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Engine Power Calculation Using Air Flow Through Engine from Flowbench Test Flow of Four Stroke Direct Injection Diesel Engines

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Abstract: The objective of this study, is to present the experimental results for diesel engine power calculation based on air flow through engine from test flow of engine cylinder head using Superflow Flowbench SF-1020. In the SuperFlow Flowbench for four-cycle engine testing, air is drawn in through the cylinder head into the machine, through the air pump and exits through the vents at each side of the flow bench. A flow test in SuperFlow Flowbench consists of blowing or sucking air through a cylinder head or other component at a constant test pressure then the flow rate is measured at various valve lift. In this experiment, test pressure is setting at 1651, 1397, 1143, 889 and 635 of mm H2O, then the valve lifts in setting at 0.05, 0.1, 0.15, 0.2 and 0.25 of L/D ratio in every test pressures. The experiment and the calculations results are shown in table and figure in this study. The result shown that increasing of the valve lift and test pressure are positive correlation to increase the air flow through engine, engine power and engine speed.

Key words: Engine power, engine speed, flow velocity, percent potential air flow, test pressure, valve lift

INTRODUCTION

Since air breathing capacity and utilization determine the output of the diesel engine, the flow characteristics of the intake and exhaust are crucial in the achievement of good performance (Challen *et al.*, 2003). The design and development of effective intake and exhaust systems remains an engineering challenge. Routinely intake and exhaust external flow systems are now designed and optimized using various analysis equipments and software systems. These are based on various mathematical model approaches, one dimensional solution and three dimensional solutions.

The intake valve and intake port is designed to give the freest possible entry to the air and timings are chosen to obtain the highest practical volumetric efficiency. The type of intake port has used on the direct injection engine, where the fuel is sprayed directly into the cylinder, the piston having a central circular combustion chamber formed in it. The intake port form must be designed to caused the ingoing air to rotate or swirl about the cylinder axis. The degree of swirl needed varies with the size and type of combustion chamber employed which is interrelated to the number of nozzle sprays used (Challen *et al.*, 2003; Heywood, 1998).

The valve, or the valve and the port together is usually the most important flow restriction in the intake and the exhaust system of four stroke cycle engine (Kowalewicz, 1984; Stone, 1997; Heywood., 1998; Ganesan, 1999). The inlet port is generally circular or nearly so and the cross-sectional area is no larger than is required to achieve the desired power output. For the exhaust port, the important of a good valve seat and guide cooling, with the shortest length of exposed valve stem, leads to different design. Although a circular cross section is still desirable, a rectangular or oval shape is often essential around the guide boss area.

The instantaneous valve flow area depends on valve lift and the geometric details of valve head, seat and stem. There are three separate stages to the flow area development as valve lift increases, as shown in Fig. 1. For low valve lifts, the minimum flow area corresponds to a frustrum of a right circular cone where the conical face between the valve and the seat, which is perpendicular to the seat, defines the flow area. According to Heywood (1998) for the stage as below:

$$\frac{w}{\sin\beta\cos\beta} > L_v > 0 \quad (1)$$

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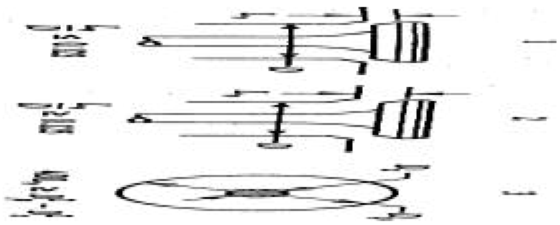

Fig. 1: Schematic three stage of valve lift (Heywood, 1998)

And the minimum area is:

$$A_m = \pi L_v \cos\beta \left(D_v - 2w + \frac{L_v}{2} \sin 2\beta \right) \qquad (2)$$

Where, β is the valve seat angle, L_v is the valve lift, D_v is the valve head diameter and w is the seat width. For the second stage, the minimum area is still the slant surface of a frustrum of a right circular cone, but this surface is no longer perpendicular to the valve seat. The base angle of the cone increase from (90-β)° toward that of a cylinder 90° (Heywood, 1998).

$$\left[\left(\frac{D_p^2 - D_s^2}{4D_m} \right) - w^2 \right]^{1/2} + w \tan\beta \geq L_v > \frac{w}{\sin\beta \cos\beta} \qquad (3)$$

and then,

$$A_m = \pi D_m \left[\left(L_v - w \tan\beta \right)^2 + w^2 \right]^{1/2} \qquad (4)$$

Where, D_p is the port diameter, D_s is the valve stem diameter and D_m is the mean seat diameter (D_v-w).

Finally, when the valve lift is sufficiently large, the minimum flow area is no longer between the valve head and seat; it is the port flow area minus the sectional area of the valve stem.

So, the L_v is:

$$L_v > \left[\left(\frac{D_p^2 - D_s^2}{\quad} \right)^2 - w^2 \right]^{1/2} + w \tan\beta \qquad (5)$$

then,

$$A_m = \frac{\pi}{4} \left(D_p^2 - D_s^2 \right) \qquad (6)$$

Intake and exhaust valve open areas corresponding to a typical valve-lift profile are plotted versus camshaft angle. These three different flow regimes are indicated.

The mass flow rate through a poppet valve is usually described by the equation for compressible flow through a flow restriction. This equation is derived from a one-dimensional isentropic flow analysis and real gas flow effects are included by means of an experimentally determined coefficient of discharge or CD (Heywood, 1998; Superflow, 2004). The port flow coefficient of discharge for a particular flow discontinuity is defined as the ratio of actual discharge to ideal discharge. In an engine environment, ideal discharge considers an ideal gas and the process to be free from friction, surface tension, etc (Blair *et al*., 1995, 1998; Blair, 1999). Discharge coefficients are widely used to monitor the flow efficiency through various engine components and are quite useful in improving the performance of these components (Fleck *et al*., 1996; Danov, 1997). The air flow rate is related to the upstream stagnation pressure P_0 and stagnation temperature T_0, static pressure just downstream of the flow restriction and a reference area A_R characteristic of the valve design as below:

$$m = \frac{C_D A_R P_0}{(RT_0)^{1/2}} \left(\frac{P_T}{P_0} \right)^{1/\gamma} \left\{ \frac{2\gamma}{\gamma - 1} \left[1 - \left(\frac{PT}{P0} \right)^{(\gamma-1)/\gamma} \right] \right\}^{1/2} \qquad (7)$$

The objective of this research is to present the experimental results for diesel engine power calculation based on air flow from test flow of engine cylinder head using Superflow Flowbench SF-1020. The SuperFlow Flowbench is designed to measure the air-flow resistance of engine cylinder heads, intake manifolds, velocity stacks and restrictor plates (SuperFlow, 2004). In the SuperFlow Flowbench, for four-cycle engine testing, air is drawn in through the cylinder head into the machine, through the air pump and exits through the vents at each side of the flow bench. A flow test in SuperFlow Flowbench consists of blowing or sucking air through a cylinder head or other component at a constant test pressure. Then the flow rate is measured at various valve lift. A change can be made and then the component can be re-tested. Greater air flow indicates an improvement. If the tests are made under the same conditions, no corrections for atmospheric conditions or machine variations are required. The results of the experiment investigation may be compared directly. For more advanced tests, it is possible to adjust and correct for all variations so test results may be compared to those of any other head, tested under any conditions on any other SuperFlow Flowbench.

MATERIALS AND METHODS

The experiment and calculation of four stroke direct injection diesel engine power based on air flow from test flow of engine cylinder head using Superflow Flowbench SF-1020 is presented in this study. The specification of the selected four-stroke direct-injection diesel engine is shown in Table 1 (Bakar *et al*., 2006, 2007a, b; Semin *et al*., 2007).

In the experiment, the diesel engines cylinder heads are mounted onto the Flowbench by a cylinder adapter. The adapter consists of a engine cylinder replica about 86 mm, long with the same bore as the engine cylinder in 70 mm and a flange on one end. The flange is bolted to the flow tester and the upper flange is bolted or clamped to the test cylinder head. The flanges must be flat or gasketed to make an airtight seal. According to Superflow (2004) the adapter cylinder clearance may be 0.06 inch or 1.5 mm, larger or smaller than the actual diesel engine cylinder. The adaptor and the thread to open and close the engine valve were developed from metal using CNC machine. As the thread is rotated, it pushes open the valve. In this experiment, in one rotation of the thread the valve is opened in 0.5 mm. Dial indicator may be mounted to the same fixture with its tip contacting the valve spring retainer to measure the amount of valve opening. The original valve springs is used in this experiment. The intake valve opened in the experiment of cylinder head air flow investigation the four stroke direct injection diesel engines is shown in Table 2.

All diesel engine valves in this test should be performed at the same ratio of valve lift to valve diameter or L/D ratio. Then the flow efficiencies of any valves can be compared, regardless of size. In this research, multiply L/D ratios are shown in Table 2. The L/D ratios are to obtain the valve lift test points. To perform the experiment is used the test orifice plate for calibration. Remove the test orifice plate from the Flowbench and install the test head, cylinder adapter and valve opener for the actual flow investigations. In this research, the dial indicator was set in zero with the valve closed. Then install either the intake manifold or an air inlet guide on intake port. The test pressure is setting at 1651, 1397, 1143, 889, 635 of mm H2O and the test range is based on intake valve diameter or in 71 liter per second flow range. To calculate the air flow experiment results data, it is necessary to measure the corrected test flow. In this experiment no atmospheric corrections. To obtain the valve efficiency, it is necessary to calculate the flow in (l/s)/cm^2 or liters per second (l/s)/square centimeter (cm^2) of the valve area and then to compare that flow to the best yet achieved. Potential flow of the engine intake and exhaust manifold investigation is using the potential flow chart (Superflow, 2004), of valve flow potential per unit area based on the test pressure of experiment. To calculate of percent potential flow is equal test flow divided by potential flow. The % potential flow can be used as an indicator of the remaining improvement possible. To determine the air flow Coefficient of Discharge (CD) is divide the test flow per unit area by maximum potential flow per unit area for test pressure.

Table 1: Specification of diesel engine

Engine parameters	Value
Bore×stroke (mm)	86×70
Normal power (kW)	6.6
Normal mean effective pressure (kPa)	543.5
Maximum intake valve open (mm)	7.095
Maximum exhaust valve open (mm)	7.095
Intake valve diameter (mm)	35.54
Exhaust valve diameter (mm)	29.04
Intake valve stem (mm)	7.0
Exhaust valve stem (mm)	7.0
Intake valve effective area (sq.cm)	9.531
Exhaust valve effective area (sq.cm)	6.235

Table 2: Intake valve position in experiment

		Intake valve	
Test No.	L/D ratio	Lift (mm)	Flow range
1	0.05	1.78	72.69
2	0.10	3.55	72.69
3	0.15	5.33	72.69
4	0.20	7.11	72.69
5	0.25	8.89	72.69

The power of an engine is directly proportional to the amount of air drawn into the cylinder and retained until ignition occurs. By reducing the air flow resistance of the intake and exhaust tract, cylinder filling is improved and engine power is increased directly. In this experiment is to investigate the relationship of capacity on the flowbench, the power and the speed of the engine. Test has been developed to show if measured at maximum valve lift, a well developed engine will produce power engine per cylinder and engine speed are follow:

$$\text{Engine power (kW)} = 0.44 \times T_{FC} \times \left(\frac{60}{T_p}\right)^{1/2} \quad (8)$$

$$\text{Engine speed (RPM)} = 42,200 \times T_{FC} \times \frac{\left(\frac{60}{T_P}\right)^{1/2}}{V_d} \quad (9)$$

Where, T_{FC} is the correction test flow value of intake flow in liter per second, T_P is test pressure value of intake flow in mmH2O and V_d is engine volume displacement in millititer.

RESULTS AND DISCUSSION

The total air flow through the engine cylinder is ultimately determined by the valve diameters. While well-designed smaller valves will out-perform larger valves on occasion a good, big valve will always out-flow a good

Table 3: Air flow at 1651 mm H2O test pressure

Object	Experiment data result					
Valve lift (mm)	0	1.78	3.55	5.33	7.11	8.89
Flow range (l/s)	72.69	72.69	72.69	72.69	72.69	72.69
Correction test flow (l/s)	0	16.05	30.68	43.9	53.81	57.11
% velocity	3.9	4.2	4.1	5	4.5	4.1
Velocity (m/s)	6.34	6.86	6.74	8.05	7.10	6.34

Table 4: Air flow at 1397 mm H2O test pressure

Object	Experiment data result					
Valve lift (mm)	0	1.78	3.55	5.33	7.11	8.89
Flow range (l/s)	72.69	72.69	72.69	72.69	72.69	72.69
Correction test flow (l/s)	0	11.8	25.96	38.23	47.67	51.45
% velocity	4	4.1	3.8	4.6	3.7	4.1
Velocity (m/s)	5.92	6.311	5.76	6.86	5.61	6.19

Table 5: Air flow at 1143 mm H2O test pressure

Object	Experiment data result					
Valve lift (mm)	0	1.78	3.55	5.33	7.11	8.89
Flow range (l/s)	72.69	72.69	72.69	72.69	72.69	72.69
Correction test flow (l/s)	0	10.38	23.6	33.98	42.48	45.78
% velocity	4.1	3.9	4.3	4	3.5	4.3
Velocity (m/s)	5.61	5.28	5.76	5.48	4.76	5.76

Table 6: Air flow at 889 mm H2O test pressure

Object	Experiment data result					
Valve lift (mm)	0	1.78	3.55	5.33	7.11	8.89
Flow range (l/s)	72.69	72.69	72.69	72.69	72.69	72.69
Correction test flow (l/s)	0	9.44	20.77	29.74	36.82	39.18
% velocity	6.1	5.4	5.8	5.4	4.7	5.1
Velocity (m/s)	7.23	6.46	6.86	6.46	5.61	6.07

Table 7: Air flow at 635 mm H2O test pressure

Object	Experiment data result					
Valve lift (mm)	0	1.78	3.55	5.33	7.11	8.89
Flow range (l/s)	72.69	72.69	72.69	72.69	72.69	72.69
Correction test flow (l/s)	0	7.55	16.99	25.02	30.21	32.10
% velocity	6.1	6.5	6.6	5.7	6.1	6.9
Velocity (m/s)	6.19	6.62	6.62	5.76	6.19	6.98

than smaller valve. Valve size is limited by the diameter of the engine bore. For wedge-shaped combustion chambers, the practical maximum intake valve diameter is 0.52 times the bore diameter. Hemi-heads and pen-roof chambers permit intake valves up to 0.57 times the bore diameter due to the extra space available in the combustion chamber. The air flow through the engine cylinder is directly controlled by the valve lift. The farther the valve opens, the greater the flow, at least up to a point. In order to discuss a wide variety of valve sizes, it is helpful to speak in terms of the ratio of valve lift to valve diameter, or L/D ratio. Stock engines usually have a peak lift of 0.25 of the valve diameter and for racing engines open the valves to 0.30 of the valve diameter or even 0.35 of the valve diameter (SuperFlow, 2004).

Power, displacement and engine air flow capacity are all related in a definite fashion. With the wide spread use of accurate engine flowbenches, it has become possible to measure the air flow potential of an engine and predict its potential power. The air flow through engine and its effect in engine power experiment is conducted in this research. The air flow research results are shown in Table 3-7.

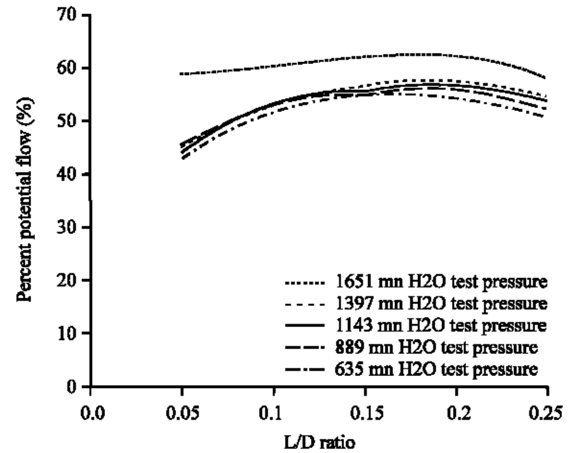

Fig. 2: Intake valve percent potential flow in variation pressure

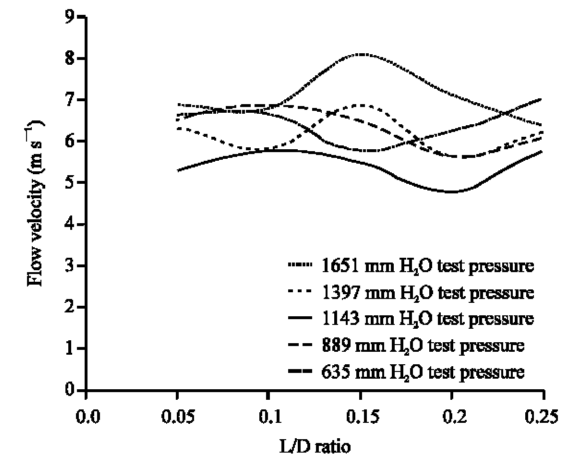

Fig. 3: Intake valve air flow velocity in variation pressure

Table 3 shows the intake valve lifts effects on air flow in the intake valve through engine at 1651 mm H2O test pressure. Table 4 shows the intake valve lifts effects on air flow of the intake valve through engine at 1397 mm H2O test pressure. Table 5 shows the intake valve lifts effects on air flow of the intake valve through engine at 1143 mm H2O test pressure. Table 6 shows the intake valve lifts effects on air flow of the intake valve through engine at 889 mm H2O test pressure. Table 7 shows the intake valve lifts effects on air flow of the intake valve through engine at 635 mm H2O test pressure.

The experiment data results from Table 3 until Table 7 is used to calculate the percent potential air flow, air flow velocity in the intake valve through to engine cylinder, engine power and engine speed. By using the Eq. 8, it can to calculate the engine power prediction based on air flow from the Superflow Flowbench and by using the Eq. 9, it can to calculate the engine speed prediction based on air flow from the Superflow

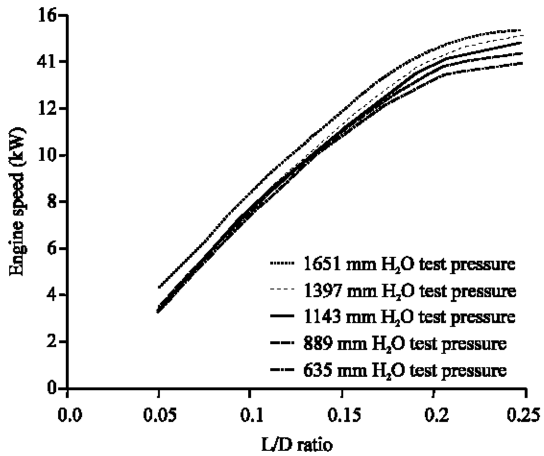

Fig. 4: Air flow in intake valve effect for engine power in variation pressure

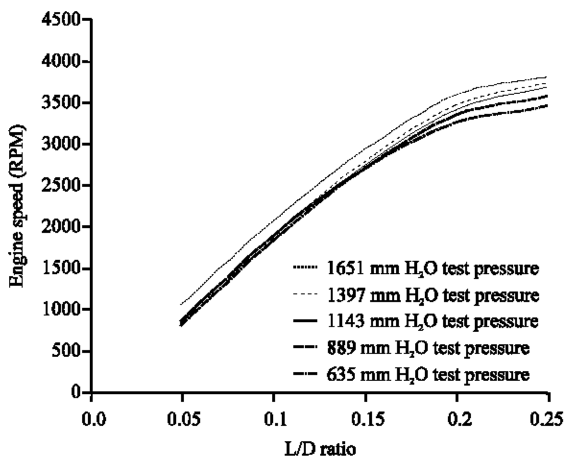

Fig. 5: Air flow in intake valve effect for engine speed in variation pressure

Flowbench. The air flow, percent potential air flow, air flow velocity, engine power and engine speed calculation for the four stroke direct injection diesel engine is shown in Fig. 2-5. Figure 2 shows the percent potential air flow of intake valve in different of intake valve lift and intake air pressure. Figure 3 shows the air flow velocity of intake valve in different of intake valve lift and intake air pressure. Figure 4 shows the power engine based on intake valve in different of intake valve lift and intake air pressure. Figure 5 shows the engine speed based on intake valve in different of intake valve lift and intake air pressure.

The percent potential flow is test flow divided by potential valve flow. The Fig. 2 shows that percent potential flow in the experiment is low in low L/D ratio or low in intake valve lift. The percent potential velocity is increasing if the valve lift is increase until near maximum valve lift. In the maximum valve lift the percent potential flow is decrease because in the maximum valve lift the air flow can't increase and the trend is stabile or horizontal. So, the stabile air flow divided by potential flow in higher is give the percent potential flow is decreasing in maximum valve lift and after maximum valve lift. The higher test pressure is given the higher percent potential flow in the intake valve. So, the test pressure at 1651 mm H2O is given the higher value in the percent potential flow, the maximum percent potential flow value is 62.04% in 0.2 L/D ratio or 7.11 mm intake valve lift and the lower value of percent potential flow is 42.81% in 0.05 L/D ratio or 1.78 mm intake valve lift.

The Fig. 3 shows the flow velocity in the experiment. The value of flow velocity is low in low L/D ratio position or low in intake valve lift because, air flow in the low intake valve lift is low and become to spectacular increasing after lower position of valve lift until near the maximum valve lift. From point to point of valve lift in these steps is higher increasing in position after minimum valve lift until near maximum valve lift. In the position valve lift near the maximum valve lift, in the maximum valve lift and after maximum valve, the velocity of air flow is lower than in position in 0.15 L/D ratio or 5.33 mm valve lift. The maximum flow velocity is in the higher test pressure position. The value of maximum flow velocity is 8.05 m/s and the minimum flow velocity is 4.756 m/s in position of 0.2 L/D ratio at 1143 mm H2O test pressure.

The air flow in intake valve effect for engine power is shown in Fig. 4. The correlation of engine power and intake valve lift and the intake air pressure is positive. Increasing the valve lift and increasing the test pressure or air intake pressure can be increase the engine power. The higher valve lift is given the higher engine power and higher test pressure is given the higher engine power too. Both of increasing intake valve lift and test pressure is increase intake air flow, potential air flow. In this experiment, effect of increasing air flow and potential flow is given increasing the engine power. But, the higher air flow through engine can be increase the combustion optimum, lower un-burn fuel and higher burn fuel in engine cylinder or combustion chamber. From the Fig. 4 shows that, the maximum engine power is 15.15 kW at test pressure 1651 mm H2O and 0.25 L/D ratio and the minimum engine power is 3.230007 kW at 635 mm H2O test pressure and 0.05 L/D ratio.

The air flow effect in engine speed is shown in Fig. 5. Trend of the experiment and calculation results are shown that, the increasing test pressure and increasing L/D ratio of intake valve lift or increasing valve lift have been increase the engine speed. Based on this experiment, increasing the valve lift and test pressure is increased the air flow through engine and increase the air flow through engine is given the engine speed increase, because the increasing air flow through engine is given the combustion optimum. The value of maximum engine speed in this experiment is 3824 rpm at 1651 mm H2O test pressure or highest pressure and at 0.25 L/D ratio or at the maximum intake valve lift. The value of minimum engine speed is 861.27 rpm at 635 mm H2O test pressure and 0.05 L/D ratio.

CONCLUSION

The air flow through engine, engine power and engine speed has been successfully examined and calculated in various test pressures and valve lift in this experiment using Superflow testbench. The result shows that the increasing of air flow in the intake port can be increase the engine power and engine speed. From this experiment also, the value of engine power and engine speed has been successfully calculated and its value varies depending to test pressure and valve lift to diameter ratio. Both of he maximum value of engine power and engine speed are at the highest test pressure and the maximum valve lift.

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