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COMPARATIVE STUDY ON THE IMPROVEMENT OF THE GAS EXCHANGE PROCESS OF A HIGH SPEED IC ENGINE USING SWINGING VALVE

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ABSTRACT

Using poppet valves to control the air-fuel mixture entering and leaving the combustion chamber of an engine is just one among many other more flow efficient alternative solution. The geometry of the poppet valve and its valve seat are the main causes of the flow restriction in the internal combustion engines. The engine downsizing concept dictates to obtain more power from a given engine volume, therefore proportionally more air should be drawn into the cylinders to burn more fuel. These criteria best fulfilled with a new Swinging Valve (SwV) solution that enables the unhindered flow of air and exhaust gas through an engine’s cylinder. The filling of a cylinder is improved while the pumping losses are decreased. In this experiment, a Super Flow SF600 flow bench was used to examine a Suzuki SV650 motorcycle engine’s normal poppet valve cylinder head and a Swinging Valve cylinder head was constructed as well. First the flow parameters of the original cylinder head were obtained then the Swinging Valve head was investigated in the same way. The outcomes of the tests show the superiority of the new concept. The results will also be the base of further 0D/1D engine simulations.

Keywords: IC engine, swinging valve, poppet valve, flow test, 0D/1D engine simulation

1. INTRODUCTION

From the beginning of the earliest times of four stroke internal combustion engines, many ways of controlling air and exhaust flows were experimented. From these poppet valves came out as the most widely used solution. A very handy feature of them is that the gas pressure on the compression and expansion stroke increases the valve seating pressure thus improving sealing. To be able to do so the valve head has to be arranged in its port to practically block the way of gases. This characteristic is quite unwelcome because when the valve is opened the valve head is still located in the middle of the gas stream forcing the flow to change direction and decreasing the engine’s breathing ability and effectively reducing its power capability. Another problem with the universally used poppet valve systems is their control method. The opening of the valve is done by a cam and lifting mechanism (tappets, rockers, etc.) while its retention to its seat is usually performed by a spring. The spring with the mass moving together with the valve creates an oscillating system that will resonate at certain engine speeds. This causes the valves to lose contact with the control mechanism and float. If the speed of the piston reaching TDC is sufficiently high the floating valve can smash into it. Another way of destruction is when the valve smashes into its own seat without the control of the cam/lifter. The result is overly high local stress and premature wear or breaking of the components. With the employment of Swinging Valves (Fig 1.) these boundaries of engine downsizing in general can be done more efficiently and environmental problems could be solved cheaper.

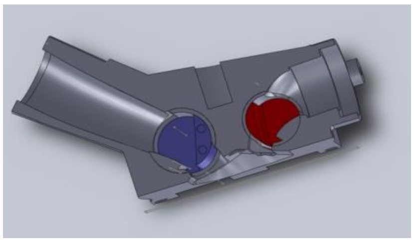

Figure 1. Swinging Valve cylinder head section view

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2. MATERIALS AND METHODS

2.1. Suzuki SV650

To find a suitable engine for the Swinging Valve project a Suzuki SV650 motorcycle engine had been chosen to validate the results. The reason of the choice was its availability; and the fact that it has separate cylinder heads that made actual manual work easier. Also this power-plant is a very robust design. If the Swinging Valve concept can be made working on this type of engine it would probably withstand the abuse in other designs as well Table 1 shows the engine specifications for the SV650 engine and Fig. 2. the torque and power curves as measured on a SuperFlow CycleDyn dynamometer.

Table 1. Technical specifications of the tested engine and cylinder head

Engine Make and type:	Suzuki SV 650	
Engine configuration:	V2	
Year of Manufacture:	2003	
Bore:	81 *mm*	
Stroke	62,6 *mm*	
Swept volume:	0,645 *litres*	
Valve stem diameter	4,5 *mm*	
Valve head diameter	Intake: 31 *mm*	Exhaust: 25,5 *mm*
Valve seat inner diameter:	Intake: 28 *mm*	Exhaust: 23,5 *mm*

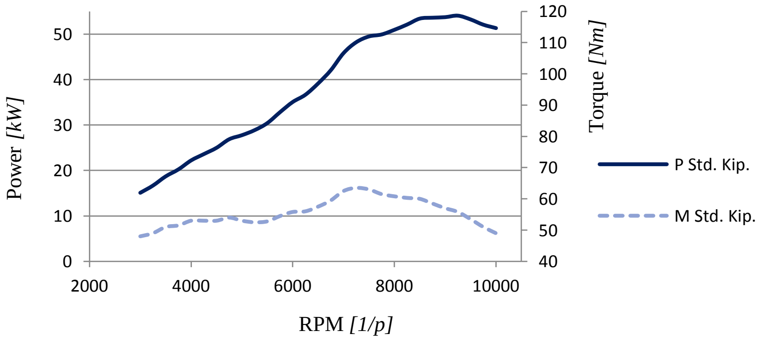

Figure 2. Torque and power curves of the Suzuki SV650 test engine as measured on a SuperFlow CycleDyn dynamometer

2.2. Superflow SF600 flow bench

To examine the given cylinder head's flow capability at different valve lifts a SuperFlow SF600 steady state flow test bench was used (Fig. 3.), which is designed to measure the air-flow resistance of the intake and exhaust conduits of internal combustion engines. The actual test conditions can be found in Table 2., while specifications of the SF600 flow bench is in Table 3. In the case of intake testing it means the air flow resistance of valves, valve seats, manifolds, velocity stacks, and if applicable, restrictor plates. During the test, air is drawn through the cylinder head which is attached to the flow bench via an adapter that replicates the flow masking effects of the cylinder wall. The necessary pressure difference to perform the test is created by specially designed fans that are driven by a set of electric motors. The pressure drop across the valve/valve seat opening is kept constant at each valve lift points in comparison to ambient air

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pressure and is observed on a vertical U-tube manometer (Fig 4.). Further along the air passage there are calibrated measuring orifices. Another U-Tube manometer is located on the machine and it measures the pressure drop across the actual open orifices. The scaling is in percentage of flow. For better accuracy and visibility the readout section of the manometer is inclined. The percentage value indicated on the inclined manometer is used together with a chart attached to the flow bench. The LCD display indicates flow values in cubic feet per second.

Table 2 Information on the volumetric flow test of the SV650 cylinder-cylinder head assembly

Date and place of test	18th February 2019, Szentes
Test pressure drop across valves	62,27 *mbar* (25 *inches* of water)
Ambient temperature	18°*C*
Relative humidity	49%
Barometric pressure	1014 *mbar*
Pressure ratio	1,0653

Table 3 Specifications of Superflow SF-600 FC Flowbench

Calibration Test Pressure:	62,27 *mbar* (25 *inches* of water)
Test Flow Range:	0 – 0,28 m^3/s (0-600 *CFM*)
Capacity:	0,28 m^3/s at 62,27 *mbar* pressure drop (600 *CFM* @ 25 *inches* of water)
Flow measurement accuracy:	± 0,5%
Repeatability:	± 0,25%
Test pressure accuracy:	± 0,127 *mbar*
Temperature measurement accuracy:	± 0,3 °*C*

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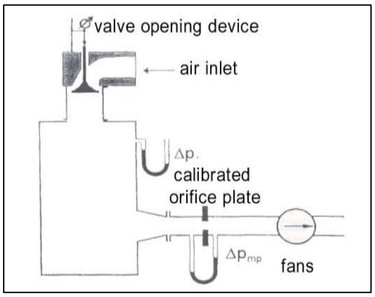
2.3. Flow test methodology

For the actual test a specially designed valve opener device was used, which allows to open and close both valves independently. The valve opener had 1 *mm* pitch on the screw and was placed in line with the axis of the valve stem, each full turn meant 1 *mm* lift at the valve itself.



Δp: pressure drop across the cylinder head
Δp_{mp}: pressure drop across the orifice plate

Figure 4. Typical flow bench testing layout [1]

Regarding data to be processed, the general practice was followed and relative lift values were obtained in a dimensionless format, L/D:

$$L/D = \frac{L}{D} \tag{1}$$

where :
L: Actual valve lift [*mm*],
D: Valve diameter [*mm*].
This way direct comparison is made possible to other designs and makes.

According to [2] the valve was open to 0,3 x valve diameter which equalled approximately 1 *mm*. After turning the device on and adjusting the test pressure to the standard value (62,27 *mbar*/25 *in* H_2O) the flow range was set to get at least a flow of 70%. That was indicated on the inclined manometer. During the test the flow range was adjusted as requested to keep the flow in the desired range. At each lift point the flow was recorded in cubic feet per second and then converted to ISO unit m^3/s.
Leakage test:
Before the flow test the standard test depression was applied to the cylinder head while all the valves and openings were kept closed. To get repeatable results, it was necessary to take any leakage into account and deduct it from the actual results. According to flow bench manufacturer recommendation the amount of leakage flow could be accepted if it was between 0...10 *Cubic Feet per Minute* (*CFM*, 0...16,99 m^3/h). During testing of the SV650 cylinder head the leakage was 1,4 *CFM* (2,379 m^3/h) for the intake test and 1 *CFM* (1,699 m^3/h) for the exhaust test so all the openings and flat surfaces of the tested assembly were acceptably sealed to the outside environment.

3. RESULTS AND DISCUSSIONS

As stated in [1] an intake system that is more efficient in terms of flow losses is favourable as the volumetric efficiency and specific fuel consumption is improved. With better flow characteristics the spread of torque and exhaust gas emissions are also improved. In the light of engine downsizing efforts this translates to smaller engines with the same characteristics and driver perception that a larger, heavier

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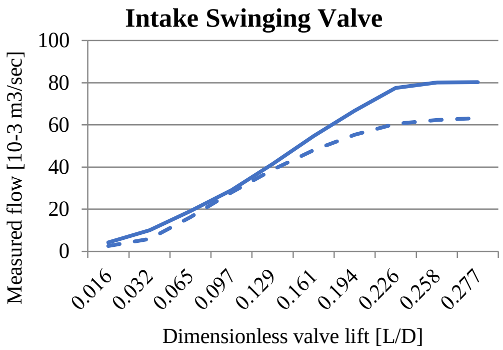
engine would produce with higher fuel consumption. With the flow bench test the flow capabilities of these boundaries are established to the maximum valve lifts. The obtained volumetric flow data was plotted (Fig. 5.) and to allow further comparison with other engine data the Coefficient of Flow (Cf) for all valve lifts were calculated as well according to the test equipment's Operators Manual [2]. The cylinder head was tested without any attachment. To smooth out the flow over the edge of the intake tract modelling clay was used to streamline the entrance of the port and eliminate any detachment of flow downstream the intake entrance. After the measurements taken on the poppet valve cylinder head the newly designed Swinging Valve head was also tested with the same procedure as mentioned above. To get comparable results the flow measurements of the Swinging Valves were performed at the exact opening positions where the SwV flow areas exactly matched the flow areas of the poppet valves at their respective lift points. The results can be seen in graphical form below in Fig. 5.



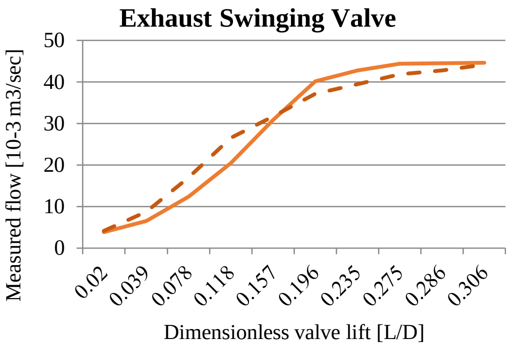


Figure 5. Graphical representations of measured flow parameters of the intake and exhaust Swinging Valves. For comparison, flow values of the original poppet valves are shown with dashed lines

From the measured data the Coefficient of Flow (Cof) was established at each lift point. The calculation was performed according to [2] where the Theoretical Volumetric Flow Rate (TVFR) through a unit area of a perfectly streamlined orifice is used as a datum in the calculations. The value of TVFR is equal to the speed of flow of uncompressible, ideal gas but the unit used adapts to the requirement of evaluating the specific flow capabilities of IC engine valves. For the definition of TVFR the following formula is used:

$$\mathrm{TVFR} = \frac{\dot{V}}{A_t} = \frac{A_t\sqrt{\frac{2\Delta p}{\rho}}}{A_t} \tag{2}$$

where:
V: Volumetric flow rate [m^3/s],
A_t: Flow area of test orifice [m^2],
Δp: pressure difference across the orifice [*Pa*],
ρ: Density of air [kg/m^3].

Entering the values of standardized test pressure of 62,27 *mbar* and the density of air of 1,222 kg/m^3 the resulting Theoretical Volumetric Flow Rate is 100,95 m^3/sm^{-2} or *m/s*. Since this value is just reaching 0,3 Mach so air is considered to be an ideal, incompressible gas while the flow as adiabatic. This value is used as a datum for the calculation of Coefficient of Flow (Cf) of the intake and exhaust valves:

$$\mathrm{Cf} = \frac{\dot{V}/A_{ac}}{\mathrm{TVFR}} \tag{3}$$

where:

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A_{ac}: Actual valve flow area at each valve lift point, where the flow is perpendicular to an imaginary cylindrical flow surface [m^2]

This type of definition is rather idealized because it assumes that flow across the valve is determined by valve diameter. Moreover the imaginary flow surface generated during valve event is cylindrical and particles travel through it perpendicularly (Fig. 6).

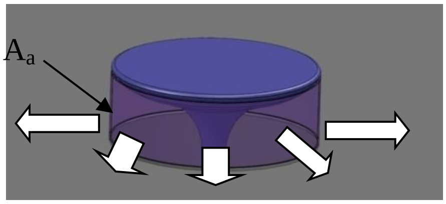

Figure 6. Imaginary flow area of a poppet valve. The white arrows show the assumed particle flow trajectories

After entering the actual values of valve lift points for the intake and exhaust valves the values of Coefficient of Flow were determined for each valve lift points. These are represented graphically in Fig. 7. From the graph it is clear that the exhaust valves and ports with their 44,4% Cf are far from their capacity. Also the waviness of the graph indicates irregularities in the flow field. The intake port is just the opposite in terms of flow quality: the graph is smooth without disturbances in the flow field and plateaus at 42,5% .It shows that the design reaches its limit within the available valve lift. Therefore any development should be focused on this area to improve the intake breathing characteristics because that would largely improve the engine's fuel consumption and overall efficiency.

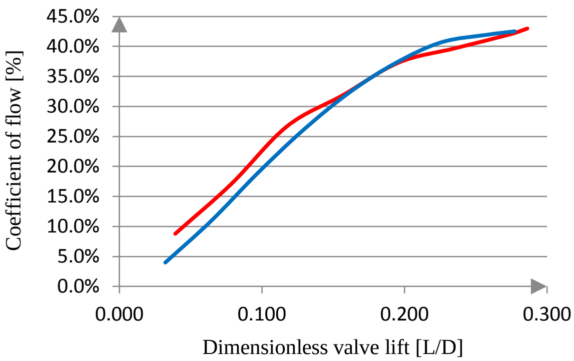

Figure 7. Calculated Coefficient of Flow for the intake (blue) and exhaust (red) poppet valves

The same procedure was repeated with the Swinging Valve cylinder head. The graphs of the calculated coefficient of flows show a marked increase in the flow capacity of the Swinging Valve cylinder head that can be seen in Fig. 8.

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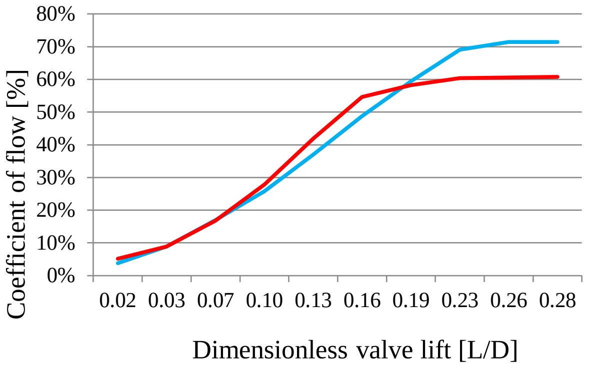

Figure 8. Calculated Coefficient of Flow for the intake (blue) and exhaust (red) Swinging Valves

To ease numerical comparison from the obtained data the area based average values of Coefficient of Flow were determined. For the poppet valve cylinder head the average value of CoF is 25,38% for the intake valves and 25,48% for the exhaust valves. The same parameters for the SwV cylinder head are 39,94% and 36,84% respectively which is 14,5% and 11,36% increase for the intake and exhaust valves respectively.
To facilitate further research with 0D/1D engine simulations Coefficient of Discharge (Cd) was also calculated, which is the actual rate of contraction of flow. In this case the flow area is is conical and it changes during valve lift. This behaviour is embedded in the calculation procedure [3] (Fig 9.).

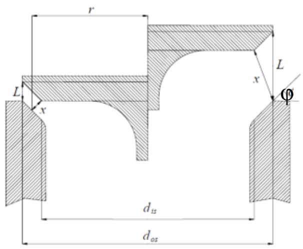

Figure 9. Graphical representation of the shape and position of the flow area [3]

For the calculation first the theoretical flow area (A_t) must be determined. The surface area of a cone frustum is given by:

$$A_t = X\pi\left(\frac{d_{max} + d_{min}}{2}\right) \tag{5}$$

where:
d_{max}: Greater diameter of the cone frustum [*m*],
d_{min}: Smaller diameter of the cone frustum [*m*],
X: Side wall height of the cone frustum [*m*].

If the lift is sufficiently small then the value X is perpendicular to the valve seat. As the valve lifts ever higher, the value of X is such that the wall of the frustum is no longer perpendicular to the valve seat (Fig. 7). The limiting value of lift when this occurs is given by the following equation:

$$L_{lim} = \left(\frac{d_{os} + d_{is}}{\sin 2\varphi}\right) \tag{6}$$

Up until the valve lift is equal to the limiting lift, the minimum flow area is the frustum cone area, such that:

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$$A_t = X\pi\left(\frac{d_{is}}{2} + r\right) \tag{7}$$

Up to the limiting lift point X can be calculated using the following formulae:

$$X = L\cos\varphi \tag{8}$$

The value of r can be found:

$$r = \frac{d_{is}}{2} + X\sin\varphi \tag{9}$$

At lifts greater than the limiting value, the value of X is given by:

$$X = \sqrt{\left(L - \frac{d_{os} + d_{is}}{2}\tan\varphi\right)^2 + \left(\frac{d_{os} + d_{is}}{2}\right)^2} \tag{10}$$

Using these relations between valve and valve seat and from the above definition, Cd can be determined as follows:

$$Cd = \frac{A_{ac}}{A_t} \tag{11}$$

where:
A_{ac}: Actual conical valve flow area at each valve lift point [m^2],
A_t: Theoretical valve flow area at each valve lift point [m^2].

Since the experiment was a steady state flow test and the pressure drop across the valve annulus was kept constant at all opening point the flow speed was constant as well regardless of valve lift. For this reason the measured flow values collected during the test were determined only by the actual flow area. Dividing the volumetric flow rate values by the flow speed we obtain the actual flow area [4]. Using this theory the Coefficient of Discharge is calculated as follows:

$$Cd = \frac{\dot{V}_{ac}}{A_t\sqrt{\frac{2\Delta p}{\rho}}} \tag{12}$$

After completing the calculations the following graphs were plotted for the Cd values of the poppet valve cylinder head (Fig. 10.).

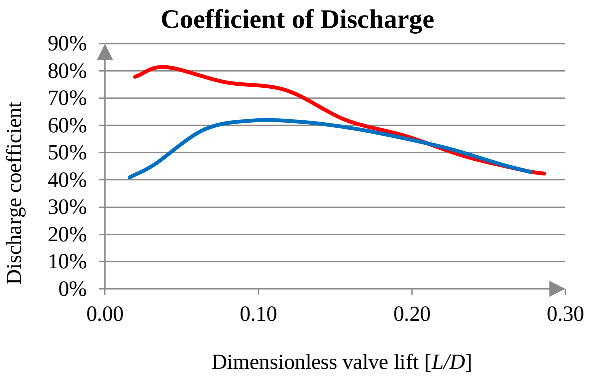

Figure 10. Coefficient of Discharge (Cd) values for Suzuki SV650 exhaust (red) and intake (blue) poppet valves

Even though the parameters remained unchanged during the test the flow changes throughout the valve lift range because the shape of the actual flow passage changes. In [5] is stated that there are three stages of the port opening where the incoming charge has different flow conditions (Fig. 11.). These are:

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- Low lift: the main restriction is caused by the vortices at the edge of the intake valve sealing area and the inner diameter of the valve seat.
- Intermediate lift: The flow separates from the edges of the valve. The effective flow area is decreased due to the separated flow.
- High lift: The flow separates from the valve seat edge as well and the effective flow area further decreased but it is compensated by the increasing opening of the valve.

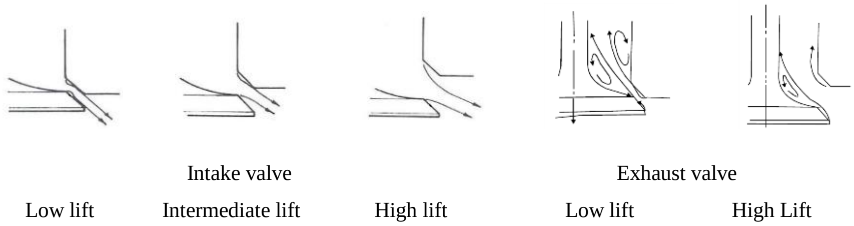

Figure 11. Three different stages of intake valve lift and the two phases of exhaust valve lift. Note how heavily affects the lift the actual flow patterns through the different phases [5]

As can be seen from the graphs above, the intake valves reached Cd value 61,8% highest because eddies generated by the edge of the valve seating rim and inner edge of the valve seat effectively limit the achievable Cd values. The exhaust valves behave differently since the flow direction is reversed (Fig 11.). Air moves through the moderately streamlined flow passage that is composed of the cylinder head surface and valve seat from one side and the valve greater diameter and edge on the other side. At very low lifts these two surfaces create a Venturi-like channel therefore the highest Cd value of 81,4% is reached at 0,039 L/D. As the valve lifts farther away from its seat the flow detaches from the surfaces and from about half of the useful valve lift the exhaust side performs similarly to the intake. For comparison, the Swinging Valve cylinder head Cd values were also plotted and were overlaid on the graphs of the poppet valve cylinder head (Fig. 12-13.). It is clear that the Swinging Valve arrangement generates better Cd values than the original poppet valve.

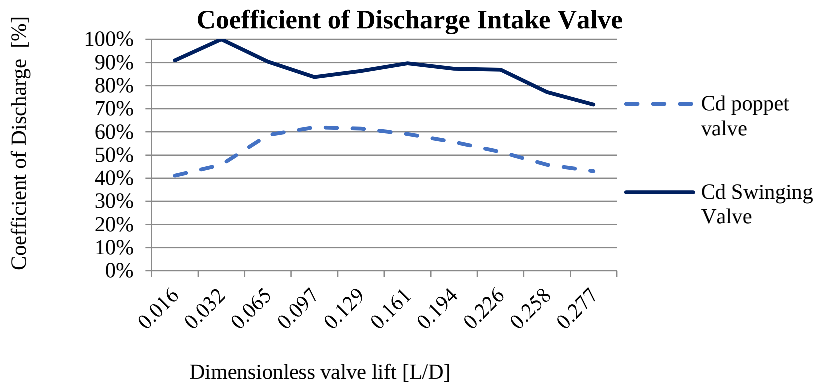

Figure 12. Coefficient of Discharge (Cd) values for intake Swinging Valve. For comparison, Cd values of the original poppet valves are shown with dashed lines

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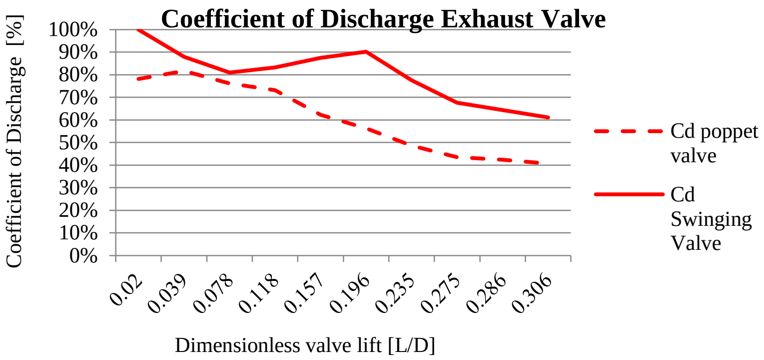

Figure 13. Coefficient of Discharge (Cd) values for exhaust Swinging Valve. For comparison, Cd values of the original poppet

4. CONCLUSIONS

The collected data shows that the traditional poppet valves heavily constraint any engine's breathing ability. It is stated in [2] that the flow capabilities of an engine's cylinder head closely correlate its performance and efficiency parameters. Also through empirical assumptions some other engine data can be derived from the flow numbers (eg. valve opening and closing points, power/torque values, etc.). As can be seen from the previous chapters a more efficient engine can be produced from the existing SV 650 with the unconventional Swinging Valve systems which present greater scope. This research program proved that an arrangement where the intake and exhaust ports are uncovered by a fully or partially rotating assembly can perform vastly better than the original poppet valve system. This special solution will eliminate the problems associated with poppet valves. The Cd values collected during this present work will be used for validation of CFD simulations and 1D engine models of poppet valve and engines with unconventional valve systems.

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