

Powertrain Design and Efficiency

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Abstract

During this assignment, air flow through an engine was studied and a flow bench test carried out to analyse the effects of cylinder head modification for a performance application. The results were entered into a Microsoft Excel sheet and graphs were generated from this to help with the analysis. Through this testing, positive differences were found in the results between the standard and modified cylinder heads. However, a number of issues were identified in the testing process, indicating that the results can only give a general idea at this time, and require further validation to be treated as accurate.

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1. Introduction

In this assignment, airflow into an internal combustion engine will be investigated. This will include research into different induction and fuel delivery systems, as well as flow bench testing to analyse the effects of cylinder head modification. Airflow is the main element to increasing and manipulating an engine's performance. Fuel and air are required to produce torque and power through combustion. Fuel delivery systems have developed over years, and with modern technology, fuelling can be very accurately controlled. These developments of fuel delivery systems will be discussed in the Research section. As internal combustion engines use atmospheric air, it is very difficult to control its induction. Different manufacturers often take different approaches, and many aftermarket companies and race teams then approach the optimisation of these engines for motorsport in different ways.

The aims of the assignment will be to compare the induction systems used with various fuel delivery systems, as well as testing and evaluating the effects of cylinder head modification using a flow bench.

2. Research

Over time, fuel delivery systems have developed to be more efficient, and the induction systems to work with them have also been adapted. Single-carburettor systems were the most common in use on road cars until the 1980's. Carburettors are simple mechanical devices, requiring a relatively low fuel pressure from the pump. They are inexpensive to produce, and they do not require control electronics, also making them inexpensive to implement. The drawbacks of single-carburettor systems include the need for manual calibration (which is time consuming and limited in accuracy and repeatability), inability to adapt to different driving conditions, and the uneven distribution of fuel to cylinders in engines with more than two cylinders. Carburettors also have a tendency to require time consuming maintenance – they can build up debris and dirt, requiring a full strip, clean and rebuild.

Multi-carburettor systems were in use on more performance-oriented vehicles and are still used today for club-level (or historic) motorsport applications. Multi-carburettor systems are more expensive due to the cost of additional carburettors and the manifold(s) and throttle linkage to make it work. They are also more difficult to calibrate, as each carburettor must be perfectly in balance with the others. Having multiple carburettors does give more control over the fuelling and eliminates the issue of unequal fuel distribution to the cylinders.

Looking at air flow, single and multiple carburettor systems are similar. The carburettors are placed high on top of the intake manifold. This is to give the air and fuel plenty of time to mix, but the drawback of this is that the fuel has more area to condense on. It is advantageous for the intake manifold and port surfaces to be of a rough texture to promote turbulence (which helps the air and fuel to mix).

Single point injection systems are a very basic evolution of a single carburettor. Initially, injectors were quite expensive due to their complexity and component size. Over time, production has developed, and they became more cost effective. A single injector in place of a carburettor gives more control over the fuelling, as this is done by an Electronic Control Unit rather than a mechanical device. This means the fuelling can be adjusted to different driving conditions. Controlling it electronically also means the calibration is easy, accurate and repeatable – there is just the initial programming of the ECU, which is then implemented into all of the produced engines. The drawbacks include uneven fuel distribution (similarly to single carburettors), fuel condensation and, as already mentioned, cost. A higher pressure fuel pump is also required for an injector to operate.

The air flow for single point injection systems is similar to carburettors, but the injector is capable of atomising the fuel very well initially. It has to be placed far up the intake manifold to allow the fuel to be distributed to all of the cylinders.

Multi-point injection is more advantageous over the other systems, as it allows precise control via the ECU and ensures equal fuel distribution to each cylinder. Combined with sequential injection, it also minimises condensation within the port. The disadvantages are that it is more expensive, requires slightly more complex calibration and also requires additional sensors on the engine.

For multi-point injection systems, the injectors are placed much closer to the valves, giving less surface area for fuel condensation. The injectors can atomise the fuel and allow it to mix

with the air closer to the valve. This means more work can be done to optimise the airflow before the injectors. The manifold and port surfaces before the injector can be smooth without affecting the fuel mixing. Manufacturers often place a plenum chamber between the throttle body and the intake manifold runners. This allows the air to slow down and build some pressure before entering the combustion chamber. The manifold length can also be increased with multi-point injection systems, to optimise peak torque for lower engine speeds (improving road drivability).

3. Methodology

3.1 Flow Bench Testing

The testing will be carried out using the SuperFlow SF-110 flow bench located in the vehicle inspection workshop (see fig. 1 below). This flow bench allows testing of cylinder heads at a pressure drop of 10" H₂O, rather than 25" H₂O which the SF-600E flow bench can test at. Testing at 25" H₂O would potentially provide more accurate results, however the SF-600E was not available at the time.



Figure 1 - The SF-110 Flow Bench. (Kot, 2018)

To set up the flow bench, an adapter must be used which is appropriate for the bore of the engine. This needs to be clamped onto the cylinder head, with a suitable putty used to temporarily seal the surfaces. Care needs to be taken to ensure that the adapter sits central to the combustion chamber in the cylinder head. The adapter then needs to be bolted down onto the flow bench surface, again using putty as a temporary sealant. At this point, it is a good idea to run the flow bench with the valves closed, to ensure that the cylinder head is sealed well and there are no air leaks. Next, a plate should be bolted to the top of the head, with screws to allow controlled opening of the valves. Dial Test Indicator gauges should be set up on the screws to measure the valve lift.

Before the testing can begin, life values need to be calculated from the required L/D values. In this case the valve head diameter was measured as 27.8mm for the intake and 24.1mm for the exhaust, and the L/D values are 0.05 to 0.4 (in 0.05 increments). To obtain the lift values, the valve head diameter must be multiplied by the desired L/D value (e.g. $27.8 \times 0.05 = 1.39\text{mm}$). The valves were measured using a vernier caliper (see fig. 2 below).



Figure 2 - A valve being measured with a vernier caliper. (Kot, 2018)

Once the valves are opened to the desired lift, the flow bench should be set to the desired flow direction (intake or exhaust) on the control panel (see fig. 3 below). The flow bench should then be turned on, and the flow control (intake or exhaust) turned anti-clockwise until the pressure drop reaches 10" H₂O on the scale (see fig. 4 below).



Figure 3 - Flow bench controls. (Kot, 2018)

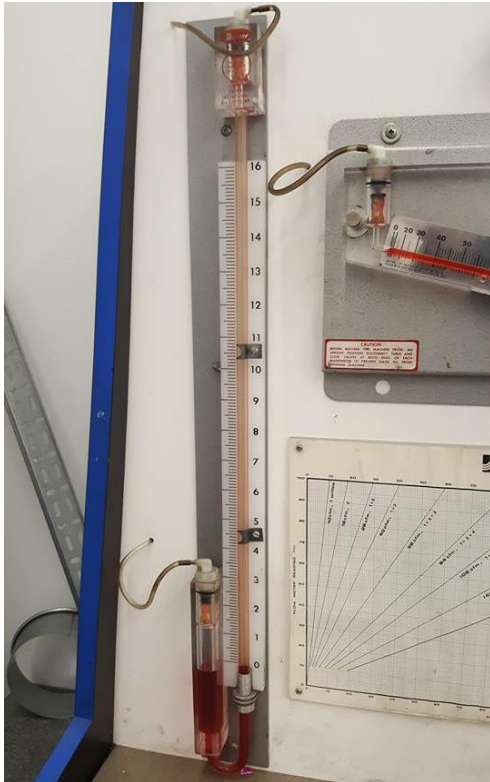


Figure 4 - Pressure drop scale. (Kot, 2018)

The percentage flow measurement can then be taken from the flow scale (see fig. 5 below). Care should be taken to ensure that the flow does not exceed 100% before the pressure drop reaches 10" H₂O. If this is the case, rubber bungs should be removed accordingly to open up the orifices at the top of the flow bench (see fig. 6 below).

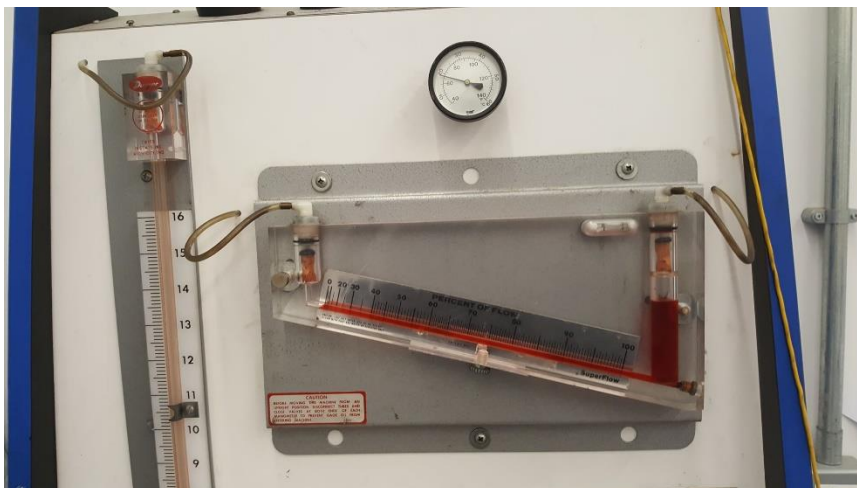


Figure 5 - Percentage Flow Meter. (Kot, 2018)



Figure 6 - Orifice bungs. (Kot, 2018)

The percentage flow needs to be recorded, along with the open orifices. The volumetric flow rate can then be obtained from the graph (see fig. 7 below) in cubic feet per minute.

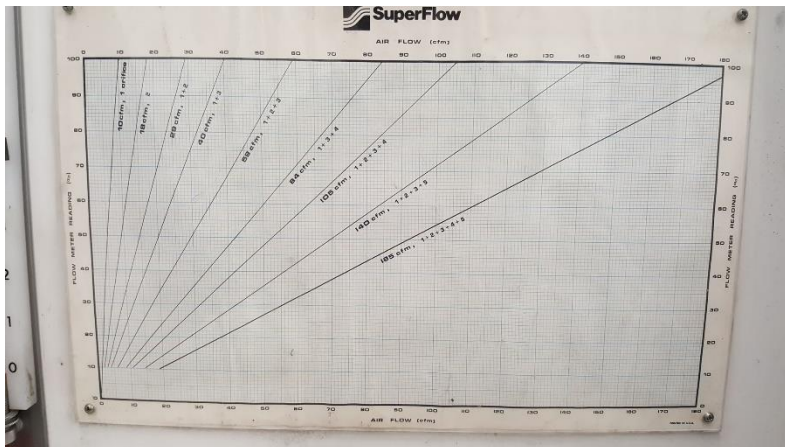


Figure 7 - Percentage flow to volumetric flow rate conversion graph. (Kot, 2018)

3.2 Calculations

As the flow bench uses imperial units, the results need to be converted to SI units.

The cubic feet per minute figures need to be converted to m³/sec. The conversion from feet to metres is 1' = 0.3048m (Metric Conversions, 2018). The volumetric flow rate in cf/m should be multiplied by 0.3048³ to achieve a figure in m³, and then divided by 60 to convert minutes to seconds. The values were obtained in Microsoft Excel using a formula.

To calculate the flow velocities, the valve area is required. This was a simple calculation:

$$A = (\pi r_{valve\ head}^2) - (\pi r_{valve\ stem}^2)$$

The valve stem diameter was 6mm for both intake and exhaust valves. This gave a valve area of 578.713 x10⁻⁶ m² (or 5.787cm²) for the intake valves, and 427.893 x10⁻⁶ m² (or 4.279cm²) for the exhaust valves.

The volumetric flow rate was then divided by the appropriate valve area for each L/D value, again using Excel, to obtain the flow velocities in m/s.

To calculate the coefficient of discharge, the valve flow potential from the flow bench manual needed to be converted to SI units. The valve flow potential is given per cm² of valve area, therefore to simplify, the valve flow potential was calculated for the specific valves. Another formula was used in excel, to multiply the given valve flow potential by the appropriate valve area (in cm²). This gave a potential flow for the valves at each L/D value, but in litres per second. These values were divided by 10³ to convert to m³/s. Another Excel formula was set up to divide the measured volumetric flow rate by the newly calculated and converted potential flow. This gave a discharge coefficient for each L/D value.

4. Results

4.1 Standard Cylinder Head

Table 1 below shows the collected results for the intake port on the standard cylinder head. Table 2 shows the calculated and converted results.

Table 1:

L/D	Lift (mm)	Flow (%)	Bungs Removed	Flow Rate (cf/m)	Temp. (°C)
0.05	1.39	70	1, 2	20.5	38
0.1	2.78	66	1, 2, 3	39	38
0.15	4.17	90	1, 2, 3	53	40
0.2	5.56	76	1, 3, 4	63.5	40
0.25	6.95	84	1, 3, 4	70	40
0.3	8.34	72	1, 2, 3, 4	75.5	40
0.35	9.73	74	1, 2, 3, 4	78	40
0.4	11.12	76	1, 2, 3, 4	79.5	40

Table 2:

L/D	VFR (cf/m)	VFR (m ³ /s)	Velocity (m/s)	Valve Flow Potential	Cd
0.05	20.5	0.0097	16.718	0.00671	1.441866
0.1	39	0.0184	31.805	0.01256	1.465442
0.15	53	0.0250	43.222	0.01742	1.435891
0.2	63.5	0.0300	51.785	0.0213	1.406979
0.25	70	0.0330	57.086	0.02419	1.365702
0.3	75.5	0.0356	61.571	0.0261	1.365212
0.35	78	0.0368	63.610	0.02697	1.36492
0.4	79.5	0.0375	64.833	0.02685	1.397386

Table 3 below shows the collected results for the exhaust port on the standard cylinder head. Table 4 shows the calculated and converted results.

Table 3:

L/D	Lift (mm)	Flow (%)	Bungs Removed	Flow Rate (cf/m)	Temp. (°C)
0.05	1.21	40	1, 2	12	28
0.1	2.41	83	1, 2	24	32
0.15	3.62	91	1, 3	36.5	34
0.2	4.82	79	1, 2, 3	47	35
0.25	6.03	90	1, 2, 3	53	37
0.3	7.23	68	1, 3, 4	57	39
0.35	8.44	71	1, 3, 4	59	39
0.4	9.64	73	1, 3, 4	61	39

Table 4:

L/D	VFR (cf/m)	VFR (m ³ /s)	Velocity (m/s)	Valve Flow Potential	Cd
0.05	12	0.0057	13.235	0.00496	1.141808
0.1	24	0.0113	26.471	0.00929	1.21924
0.15	36.5	0.0172	40.258	0.01288	1.337429
0.2	47	0.0222	51.839	0.01575	1.408351
0.25	53	0.0250	58.457	0.01789	1.398167
0.3	57	0.0269	62.869	0.0193	1.393834
0.35	59	0.0278	65.074	0.01994	1.396434
0.4	61	0.0288	67.280	0.01986	1.449587

4.2 Modified Cylinder Head

Table 5 below shows the collected results for the intake port on the modified cylinder head. Table 6 shows the calculated and converted results.

Table 5:

L/D	Lift (mm)	Flow (%)	Bungs Removed	Flow Rate (cf/m)	Temp. (°C)
0.05	1.39	66	1, 2	19	-
0.1	2.78	61	1, 2, 3	36	-
0.15	4.17	85	1, 2, 3	50	-
0.2	5.56	75	1, 3, 4	63	36
0.25	6.95	68	1, 2, 3, 4	71.5	-
0.3	8.34	76	1, 2, 3, 4	80	-
0.35	9.73	80	1, 2, 3, 4	84	-
0.4	11.12	84	1, 2, 3, 4	88	40

Table 6:

L/D	VFR (cf/m)	VFR (m ³ /s)	Velocity (m/s)	Valve Flow Potential	Cd
0.05	19	0.0090	15.495	0.00671	1.336364
0.1	36	0.0170	29.358	0.01256	1.352716
0.15	50	0.0236	40.776	0.01742	1.354614
0.2	63	0.0297	51.377	0.0213	1.395901
0.25	71.5	0.0337	58.309	0.02419	1.394967
0.3	80	0.0378	65.241	0.0261	1.446582
0.35	84	0.0396	68.503	0.02697	1.469914
0.4	88	0.0415	71.765	0.02685	1.546792

Table 7 below shows the collected results for the exhaust port on the modified cylinder head. Table 8 shows the calculated and converted results.

Table 7:

L/D	Lift (mm)	Flow (%)	Bungs Removed	Flow Rate (cf/m)	Temp. (°C)
0.05	1.21	40	1, 2	12	-
0.1	2.41	83	1, 2	24	-
0.15	3.62	87	1, 3	35	-
0.2	4.82	77	1, 2, 3	45.5	38
0.25	6.03	91	1, 2, 3	53.5	-
0.3	7.23	72	1, 3, 4	60	-
0.35	8.44	78	1, 3, 4	65.5	-
0.4	9.64	84	1, 3, 4	70	40

Table 8:

L/D	VFR (cf/m)	VFR (m ³ /s)	Velocity (m/s)	Valve Flow Potential	Cd
0.05	12	0.0057	13.235	0.00496	1.141808
0.1	24	0.0113	26.471	0.00929	1.21924
0.15	35	0.0165	38.603	0.01288	1.282466
0.2	45.5	0.0215	50.185	0.01575	1.363404
0.25	53.5	0.0252	59.008	0.01789	1.411358
0.3	60	0.0283	66.177	0.0193	1.467194
0.35	65.5	0.0309	72.244	0.01994	1.550279
0.4	70	0.0330	77.207	0.01986	1.66346

5. Analysis

5.1 Physical Differences

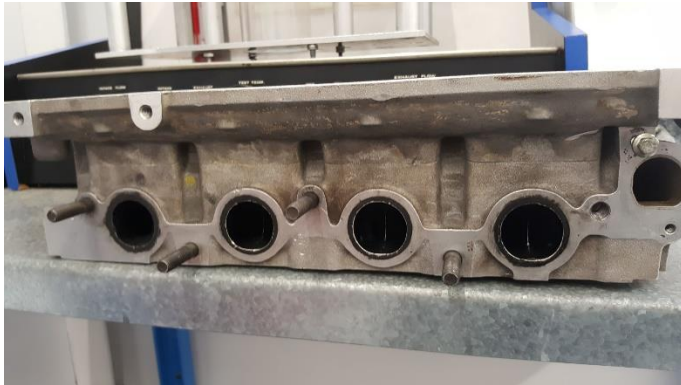


Figure 8 - The modified cylinder head (exhaust side). (Kot, 2018)

Both cylinder heads were inspected and measured to investigate the actual modifications carried out. The standard cylinder head has a rough casting texture on both the intake and exhaust ports, while the modified head has a smooth polished texture (see figures 9 to 12 below).



Figure 10 - Modified intake port. (Kot, 2018)



Figure 9 - Standard intake port. (Kot, 2018)



Figure 12 - Modified exhaust port. (Kot, 2018)



Figure 11 - Standard exhaust port. (Kot 2018)

The diameters of the circular ports were measured using a vernier caliper. The standard head had an intake port diameter of 34.3mm, and an exhaust port diameter of 33.3mm. The modified head had an intake port diameter of 34.7mm (0.4mm bigger) and an exhaust port diameter of 33.6mm (0.3mm bigger). This is quite a small difference, indicating that the only modification carried out was simply polishing the ports – however this is only based on the outer port profiles. It is difficult to measure any

other changes made to the ports. The intake side of both cylinder heads can be seen below in figures 13 and 14.



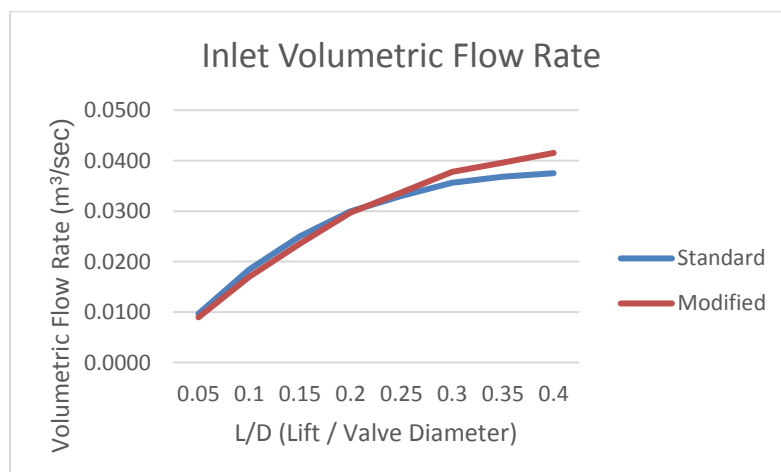
Figure 13 - Standard cylinder head (intake side). (Kot, 2018)



Figure 14 - Modified cylinder head (intake side). (Kot, 2018)

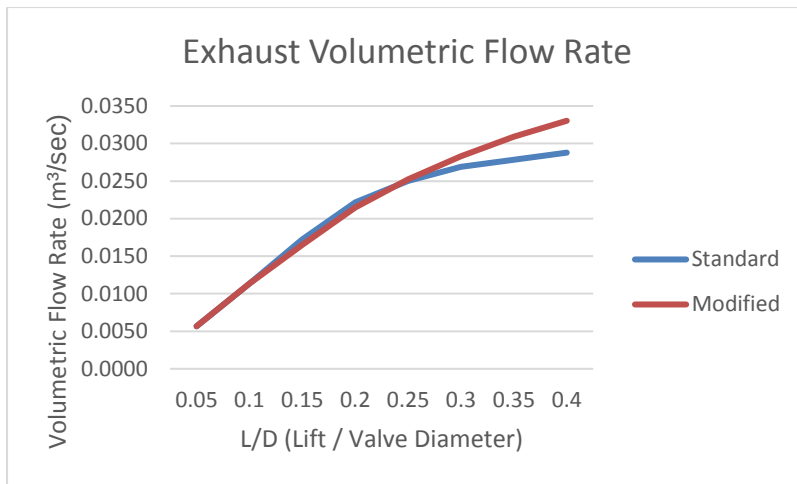
5.2 Flow Bench Results

The results show some interesting patterns in the graphs, which are consistent for both the intake and exhaust side. Graph 1 below shows the volumetric flow rate against L/D for both cylinder heads. Initially, the flow rate is actually slightly lower on the modified head. The graphs cross over at an L/D of around 0.2. The flow rate for the standard head almost reaches a plateau, while the flow rate for the modified head continues to rise and looks like further benefits would be seen from increased lift (although the actual lift of the camshaft is unknown). The initial lower flow rate may be partially due to the fact that testing was conducted at 10" H₂O instead of 25". It may also be related to how the small curtain area and the smooth surface affect the turbulence and boundary layers.



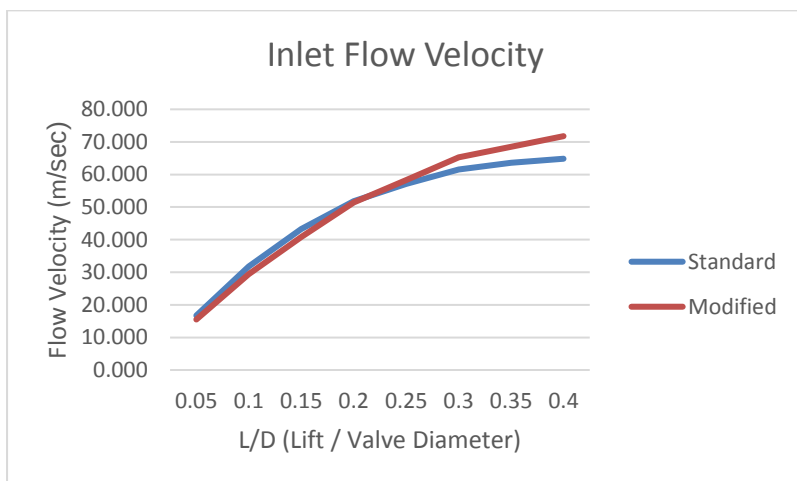
Graph 1 - Inlet Volumetric Flow Rate against L/D. (Kot, 2019)

Graph 2 below shows the volumetric flow rate for the exhaust ports on both heads. Similar results can be seen as the intake, although the modified head is initially closer to the standard head. The results at 0.05 and 0.1 L/D are identical. The flow rate of the modified head then drops very slightly, before crossing over with the standard head around an L/D of 0.2 again. Again, the standard head almost reaches a plateau, whilst the modified head continues to rise.

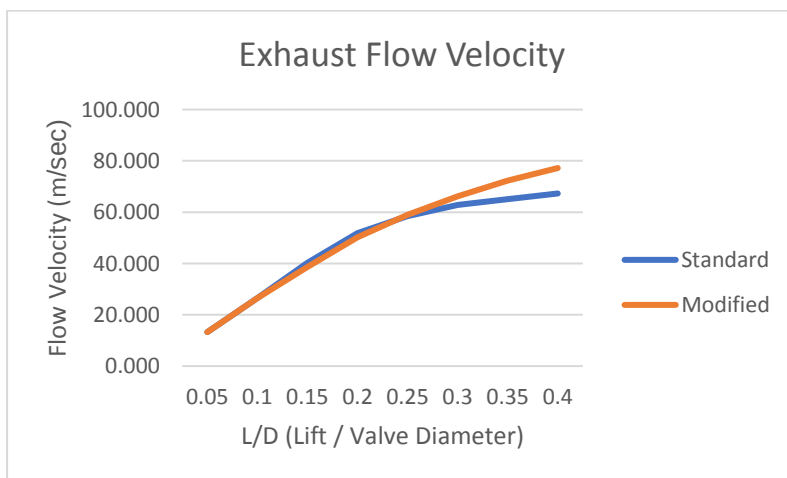


Graph 2 - Exhaust Volumetric Flow Rate against L/D. (Kot, 2018)

Graphs 3 and 4 below show the flow velocities for the intake and exhaust port respectively. As it is simply a division of the VFR by the same valve area, these graphs essentially represent the same information as graphs 1 and 2.

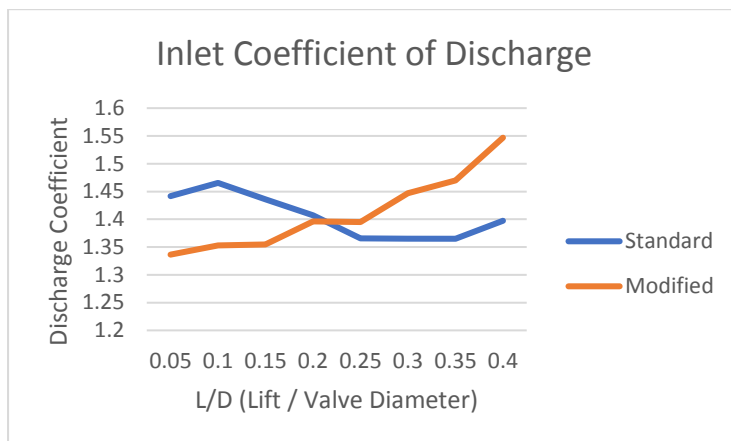


Graph 3 - Inlet Flow Velocity against L/D. (Kot, 2018)



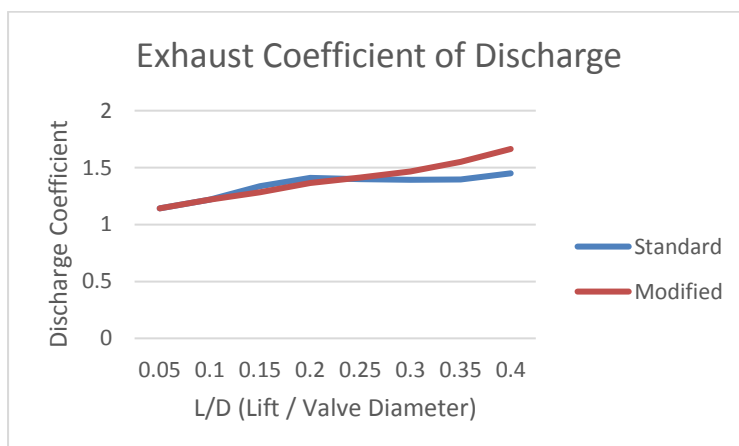
Graph 4 - Exhaust Flow Velocity against L/D. (Kot, 2018)

The bigger improvements can be seen in the discharge coefficient graphs. Graph 5 below shows the discharge coefficients for the intake ports on both heads. The standard head shows a very irregular pattern, with the coefficient dropping before beginning to rise again at the end. Overall, this means that the efficiency of flow is reduced at higher lift. The modified head, although far from being perfectly linear, gives a much better-looking graph. The flow efficiency continues to rise with lift and is much higher (by 0.23) at an L/D of 0.4. The discharge coefficient is however much lower at lower L/D values for the modified head than the standard head. Again, this shows that the modifications are focused on improving flow at high lift values.



Graph 5 - Inlet Discharge Coefficient against L/D. (Kot, 2018)

Graph 6 below shows the discharge coefficient for the exhaust ports on both heads. Again, a similar pattern can be seen, and similarly to the volumetric flow rate graphs, the discharge coefficient drops less from the standard head than the intake. At higher L/D values the modified head graph continues to rise and is much steeper than the standard head graph. In this case, both the standard and modified graphs are a more regular shape. The modified head shows almost a linear line, whilst the standard head shows a plateau between L/D's of 0.2 and 0.35. Overall, there is a good improvement in flow efficiency.



Graph 6 - Exhaust Discharge Coefficient against L/D. (Kot, 2018)

6. Conclusion

To conclude, all of the aims of the assignment have been met – the various fuel / induction systems have been compared, and the results of cylinder head modification were analysed with flow bench testing.

However, there is a number of factors which limit the reliability of the results and analysis. Firstly, the testing should have been conducted on the SF-600E flow bench at a pressure drop of 25" H₂O. Ideally, the SF-110 should have then been used as a validation tool to rule out any inconsistencies with the equipment. Also, all the tests should have been repeated a number of times, and an average taken to ensure repeatability and accuracy.

The testing was conducted over two different days – this means that a number of environmental factors could have changed, such as humidity, ambient temperature, air density etc. All these factors will have an effect on the results to some degree. Ideally, all of the testing should be conducted on the same day in the same environmental conditions.

Test temperature is also an important factor. From the collected results, the observed temperatures ranged from 28°C to 40°C. These temperatures will affect the air density, therefore affecting the volumetric flow rate results. Most of the collected results were at temperatures close to 40°C, however the temperature was not always monitored. This is something that should have received more attention.

Measuring of the ports is also of highly questionable accuracy. Further modifications could have been made deeper into the ports. A far better method to investigate the modifications carried out would be to recreate the port shapes using expanding foam. The port shape could then be scanned, and CAD software used to calculate the volume of the port. Comparing the volumes of the standard and modified ports would be a much more accurate representation of the modifications carried out. Port profiles could also be measured at various locations along the port.

The overall conclusion is that a lot more work would be required to generate an accurate and reliable analysis and comparison of the two cylinder heads, however a general idea can be taken from the analysis carried out.

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