing was completed on a 2-stroke, 2-cylinder engine for cost and manufacturing time/cost.

2.5 Draw backs to each method of improving volumetric efficiency

2.5.1 Turbocharging

Turbochargers use exhaust gases to power them, this creates unwanted heat. With inlet air temperature, you want it to be as cold/dense as possible for a better combustion (Power boost technology- 2007).

Additional components and possible internal upgrades makes the cost of this upgrade difficult for some applications Charge air after cooling- 2017).

2.5.2 Supercharging

Superchargers don't necessarily need intercoolers but this doesn't make them a cheaper. The disadvantage for superchargers are that they are run directly from the engine. Superchargers are also usually left for higher output engines where the disadvantage of taking power isn't felt (Tingwall, E- 2017).

2.5.3 Valve timing/overlap

Camshafts allow more air to enter as they may hold the valves open longer this doesn't mean an increase in volumetric efficiency as from turbocharging and supercharging. (Vizard. D, 2012).

2.5.4 Port design and modification

Smoothing out of a port can increase volumetric efficiency but could also hinder engine torque (Chen et al., 1995).

Porting is a very precise task and significant engine power can be lost if the ports shape, size and direction are changed (Baechtel. J, 2011).

2.6 The important of volumetric efficiency in relation to emissions, health issues and legislation.

2.6.1 Emissions

As volumetric efficiency is key to a more efficient combustion which reduces emissions it should be a main part of the manufacturing process of auto-manufactures. Many manufactures have decreased engine size whilst increasing performance to match a bigger capacity engine. This is usually achieved by adding longer stroke bores, turbocharging/supercharging, direct injection and modified ports.

A more efficient combustion reduces the emissions that an internal combustion engine produces e.g. carbon monoxide (CO), unburned hydrocarbons (HC), nitrogen oxides (NO_x).

Port design can help prevent a ‘puddling’ of the fuel, which increases hydrocarbons. Achieving a better ratio between the air reaching the combustion chamber and the potential.

2.6.2 Health Issues

A scientific research found a relation between exhaust emission/waiting in traffic against a higher risk of developing Dementia (Alzheimer's Research UK, 2017). As internal combustion engines create emissions this effects the quality of air which increases the health risk to the planet and its population (Union of Concerned Scientists, 2017).

2.6.3 Legislation

The biggest legislation in the automotive industry is the Euro Emission Regulations which was introduced in 1992 with Euro 1 and is currently in Euro 6. This made it compulsory for vehicles to go through emissions test during MOT testing. This process was aimed at reducing emissions and to make manufactures produce more environmentally friendly cars (See Table 1).

Euro Emission Standard	Start (Year)	Ended (Year)
Euro 1	1992	1995
Euro 2	1995	1999
Euro 3	1999	2005
Euro 4	2005	2008
Euro 5	2008	2014
Euro 6	2014	+

Table 1: Euro Emission Standard and dates.

As new Euro standards were introduced the limit for specific emission gases e.g. Carbon Monoxide, Hydrocarbons and Nitrogen Oxide were significantly reduced so manufactures would have to manufacture more efficient engines.

‘Vehicle emissions are presently tested above the New European Drive Cycle (NEDC) chassis dynamometer process (In-Service performance 6/VI Vehicles, 2015).

2.7 Software

2.7.1 Computational Fluid Dynamics- Solidworks Flow Simulation

A software commonly used to create real life objects into a programme to be improved and adapted. Components can be made from CAD files on a CNC machine for real life testing and/or use.

Solidworks flow simulation package allows for the same testing from a flow bench but with much more variety in features (Solidsolutions.co.uk, 2018).

CFD is used to calculate the mass-flow rates, pressure drops, heat transfer rates, and fluid dynamic forces such as lift, drag and pitching moments for all sorts of components. These can be analysed on an objects external or internal features to see the objects efficiency in different areas (Concept, Product, Design, Engineering, Manufacture, Analysis, & Testing - PES, 2018).

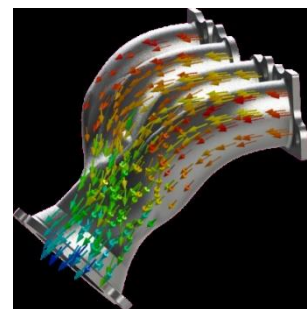


Figure 10: Inlet manifold and CFD testing (Hawkridgesys.com, 2018).

2.8 Hardware

2.8.1 Flow Bench

A flow bench is used to test the flow of passages but is most generally used in testing cylinder head ports. A flow bench draws air through to simulate the intake cycle. This is achieved by creating a pressure difference in the cylinder bore and the external air.

2.8.2 3D Scanner

A 3D scanning is a non-contact technology that captures the form of objects using a laser emitted light (Wright, 2018). They create point clouds of data from the external surface of an object. The data can then be used in different software's (Laserdesign.com, 2018).

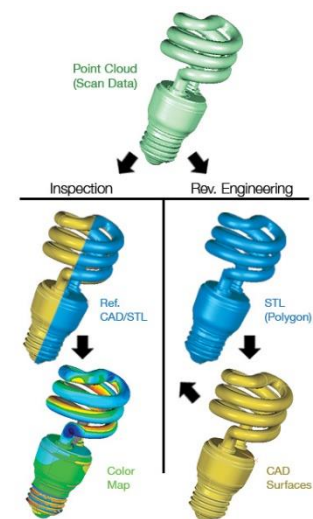


Figure 11: Process from the scan data of a 3D scanner (Laserdesign.com, 2018).

3. Methodology

3.1 Process of inlet port to CAD inlet port

3.1.1 Making a mold of an inlet port

A silicone substance is used with a catalyst substance of the ratio 1:1 and once it has been mixed the solution can be poured into the desired port.

The process usually takes 24hrs but 2 days is preferred as the mold must be completely hardened so the process of scanning can be achieved properly.

Once the mold is set a fine tool is used to free the mold but after a heavy force will removed the mold. The mold needs to be cleaned to be accurately scanned.

3.1.2 Collection of initial flow bench data of the standard inlet port

By using the dimensions of the inlet valve diameter of head and stem the effective valve area is calculated. The valve lift against the valve diameter is used to calculate valve lift ratio.

Two bolts are mounted to the topside of the cylinder head to open the valves. Two DTI gauges are used to measure the amount of valve lift. The valves are opened by the set amount calculated from the valve dimensions.

With the valves closed and the Flow-bench on range 1 the machine can be turned on (must be on inlet range to create air flow through inlet track and past valves and not on exhaust). The machine needs to be taking up to 25” (60cm) of H₂O for testing purposes and to check initially that there are no leaks.

3.1.3 Scanning the inlet mold to acquire x, y, z point cloud data

The mold needed to be whitened to avoid the transparency of the silicone disturbing the scan data collected. Reference marker dots were used over the mold in red marker to allow the scanner to pick up specific points. The data needs to be converted into a mesh with the Scan-3D tool so a solid body can be made.

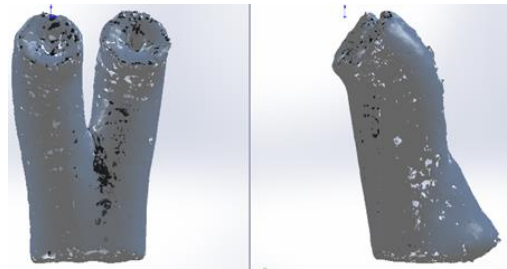


Figure 12: Point cloud data converted into a mesh.

3.1.4 Processing the mesh into a solid working body within Solidworks

By using multiple planes along the mesh surface, the perimeter can be sketched and lofted into a solid body to achieve the picture to the right.

The higher complexity of the mesh allows for more planes and a more accurate representation.

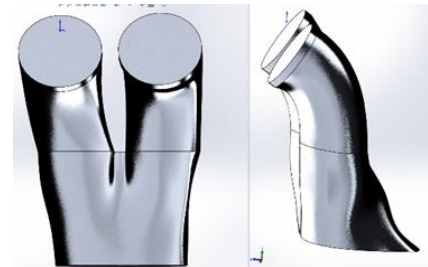


Figure 13: The inlet port as shown as a solid-body within Solidworks after the mesh has been lofted.

3.1.5 Creating cylinder head with inlet port, valves and valve seats

A cylinder head made from the dimensions of the port was made then the port was cut from the cylinder head using the 'Cavity' tool to create the port in the cylinder head.

The valves were made based on measurements taken from the ones in the cylinder head testing and using a 'revolve' tool.

The cylinder bore was made from the dimensions of the engine with a bore of 79mm and a stroke of 81.9mm.

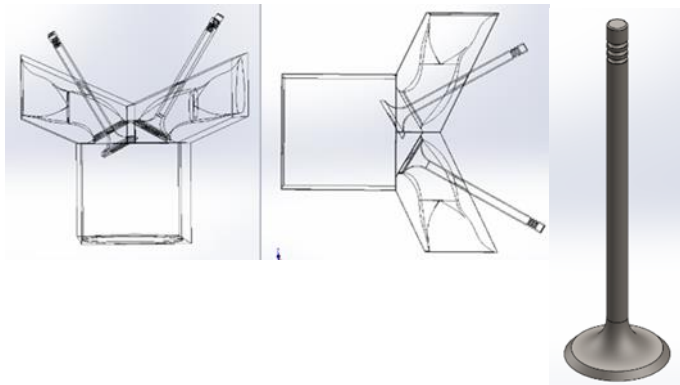


Figure 15: Drawing view of the cylinder head with valves and bore. A picture of the valve created in Solidworks.

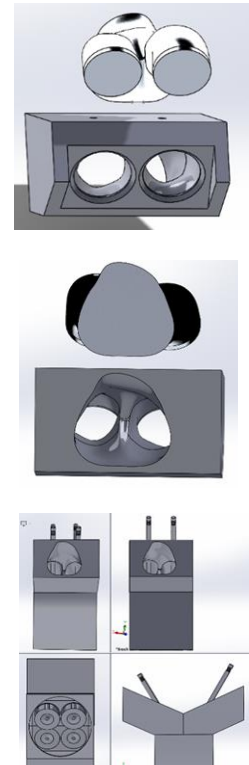


Figure 14: Showing the process of using the mould as a cavity in a ready-made cylinder head to create the port.

3.2 Process of collecting initial simulated data

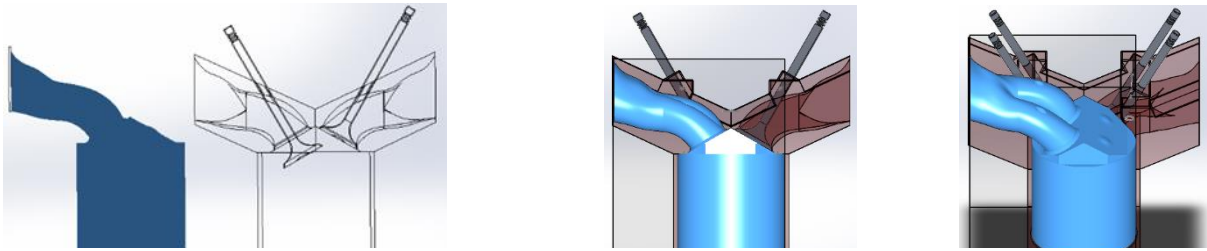


Figure 16: These pictures show the fluid volume of the inlet track and into the cylinders bore. The inlet valves can be seen to be open allowing the transfer of fluid into the bore.

Lids are used to seal the flow area and boundary conditions are set at environmental pressure at Inlet and 25 Inches of H₂O less in pressure at the cylinder. This creates a pressure drop to achieve the flow to obtain the results.

Goals are set to achieve comparable data (Volume flow rate at each lid).

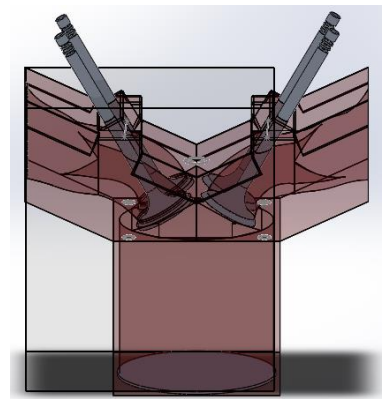


Figure 17: Flow area of inlet port and cylinder bore with valves open at 2mm of lift (showing computational domain).

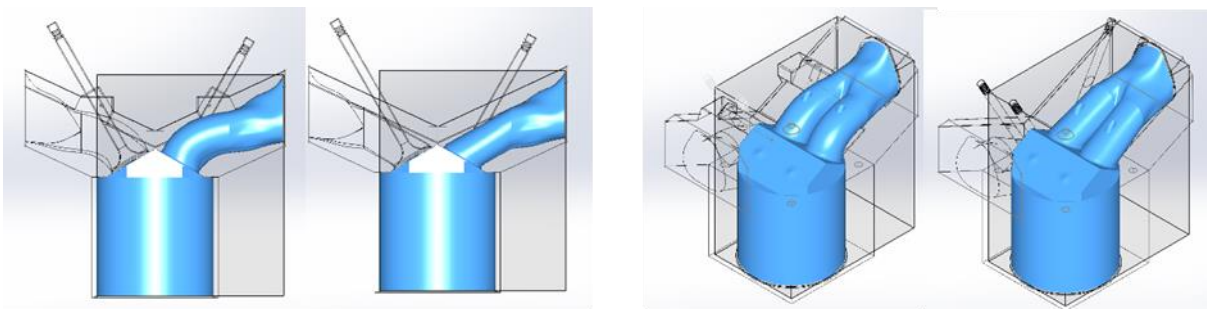


Figure 18: Side and angled view of the standard port and modified port with its fluid volume seen.

3.3 Differences between the standard and two modified ports

3.3.1 Modified Port 1- Diameter at port split and at valve seat

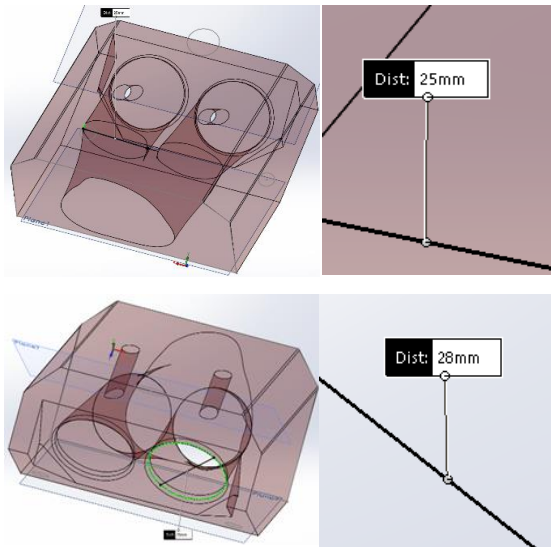


Figure 19: Port split diameter and inner diameter of the valve seat for the first modified port.

The 1st modified port doesn't incorporate a change in diameter but has been changed to obtain a straight port at the port split to the valve seat.

This modification was performed to try and see if the straight track of intake port increased the flow rate.

3.3.2 Modified Port 2- Diameter at port split and at valve seat

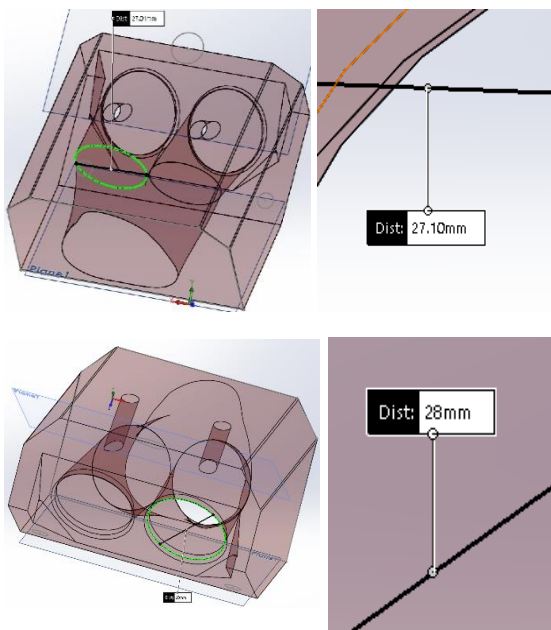


Figure 20: Port split diameter and inner diameter of the valve seat for the second modified port.

The 2nd modification adds 2mm in diameter at the port split. This achieves a constant port diameter down each port split.

The design was incorporated from the first modification with only a change in diameter.

4. Discussion

4.1 Live data

4.1.1 Valve calculations

To calculate valve effective area, see below.

	Valve Head Diameter (mm)	Valve Stem Diameter (mm)	DH Valve Head Diameter (in)	DS Valve Stem Diameter (in)	Effective Valve Area (in ²): (0.785(DH ² -DS ²))x2
Inlet Valve	30	6	1.1811	0.24	2.1

Table 2: Valve dimensions and effective valve area of the inlet valve.

Valve Lift x Diameter (LXD)	0.05	0.1	0.15	0.2	0.25	0.3	0.35	0.4
Inlet Valve Lift (mm)	1.5	3	4.5	6	7.5	9	10.5	12

Table 3: Valve lift calculated for the percentage of valve diameter.

The dimensions above show the amount of valve lift can be calculated for the length and diameter of the valve.

Valves only open to around 12% that of the cylinder bore diameter i.e. (79mm/100) x12= 9.48mm maximum valve lift.

4.1.2 Live flow bench data

As the valve lift increases so does the velocity of air and in most cases the range must be changed to obtain the correct flow-scale. Therefore, the range numbers are needed to achieve the correct CFM data for each valve lift.

Range	1	2	3	4	5	6
Intake	34.98	70.98	147.8	293	440.5	592.5

Table 4: The flow-bench ranges with the conversion numbers.

Test No.	1	2	3	4	5	6	7	8
1. Pressure	25	25	25	25	25	25	25	25
2. Valve Lift	1.5	3	4.5	6	7.5	9	10.5	12
3. Flow Range	2	3	3	4	4	4	4	4
4. Flowscale (%)	55	52.5	74.2	45	49	51	52	53
5. Flowscale/100	0.55	0.525	0.742	0.45	0.49	0.51	0.52	0.53
6. Raw Test Flow	39.04	77.58	109.6	131.8	143.6	149.4	152.4	155.3
7. Leakage	0	0	0	0	0	0	0	0
8. Test CFM	39.04	77.58	109.6	131.8	143.6	149.4	152.4	155.3
9. Test CFM/Sqin	18.59	36.94	52.2	62.78	68.36	71.15	72.54	73.94
10. Potential Flow	25.3	47.3	65.7	80.2	91.1	98.2	101.5	101.1
11. Potential Flow (%)	0.73	0.78	0.79	0.78	0.75	0.72	0.71	0.73
12. Max Flow	0.742							
13. Average Flow	0.54							

Table 5: Data collected from the flow-bench testing.

Increase in CFM should flatten out at around 7.5mm- 9.48mm of valve lift as this is where the port is more restrictive than the valve.

Valve Lift (mm)	Live CFM/ sqin	Potential CFM/ sqin
1.5	18.59	25.3
3	36.94	47.3
4.5	52.2	65.7
6	62.78	80.2
7.5	68.36	91.1
9	71.15	98.2
10.5	72.54	101.5
12	73.94	101.1

The live and potential flow readings are similar as the valves are only slightly open and restrictive but as the valve lifts increase the difference can be seen to increase until at 9mm of lift.

Table 6: Live and Potential CFM readings.

The difference in flow between 6mm of lift and 12mm of lift is only 23.44 CFM but between 1.5mm of lift and 6mm is 92.8 CFM which is almost 4 times in difference. The Zetec head flows 4 times quicker at low valve lift compared to high valve.

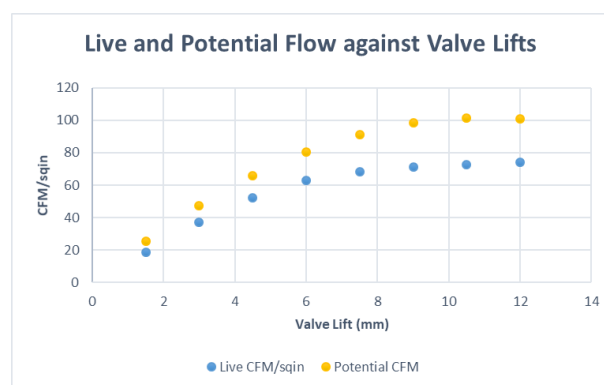


Figure 21: Live flow bench data and potential flow data (CFM/sqin) against valve lifts.

4.2 Simulation data

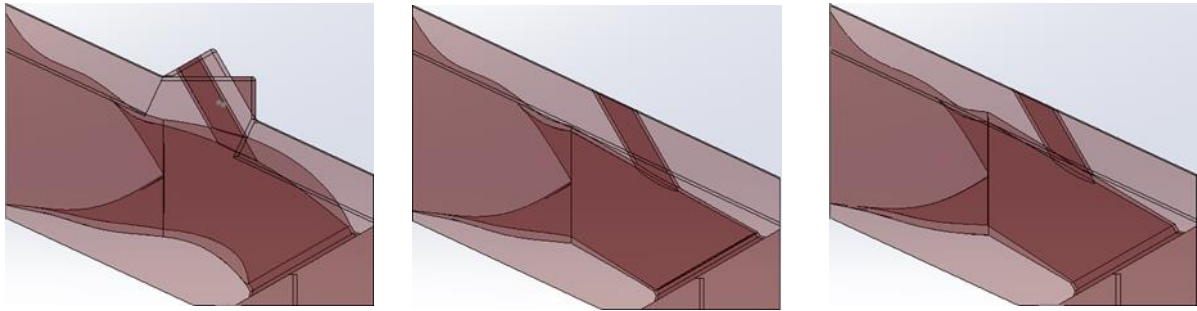


Figure 22: Three designs of the inlet port- Standard port, straight port (no change in size) and straight port with an additional 2mm in diameter.

Modification 1- This modification was undertaken to see if port design affected CFM, Velocity and Pressure without a change in port diameter. The change was made on the assumption that the air entering would be taken a shorter, less resistant path that would slow the intake air less than the standard port.

Modification 2- The modification was decided on when the CFM readings and Velocity readings of the first modification came back positive with an increase in all. It was decided to see how a change in diameter in the intake track would affected CFM, Velocity and Pressure. *See Methodology for an in-depth explanation of the changes.*

4.2.1 Original Simulated Port CFM against Potential CFM

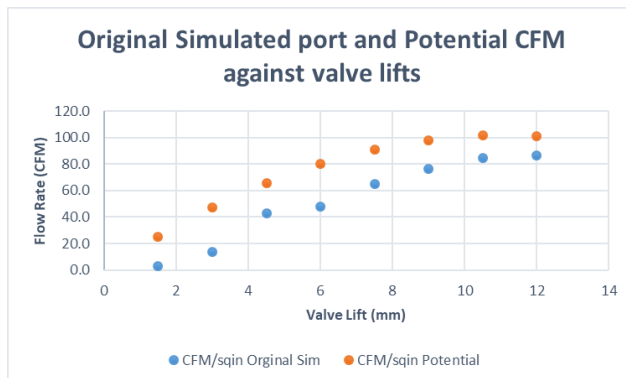


Figure 23: Data plotted of original simulated port and potential CFM against valve lifts.

	CFM/sqin	
	Original Sim	Potential
1.5	2.6	25.3
3	13.4	47.3
4.5	42.8	65.7
6	48.0	80.2
7.5	65.2	91.1
9	76.4	98.2
10.5	84.8	101.5
12	86.3	101.1

Table 7: CFM readings for original simulated port and potential against valve lifts.

The CFM increases in flow and begins to flatten around 10.5mm of valve lift, which gives it a higher volumetric efficiency but only at very high valve lifts. See 4.3.2 for volumetric efficiency comparison.

There is still potential between the live and simulated data. Both data readings flatten out and this is due to the pressure difference in the port and bore.

A bigger port will increase CFM readings but velocity is affected. Increasing port size too much can lose an engines low RPM torque meaning that higher RPM's are needed to achieve maximum power and torque.

4.2.2 1st Modified Simulated Port CFM against Potential CFM

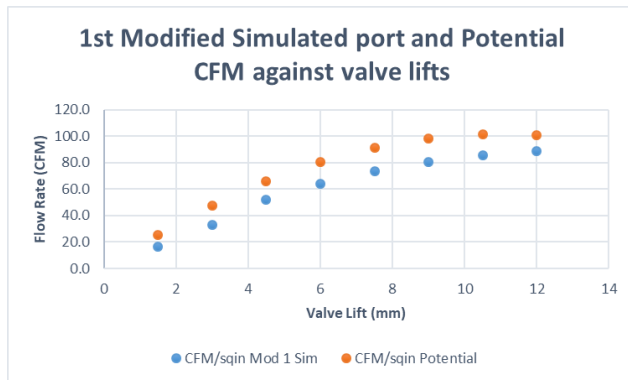


Figure 24: Graph showing CFM readings for the first modified port in the simulation software and potential against valve lifts.

	CFM/sqin	
	Mod 1 Sim	Potential
1.5	16.3	25.3
3	33.2	47.3
4.5	51.8	65.7
6	64.3	80.2
7.5	73.3	91.1
9	80.2	98.2
10.5	85.6	101.5
12	88.6	101.1

Table 8: CFM readings for first modified simulated port and potential against valve lifts.

A straighter port increases the CFM output throughout valve lifts to be closer to the potential CFM. The pressure and velocity plots will be compared further into the discussion to show how port design and size can affect these variables.

4.2.3 2nd Modified Simulated Port CFM against Potential CFM

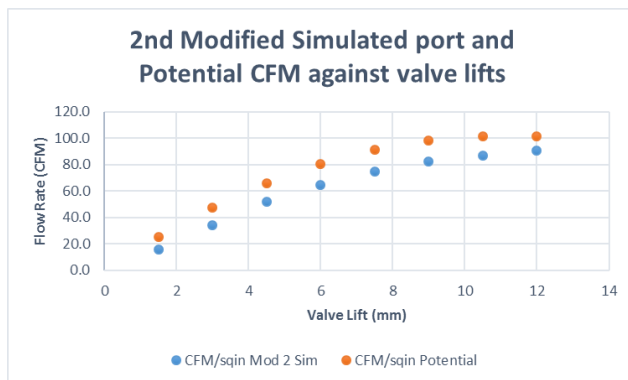


Figure 25: Graph showing CFM readings for the second modified port in the simulation software and potential against valve lifts.

	CFM/sqin	
	Mod 2 Sim	Potential
1.5	15.8	25.3
3	34.2	47.3
4.5	51.7	65.7
6	64.7	80.2
7.5	74.7	91.1
9	82.3	98.2
10.5	86.6	101.5
12	90.8	101.1

Table 9: CFM readings for first modified simulated port and potential against valve lifts.

The 2nd modified port increases CFM from the original but very slightly less until over 7.5mm of lift is achieved.

This due to the valve being restrictive up until this point in lift.

The reason for the simulated data to take longer to flatten out at higher lifts can be placed down to the limitations of the software, mesh reliability and surface roughness specification (Chen et al., 1995).

4.2.4 Comparison between the standard port, 1st modified port, 2nd modified port and potential CFM

The standard, 1st modified and 2nd modified ports all are below the potential flow but the two modified have an improved CFM.

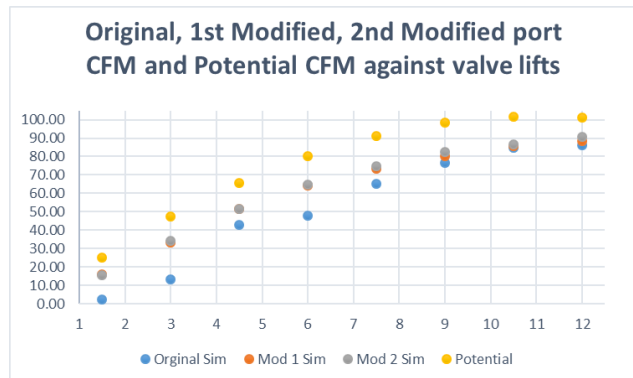


Figure 26: Graph showing CFM results of the standard port, 1st modified port, 2nd modified port and potential against valve lifts.

The 2nd modification isn't justified as an increase in flow occurs above 7.5mm of lift compared to the 1st modification. Its only at an unrealistic valve lift that they differ slightly with the 1st modified achieving 88.59CFM/sqin and the 2nd modified achieving 90.81CFM/sqin.

The calculation that the valve is the restrictive part until 4/D (quarter of its diameter of valve lift), (30/4= 7.5mm of valve lift). At 7.5mm of lift and above this is where the flow readings start to change as the port becomes the restrictive part and the slightly bigger port on the 2nd modified port has an increase flow rate.

	CFM			
	Orginal Sim	Mod 1 Sim	Mod 2 Sim	Potential
1.5	2.62	16.35	15.84	25.3
3	13.42	33.20	34.20	47.3
4.5	42.78	51.76	51.66	65.7
6	48.03	64.27	64.68	80.2
7.5	65.18	73.25	74.67	91.1
9	76.38	80.21	82.33	98.2
10.5	84.76	85.56	86.57	101.5
12	86.27	88.59	90.81	101.1

Table 10: Table showing CFM results of the standard port, 1st modified port, 2nd modified port and potential against valve lifts.

4.2.5 Original Simulated Port Pressure and Velocity trace

Velocity trace and pressure trace for 1.5mm-12mm of valve lift in order of appearance.

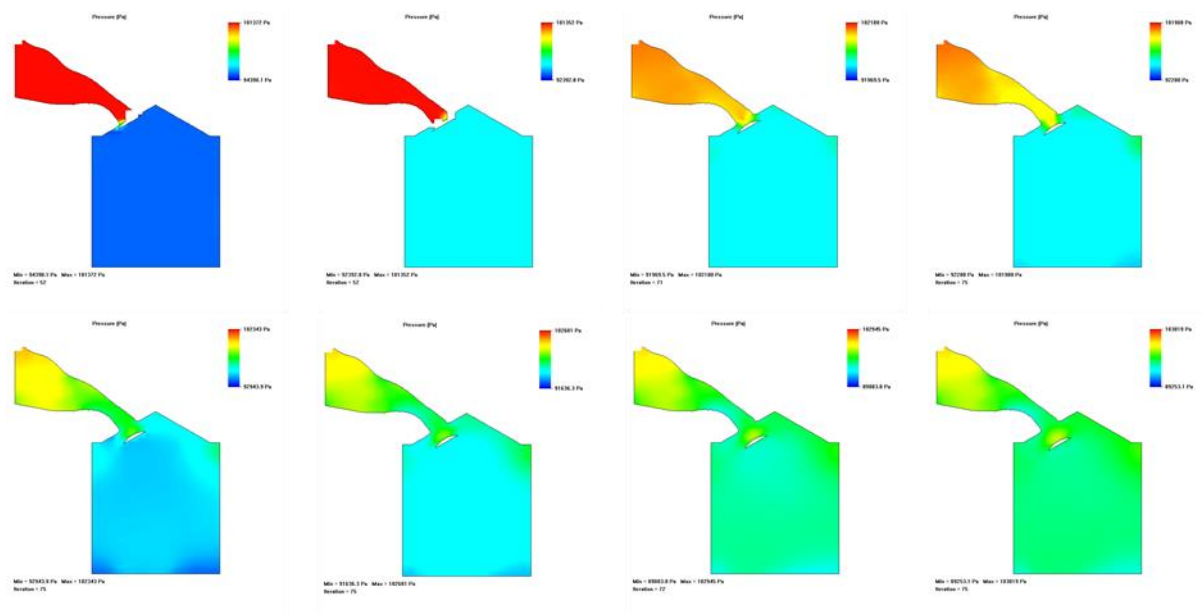


Figure 27: Pressure change at each valve lift (original port).

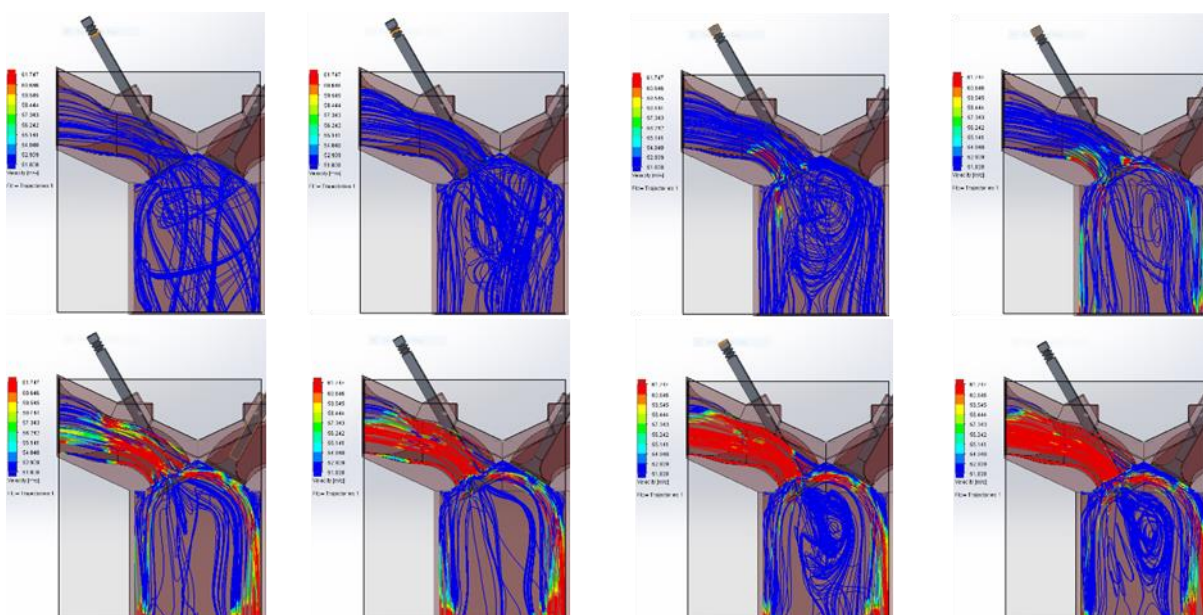


Figure 28: Velocity change at each valve lift (original port).

4.2.6 1st modified Simulated Port Pressure and Velocity trace

Velocity trace and pressure trace for 1.5mm-12mm of valve lift in order of appearance.

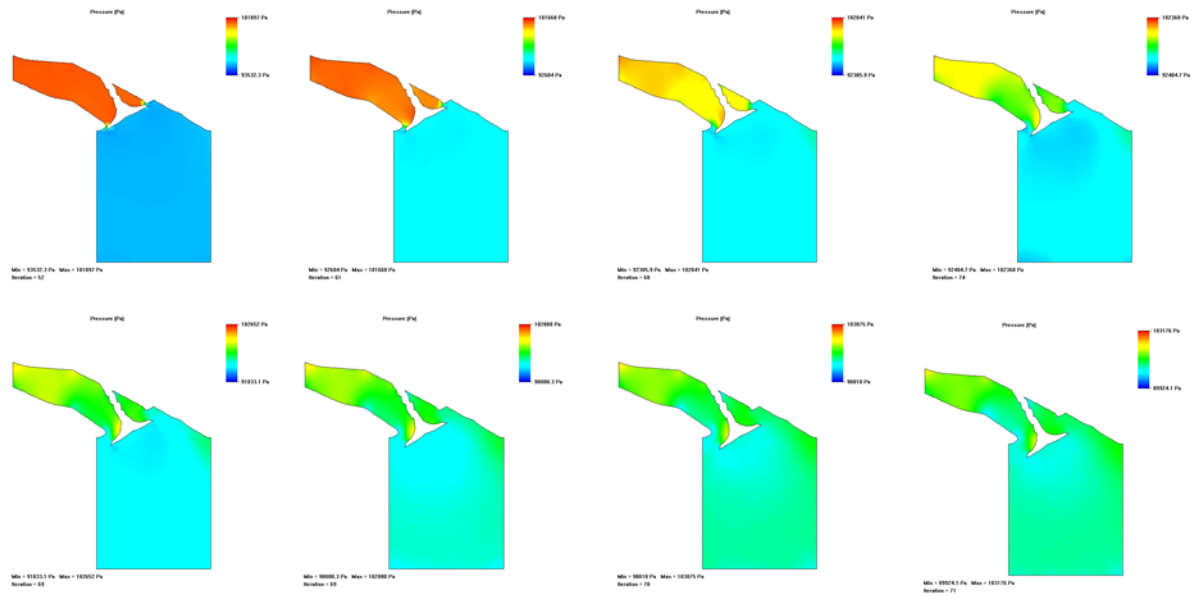


Figure 30: Pressure change at each valve lift (1st modified).

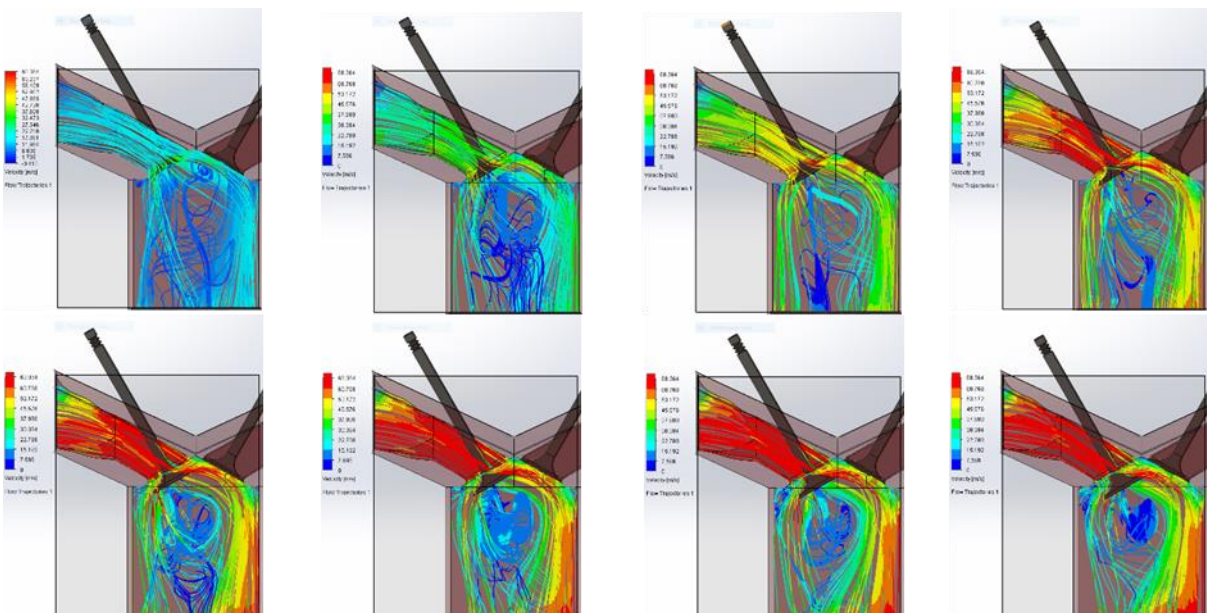


Figure 29: Velocity change at each valve lift (1st modified).

4.2.7 ^{2nd} modified Simulated Port Pressure and Velocity trace

Velocity trace and pressure trace for 1.5mm-12mm of valve lift in order of appearance.

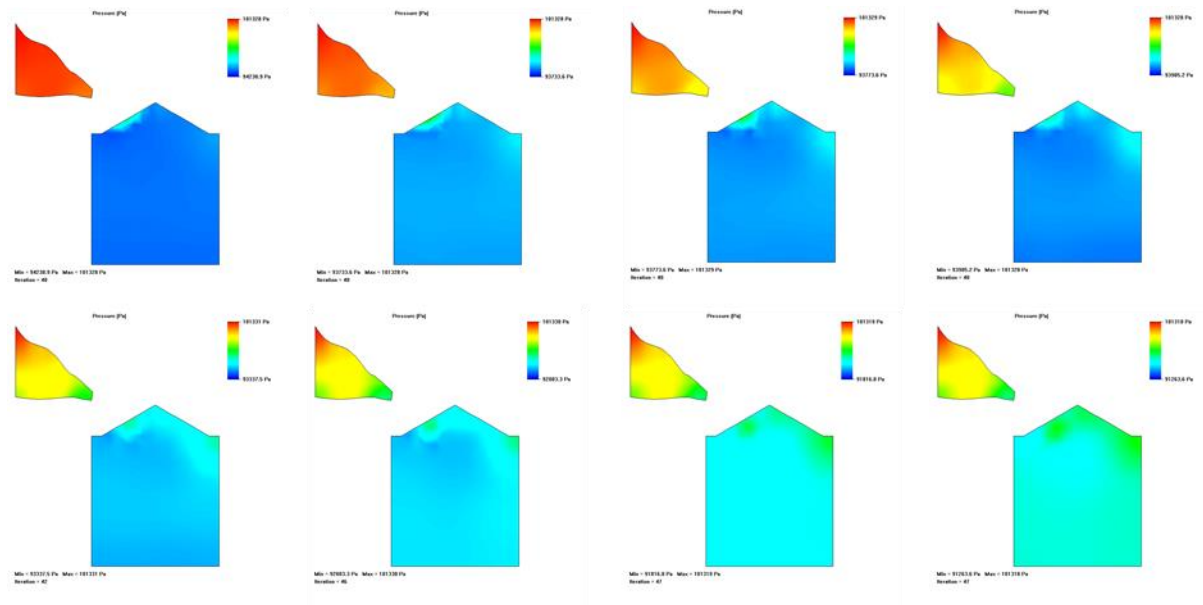


Figure 31: Pressure change at each valve lift (^{2nd} modified).

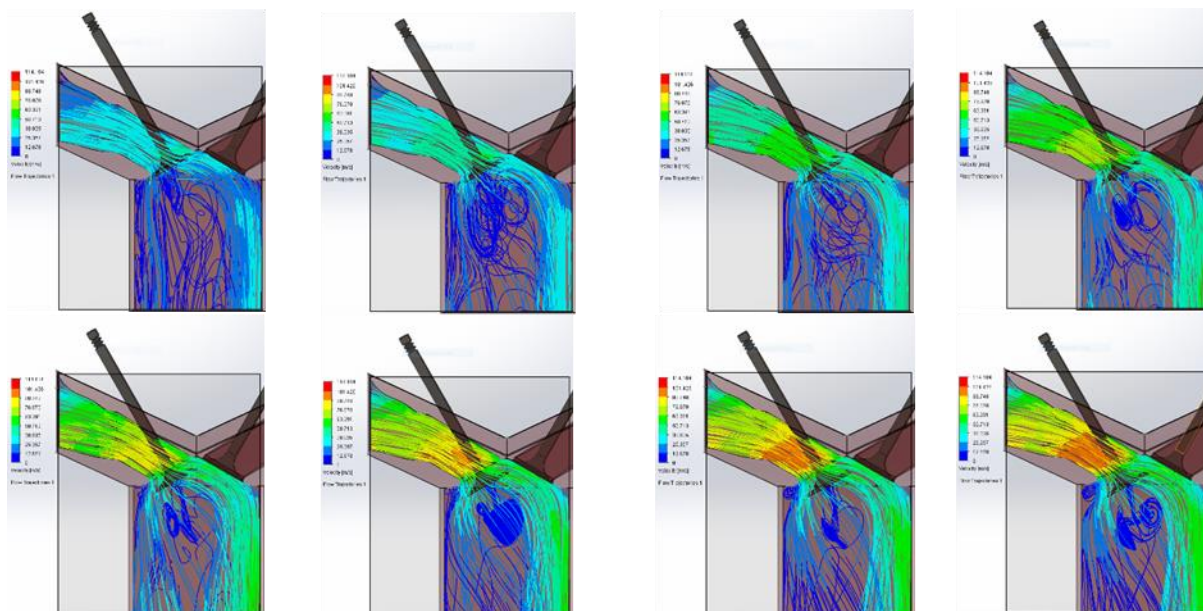


Figure 32: Velocity change at each valve lift (^{2nd} modified).

4.2.8 Analysis and Evaluation of Velocity and Pressure traces for the standard, 1st modified and 2nd modified inlet ports.

The 1st modification shows a quicker drop in high pressure in the inlet and a quicker rise in pressure in the cylinder, during valve lifts, showing the difference in efficiency that the 1st port design has over the original. The difference in pressure averages out the quickest in the 1st modified port than the other two.

The 2nd modification had a similar CFM reading to the 1st modification but a slower pressure change throughout the valves lift. The velocity traces do not show as high readings even though the CFM is the same as the 1st modification.

The standard port has a high velocity (red) at around 4.5mm but is only little compared to the low velocity (blue) around this valve lift. The 1st modified port shows high velocity readings (red) at 4.5mm of valve lift but much more of an average velocity readings (green) than the standard port. The 1st modified port has an increase in CFM and velocity compared to the standard port but the 2nd modified port doesn't show an increase in velocity to high velocity (red) but has a more consistent mid-range velocity (green).

The 1st modified port shows the best in simulation data with higher CFM readings without a negative effect to the ports velocity. The constant increase in velocity would show that torque wouldn't be affected in a negative way at low and high RPM levels.

An increase in port diameter of 2mm in the 2nd modified port is not needed as it affects velocity negatively and only increases CFM at high valve lifts.

CFM may be improved with an increase in valve size. *See next section for a brief look at an increase in valve size by 2mm (30mm-32mm)*

4.2.9 CFM, Velocity and Pressure trace for a 2mm increase in valve diameter.

The bigger valves achieve a higher CFM reading than all the data but only at very low valve lifts.

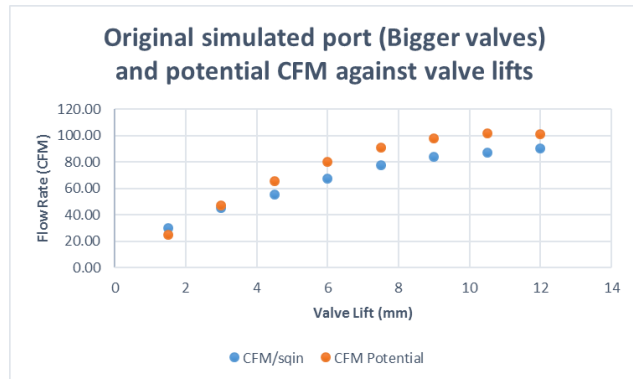


Figure 33: The original simulated port with a 2mm increase in valve diameter (32mm) and potential CFM against valve lifts.

See section 4.4 to see a comparative between all the modifications and the next section for the velocity and pressure traces.

	CFM/sqin	
	Original Sim (bigger valves)	CFM Potential
1.5	29.77	25.30
3	45.51	47.30
4.5	55.60	65.70
6	67.20	80.20
7.5	77.39	91.10
9	84.15	98.20
10.5	86.87	101.50
12	90.41	101.10

Table 11: CFM readings for each valve lift for the simulated port with an increase in valve size and potential CFM.

4.2.10 Analysis and Evaluation of Velocity and Pressure traces for the standard port with the increase in valve diameter (30mm-32mm).

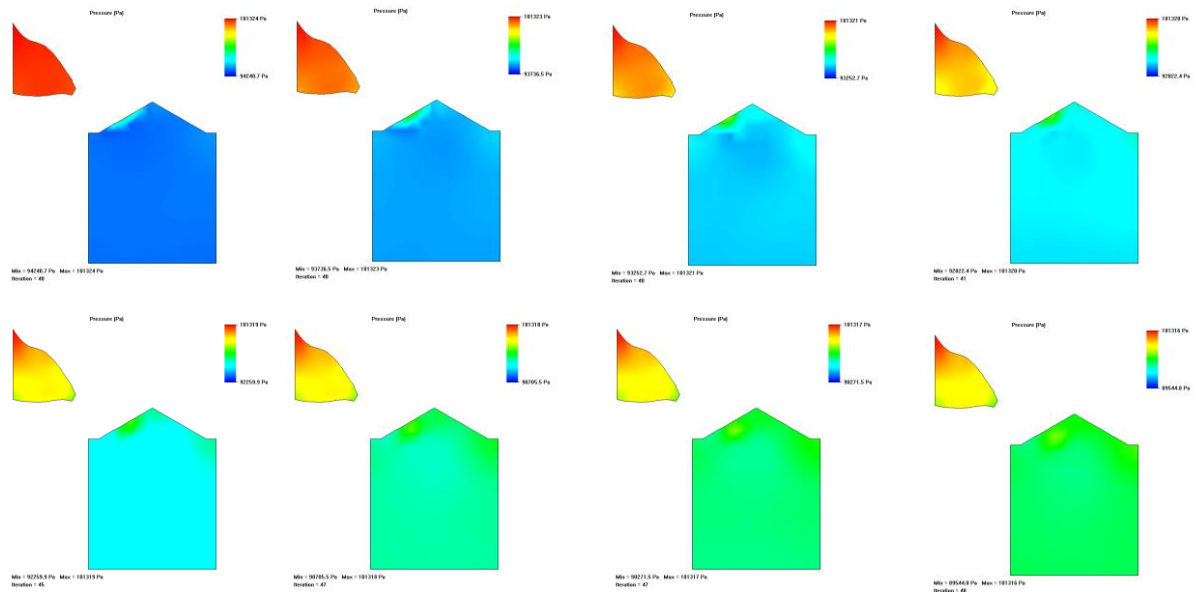


Figure 34: Pressure change at each valve lift (bigger valves).

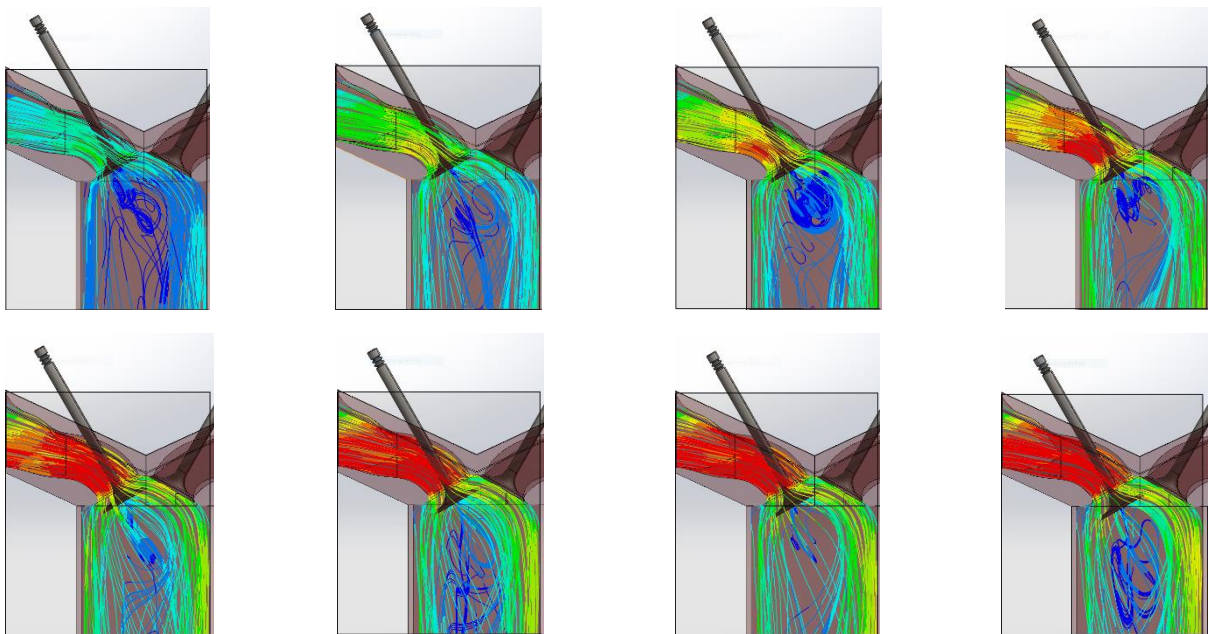


Figure 35: Velocity change at each valve lift (bigger valves).

Theoretically, bigger valves should increase CFM but the torque and power is more likely to be higher up in the RPM range. The CFM has increased without a downfall in air velocity. The pressure change also is less drastic which is the key to maintaining high velocity.

There are many more variables to consider when looking at a complete well performing engine compared to a well performing cylinder head. As said before CFM isnt the end all of cylinder head performance and the velocity and pressure needs to be noted through valve lifts to achieve a well performing cylinder head (Chen et al., 1995).

4.3 Calculated data

4.3.1 Pulse Tuning- Calculating optimum intake runner length for a specific RPM

Calculation of cylinder volume	
Cylinder bore (mm)	79
Cylinder stroke (mm)	81.4
Number of cylinders	4
Cubic capacity (cm ³)	1595.9837299121102

Calculation of ideal runner diameter	
☒ Use values from previous calculation	
Cubic capacity (cm ³)	1595.9837299121102
Number of cylinders	4
Target RPM	1000
Runner diameter (cm)	1.505872293724785

Calculation of runner length	
☒ Use values from previous calculation	
Target RPM	1000
Runner diameter (cm)	1.505872293724785
Duration intake valve (degrees)	232
Runner length (cm)	366.11294042202934 or 183.05647021101467

Calculation of cylinder volume	
Cylinder bore (mm)	79
Cylinder stroke (mm)	81.4
Number of cylinders	4
Cubic capacity (cm ³)	1595.9837299121102

Calculation of ideal runner diameter	
☒ Use values from previous calculation	
Cubic capacity (cm ³)	1595.9837299121102
Number of cylinders	4
Target RPM	5000
Runner diameter (cm)	3.3672328142021492

Calculation of runner length	
☒ Use values from previous calculation	
Target RPM	5000
Runner diameter (cm)	3.3672328142021492
Duration intake valve (degrees)	232
Runner length (cm)	73.45254246106516 or 36.72627123053258

At **1000rpm** the optimum intake length would be **3661mm/366.1cm/3.66m** this figure divided by 2 or 4 gives:

/4 = 0.915m/91.5cm/915mm. Intake runner would nearly be a meter.

/2 = 1.83m/183cm/1830mm. Intake runner would nearly be 2 meters.

At **5000rpm** the optimum intake length would be **735mm/73.5cm/0.735m** this figure divided by 2 or 4 gives:

/4 = 0.18m/18cm/180mm. Intake runner would nearly be 0.2 of a meter.

/2 = 0.36m/36cm/360mm. Intake runner would nearly be 0.4 of a meter.

Figure 36: Calculation for inlet runner length (Exx.se, 2017).

4.3.2 Theoretical Horsepower

The equation below calculates theoretical Horsepower.

Maximum Horsepower achieved- 104.61HP for the simulation and 79.86HP for the live data.

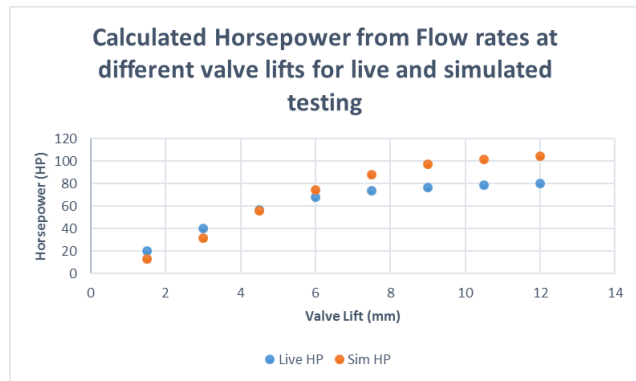


Figure 37: Theoretical Horsepower for valve lift.

$$\text{Horsepower} = (\text{Observed CFM} \times \text{Power Coefficient} \times \text{Number of Cylinders})$$

Baechtel, J. (2011).

Table 12: Theoretical Horsepower for live and sim sessions.

Valve Lift (mm)	Live HP	Sim HP
1.5	20.1	12.96
3	39.89	31.6
4.5	56.38	55.58
6	67.8	74.09
7.5	73.83	88.27
9	76.84	96.98
10.5	78.34	101.35
12	79.86	104.61

4.3.3 Calculated Volumetric Efficiency

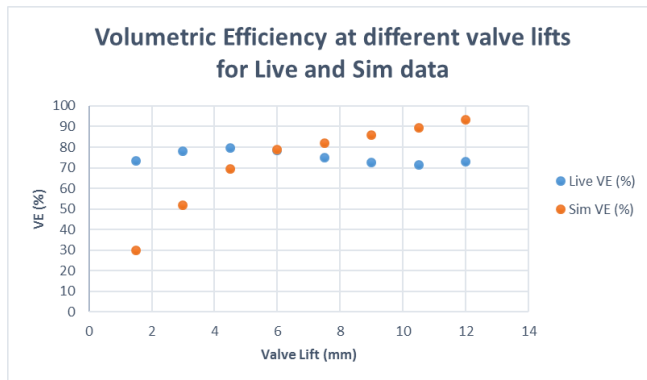


Figure 38: Calculated volumetric efficiency for live and sim.

Valve Lift (mm)	Live CFM	Potential CFM	VE (%)
1.5	18.59	25.3	73.4783
3	36.94	47.3	78.0973
4.5	52.2	65.7	79.4521
6	62.78	80.2	78.2793
7.5	68.36	91.1	75.0384
9	71.15	98.2	72.4542
10.5	72.54	101.5	71.468
12	73.94	101.1	73.1355

Table 13: Calculated Volumetric Efficiency from live and potential.

$$VE = \left(\frac{\text{Recorded CFM}}{\text{Potential CFM}} \right) \times 100$$

Baechtel, J. (2011).

Valve Lift (mm)	Sim CFM	Potential CFM	VE (%)
1.5	7.567429	25.3	29.91078
3	24.46802	47.3	51.72943
4.5	45.65682	65.7	69.49288
6	63.0619	80.2	78.6308
7.5	74.6653	91.1	81.95971
9	84.2507	98.2	85.79502
10.5	90.80914	101.5	89.46714
12	94.34061	101.1	93.31415

Figure 39: Calculated Volumetric Efficiency for sim and potential.

Compared at 6mm both achieve around 78% efficiency which is a more achievable/ realistic valve lift.

The live data achieves a more constant volumetric efficiency but the simulation data achieves higher readings.

This is down to surface finishes and limitations to the Solidworks flow software.

4.3.4 Calculated CFM

The flow efficiency is 95% of the valve area. The valve seat inside diameter is calculated using seat angle (45°) to give a close reading for the actual throat diameter to calculate the theoretical flow rate at different valve lifts.

	Specifications
Valve Diameter (mm)	30
Seat Width (mm)	1.5
Valve Seat Angle (°)	45

Table 14: Valve and seat dimensions.

$$\text{Valve seat id} = \text{valve dia.} - (\text{seat width} \times \cos.45)$$

$$28.94 = 30 - (1.5 \times \cos.45)$$

For the calculated 28.94mm at 1mm of lift the numbers must be in m².

$$\text{Curtain Area} = \pi \times \text{Diameter} \times \text{Lift}$$

$$0.0909 \times 10^{-3} = \left(\frac{28.94}{1000}\right) \times \pi \left(\frac{1}{1000}\right)$$

To calculate at alternative valve lifts the curtain area equation needs to be repeated for those valve lifts so the correct CFM can be theoretically calculated. *See following page for alternative valve lift CFM readings.*

Calculating CFM for 1mm of valve lift and assuming efficiency of 95%.

$$CFM = A \times C \times 213685.34$$

$$18.47 = (0.0909 \times 10^{-3}) \times (0.95) \times (213685.34)$$

The theoretical flow rate for 1mm of valve lift with an efficiency of 95% is 18.47CFM.

Baechtel, J. (2011).

By changing the valve lift figure in the curtain area equation, the correct curtain area figure can be found for each valve lift and thus finding the theoretical CFM.

As the valve lift increases so does the curtain area, giving more room for air the exit the port. A large curtain area produces more CFM.

Valve Lift (m)	Curtain Area (m ²)
1.5	0.000136
3	0.00027
4.5	0.00041
6	0.00055
7.5	0.00068
9	0.00082
10.5	0.00095
12	0.0011

Table 15: Curtain area for valve lifts.

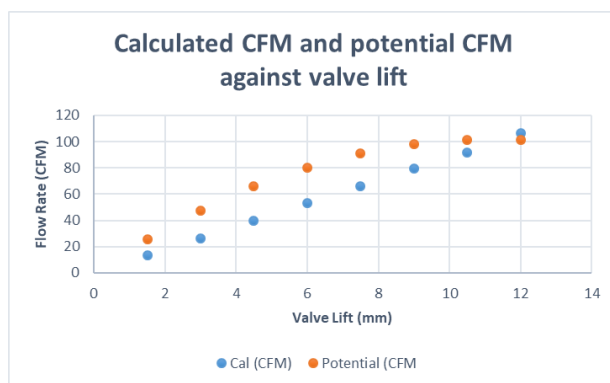


Figure 40: Calculated CFM and potential CFM against valve lift.

	Calculated (CFM)	Potential (CFM)
1.5	13.14	25.3
3	26.1	47.3
4.5	39.63	65.7
6	53.16	80.2
7.5	65.73	91.1
9	79.26	98.2
10.5	91.83	101.5
12	106.3	101.1

Table 16: Calculated CFM for each valve lift.

The data is given linearly with no drop off in CFM as what would be seen in the live and simulated data.

As plotted with the potential CFM, the readings only are similar at low and high-end valve lift. As there is no consideration for pressure differences the calculated data will continue to increase. The calculated would have a similar curve to the potential but with less CFM at each valve lift as when the live data is compared.

4.4 Live, Simulated and Calculated CFM comparison

Valve Lift (mm)	Flow Rate CFM/sqin						
	Calculation		Live	Simulation			
	Potential	Original	Original	Original	Mod1	Mod2	Bigger Valves
1.5	25.3	13.1	18.6	2.6	16.3	15.8	29.8
3	47.3	26.1	36.9	13.4	33.2	34.2	45.5
4.5	65.7	39.6	52.2	42.8	51.8	51.7	55.6
6	80.2	53.2	62.8	48.0	64.3	64.7	67.2
7.5	91.1	65.7	68.4	65.2	73.3	74.7	77.4
9	98.2	79.3	71.2	76.4	80.2	82.3	84.1
10.5	101.5	91.8	72.5	84.8	85.6	86.6	86.9
12	101.1	106.3	73.9	86.3	88.6	90.8	90.4

Table 17: All calculated, Live and Simulated data for flow rate at the same valve lifts.

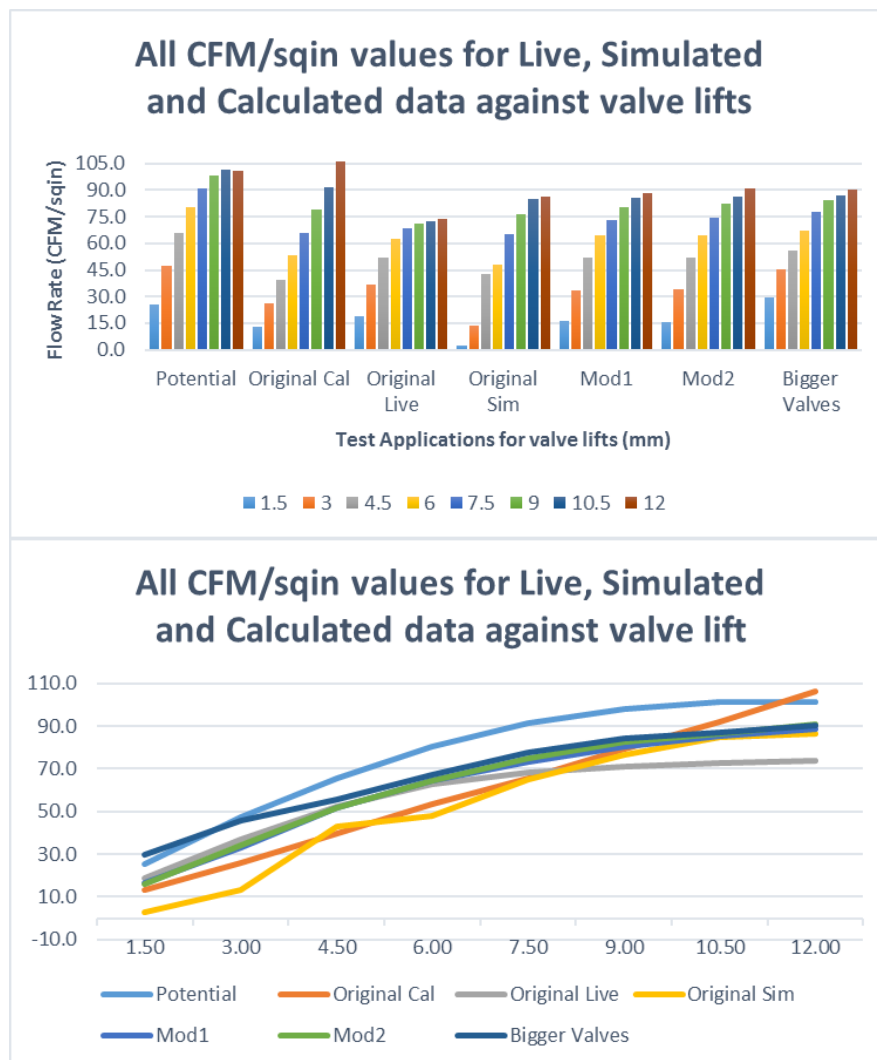


Figure 41: Two graphs depicting all tested CFM values and calculated with the potential against valve lift.

The original live and the two modified ports have very similar readings at lower valve lifts but at 7.5mm of lift the two modified ports have an increase CFM value.

The original simulated port doesn't give great data readings for lower valve lifts but between 4.5mm- 9mm of lift the data is like the calculated.

The calculated often meets the same figures as the tested ports, but can't be taken as accurate compared to the actual live and simulation data which captures velocity, pressure and CFM. The potential CFM is only met by the data traces when the bigger valve port is at low valve lifts.

The 1st modification is the best when considering CFM values and velocity values for port design but the bigger valves maybe able to improve it.

5. Conclusion

The live data shows the flow of the standard port for a 1.6, 16 valve Ford Zetec. Testing took place twice and an average of the data was found.

If testing was undertaken three or more times and the temperature of the air considered in the data then more accurate results could be achieved.

No anomalies were found in the data and the CFM readings increased in a steady linear state and started to flatten out around the same lift.

Simulation showed a higher CFM value than the live tested port. Research gave a figure for the surface finish for a cast Aluminium cylinder head but the limitations of the software gave no change in flow characteristics. Simulation data came back inconclusive but further data from testing showed that the data achieved was realistic.

With increased time more simulations sessions would have been run. And/or different software's used. The flow simulation in Solidworks was a great starting tool to understand flow.

5.1 Recommendations for improvement

Direct-injection from indirect would be a good upgrade as the fuel wouldn't have to travel the length of the port and be caught behind the valves. The injection can occur at the exact time as the fuel is injected directly into the cylinder. Moving the injector gives more freedom on designability for manufacture.

A higher lift camshaft or, if there is sufficient room, bigger valves could theoretically allow more air to enter the chamber but this is substantial to engine size and power requirements in relation to RPM, emissions and application. A longer stroke would reduce friction, thus producing a more efficient engine. Turbocharging would also be a great alternative.

Increasing RPM, as an engine is essentially a pump, will increase volumetric efficiency. The faster an engine revs the more air can travel into the cylinder bore but with increase in heat and decrease in life of moving components.

Testing would be improved with additional time for more live and simulation testing. Knowledge would decrease the time for testing as the process would be known and the chance of mistakes less frequent.

The results collated were in the range with the live data and calculated data giving validity. If the project was to be run again valuable time would be saved in the process of modelling the port. This would give more time for the understanding for the simulation software and the possibility of more modification being made.

To conclude, the simulation modifications made to the inlet port, the first modification would be the best in terms of port design as it achieves a higher than standard CFM value without affecting velocity negatively. If more time was available, the modifications made would be replicated on the actual port and tested on the flow bench.

If port modification is completed correctly, significant power gains can be achieved with little cost. This route will increase the volumetric efficiency which is the main aim of this project without effecting the driveability and emissions of the engine.

In terms of references there is a lot of data on port modification but the subject area is dated in terms of when relative papers were published.

In conclusion, it was found that CFM and in-turn volumetric efficiency can be improved by port modification. Further testing showed that an increase in inlet valve size can increase CFM at but with little room in increasing valve size a re-designed port would achieve the best flow.

6. References

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7. Appendices

