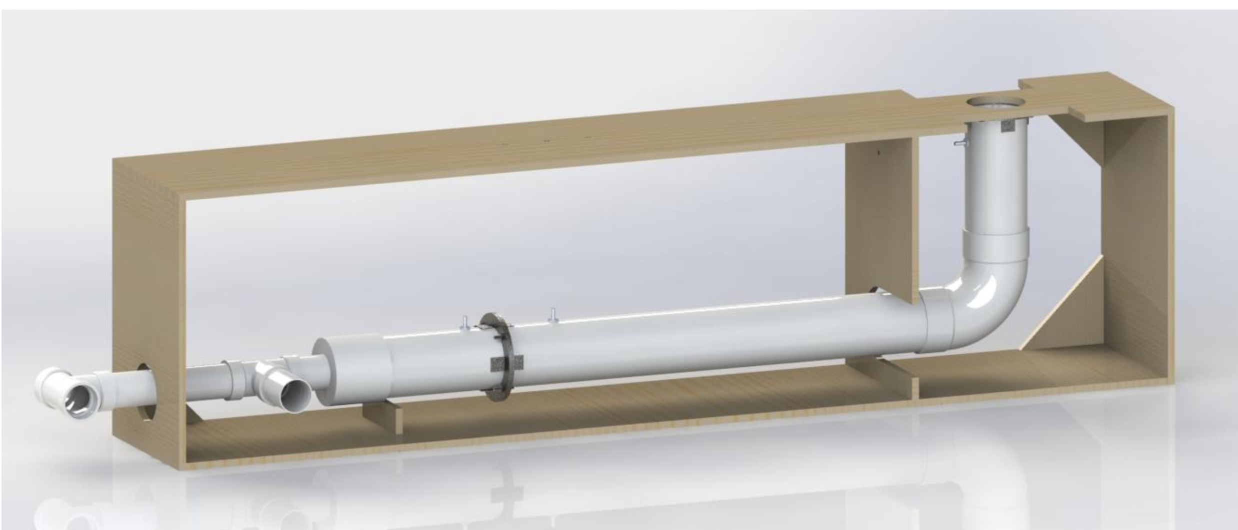
CALIFORNIA STATE POLYTECHNIC UNIVERSITY, POMONA

Design, Construction and Testing of Square-Edge Orifice Meter Flow Bench

For Automotive Use



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LIST OF FIGURES 2

ABSTRACT 3

CHAPTER 1: INTRODUCTION 4

1.1 Flow Bench 4

1.2 Concentric Square-Edged Orifice Meter 4

CHAPTER 2: THEORY 5

2.1 Flow Bench 5

2.2 Square-Edged Orifice Meter 7

CHAPTER 3: DESIGN 11

CHAPTER 4: FABRICATION AND CONSTRUCTION. 13

4.1 Orifice Plate and Flanges 13

4.2 Cabinet 14

4.3 Piping 14

4.4 Straightening Vane 15

CHAPTER 5: TESTING 16

5.1 Test Prep 16

5.2 Testing Procedures 17

CHAPTER 6: RESULTS 18

CHAPTER 7: DISCUSSION OF RESULTS 21

CONCLUSIONS 22

RECOMMENDATIONS 25

REFRENCES 26

NOMENCLATURE 28

APPENDIX A: Bill of Materials and Drawings 29

APPENDIX B: Excel Spreadsheet Used for Data Reduction 35

APPENDIX C: Sample Calculations 36

APPENDIX D: Figures 39

APPENDIX E: Original Data Sheet 42

APPENDIX F: Pettis Performance Data Sheet 43

LIST OF FIGURES

Figure 1, General Flow Bench Layout [1] 5
Figure 2, Flow Rates of Two Runs on Project Flow bench 20
Figure 3, Reference Flow Data [13] 21
Figure 4, Comparison of Averaged Project Data vs. SF-600 Data 24
Figure 5, Top Panel Drawing 31
Figure 6, Left Panel Drawing 32
Figure 7, Mid Support Drawing 33
Figure 8, Tab Flange Drawing 34
Figure 9, Rough Cut Orifice Plate 39
Figure 10, Cleaned up Orifice Plate 39
Figure 11, Pressure Tap 40
Figure 12, Head Set Up with Putty 40
Figure 13, Head Set Up with Valve Depressing Apparatus 41
Figure 14, Completed Flow Bench 41

ABSTRACT

This report details the design and fabrication of an affordable flow bench for automotive use and the analysis of the acquired data. An overall judgment based on accuracy and price between the flow bench discussed in the report and a commercially available flow bench is made finding that affordable flow bench data closely matches the data obtained from a much more expensive commercially available unit (i.e. a SuperFlow SF-600).

CHAPTER 1: INTRODUCTION

This project is rather limited in objectives: Firstly, to design and build a flow bench utilizing a square-edge orifice meter following recommendations/guidelines from *Fluid Meters: Their Theory and Application, 6th ed.* Secondly, that the constructed flow bench be affordable and reasonably accurate when compared to commercially available flow benches.

1.1 Flow Bench

Flow bench is a term used to describe devices used primarily in the automotive industry to determine the volumetric flow rate through various induction components (i.e. cylinder heads, intake manifolds, carburetors, etc.). The volumetric flow is calculated at various valve lifts so as to generate a graph that can show improved flow for modified components, differences in components from various manufacturers, and even aid in cam shaft selection; i.e. if a cylinder head flows very well between certain valve lifts, a camshaft that will open or keep valves open in that range can be selected or designed to gain maximum horse power. The amount of vacuum a flow bench can create is generally the determining factor is size and cost of commercial flow benches.

1.2 Concentric Square-Edged Orifice Meter

Concentric square-edged orifice meters are rather straight forward in design and function. The name describes them very well: a plate with an orifice in it smaller than the diameter of the main pipe will generate a difference in pressure which can then be used to calculate the mass flow rate or volumetric flow rate based on conservation of mass principles.

CHAPTER 2: THEORY

2.1 Flow Bench

Flow benches rely on differential pressure metering devices to calculate the volumetric flow (predominately orifice meters). These tests are typically conducted at constant test pressures; testing at a constant pressure (also known as depression since it below atmospheric pressure) allows for comparisons to be made between various components that may be tested on different benches. The most commonly used test pressures are -10 and -28 in.H2O; however, the greater the test pressure, the greater the volumetric flow calculated and the more accurate the profile of valve lift vs. volumetric flow. Figure 1 shows a generalized layout typical of most flow benches.

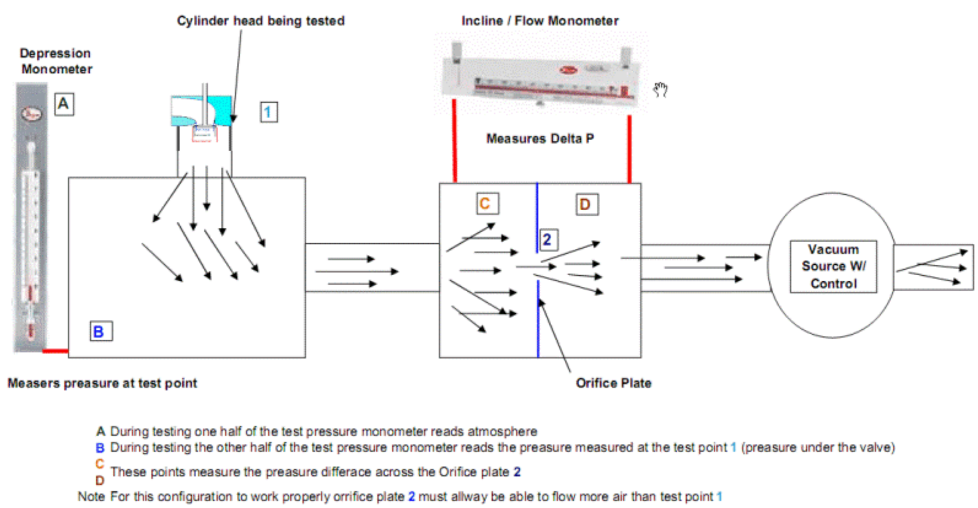

Figure 1, General Flow Bench Layout [1]

In the above diagram, the flow control is used to maintain a constant test pressure and a u-tube manometer to measure it. The differential pressure is measured across the metering element using an inclined manometer. For flow benches that use an inclined manometer to measure differential pressure, they need to have a calibrated orifice plate(s) associated with the max differential pressure of the inclined manometer. Typically, a very simplistic equation is used for calculations:

$$\dot{V} = \sqrt{h} * K * A \textbf{ (2.1.1)}$$

Where, h is the differential pressure in in.H2O, K is the combination of coefficients and A is the area of the orifice.

$$K = \sqrt{\frac{2g_c\rho_{h2O}}{12^{in}/_{ft} * \rho_{air}}} * C * \frac{60^{sec}/_{min}}{144^{in^2}/_{ft^2}} \textbf{ (2.1.2)}$$

Here, C is the discharge coefficient and g_c is the gravitational constant. By solving for K, using the appropriate density values and usually assuming a C value of .62, one can then plug this value into equation 2.1.1, along with the max differential pressure of their inclined manometer and the area of an orifice, determine the max volumetric flow the plate is capable of flowing. Conversely, a $\dot{V}$value can be chosen and then an orifice size determined.

One downside to this type of flow bench is the limited capacity of the inclined manometer; if the max differential pressure the manometer can handle is 6in.H2O then the orifice plates must be designed to flow a specific cfm at that differential pressure, or else the water will be sucked out of the manometer. Now, if an orifice plate is designed to flow 200 cfm at 6in.H2O ΔP then that is the largest volumetric flow rate that can be measured and any reading from the inclined manometer is simply a percentage of the max cfm rating for the plate. The flow bench designed for this project, however, does not rely on an inclined manometer and thus can use one orifice plate. The limiting factors are simply the amount of air the vacuum motors can move, and the operating range of the Pyle digital manometer.

2.2 Square-Edged Orifice Meter

Some general assumptions that are made in developing the equations used to analyze square-edged orifice meters are as follows: First, the pipe section is horizontal so that the effect of gravity is the same at all sections. Second, in flowing from section *A* to section *a,* the fluid performs no external work. Third, the flow is steady and axial, and the velocity profile at each section is relatively flat and normal to the pipe section. Fourth, that no transfer of heat between the fluid and the pipe takes place, which implies no friction, permits assuming and change of state between *A* and *a* is adiabatic [2].

The principle equation used for determining the mass flow rate through the orifice is essentially what is referred to as the "hydraulic" equation multiplied by an expansion factor *Y*; where β is the ratio of diameters, d/D.

$$\dot{m} = a\sqrt{\frac{2g_C\rho_1(p_1-p_2)}{1-\beta^4}} * Y \quad \textbf{(2.2.1)}\ [3]$$

However, equation 2.1 is purely theoretical and needs to be multiplied by an additional factor and coefficient in order to make it realistic. The factor required is known as the thermal expansion factor F_a. The thermal expansion factor is used to take into account any change in the area of the orifice due to thermal expansion, and is found in *Fluid Meters* [4].

The second of the coefficients is the discharge coefficient *C*. *C* is defined as follows:

$$C = \frac{Actual\ rate\ of\ flow}{Theoretical\ rate\ of\ flow} \quad \textbf{(2.2.2)}$$

The purpose of the discharge coefficient is to take into account the head loss that occurs post metering element. The swirling and turbulent air is unpredictable and thus the true head loss cannot be calculated theoretically. *C* is a function of the ratio of diameters (β) as well as the Reynolds number (R_d). When C and F_a are applied to (2.1) it then becomes:

$$\dot{m} = a * F_a * \left(\frac{C}{\sqrt{1-\beta^4}}\right) * \sqrt{2g_c(p_1 - p_2)} \textbf{(2.2.3)}\ [5]$$

Where $C/\sqrt{1-\beta^4}$ is typically replaced by *K* (this term will be defined and implemented in the mass flow rate equation towards the end of this section). With p_1 and p_2 in psia, T_1 degree R, and ρ_1 in lb_m per cubic foot, the final mass flow rate equation can be created:

$$\dot{m}\left(\frac{lb_m}{sec}\right) = \frac{d^2CY}{576}\left(\frac{F_a}{\sqrt{1-\beta^4}}\right)\sqrt{2 * 144ft^2 * 32.174\frac{ft}{sec^2} * \rho_1(p_1 - p_2)} = 0.52502\left(\frac{CYd^2F_a}{\sqrt{1-\beta^4}}\right)\sqrt{\rho_1(p_1 - p_2)}$$

(2.2.4) [6]

Because the differential pressure measured for this flow bench is measured in inches of H2O;

$$(p_1 - p_2) = h_w(in)\frac{62.3164\frac{lb_m}{ft^3}}{1728\frac{in^3}{ft^3}} = h_w * 0.03606 \textbf{ (2.2.5)}$$

Finally,

$$\dot{m}\left(\frac{lb_m}{sec}\right) = 0.099702 * \left(\frac{CYd^2F_a}{\sqrt{1-\beta^4}}\right) * \sqrt{\rho_1 h_w} \textbf{ (2.2.6)}\ [7]$$

In attempting to be completely thorough and accurate, the equation used to calculate the density of the air, ρ_1, was equation (I-3-40) on page 30 of *Fluid Meters*.

$$\rho_m = 2.6991(1 + W)\frac{p - p_v}{TZ}G \textbf{ (2.2.7)}\ [8]$$

p_d= Partial absolute pressure of dry gas in the moist gas mixture

p_v= Partial absolute pressure of the water vapor in the moist gas mixture.

p= Total pressure of the moist gas= p_d+p_v

W= Specific humidity

T= Temperature, ^{o}R

Z= Compressibility factor

G=Specific gravity ratio

This equation is meant to find the density of moist gas (ρ_m); the atmospheric ambient air that is used with flow benches doesn't really require this level of robust equation, but using it allows for a more capable spreadsheet.

The next equation that needs to be discussed is the equation for Y, the expansion factor. *Fluid Meters* requires that the following equation be used, when ρ_1 is calculated using inlet temperature and pressure, and the inlet pressure tap is located pre-metering element [9]:

$$Y = 1 - (0.410 + 0.350 * \beta^4) * \left(\frac{x}{\gamma}\right) \quad \textbf{(2.2.8)}$$

Noting that $x = (p_1 - p_1)/p_1$ and γ= 1.4 for air.

Y is referred to as the "net expansion factor" and is introduced to take into account the effects of expansion as an expansible fluid flows through an orifice and was developed empirically.

The following sets of equations are required to determine the flow coefficient *K* for D&1/2D pressure taps [10], where:

K= Flow coefficient corresponding to any specific set of values of D, β, and R_d

K_o= The limiting value of K for any specific values of D and β when R_d becomes infinitely large.

C=K/$\sqrt{1-\beta^4}$

$$K = K_o + b\lambda \quad \textbf{(2.2.9)}$$

Where,

$$K_o = \left(0.6014 - 0.01352 * D^{\frac{-1}{4}}\right) + \left(0.3760 + 0.07257 * D^{\frac{-1}{4}}\right)\left(\frac{0.00025}{D^2\beta^2 + 0.0025D} + \beta^4 + 1.5 * \beta^{16}\right)$$

(2.2.10)

$$b = \left(0.0002 + \frac{0.0011}{D}\right) + \left(0.0038 + \frac{0.0004}{D}\right) \times [\beta^2 + (16.5 + 5 * D) * \beta^{16}] \quad \textbf{(2.2.11)}$$

$$\lambda = \frac{1000}{\sqrt{\beta * R_d}} \quad \textbf{(2.2.12)}$$

and the Reynolds number for the orifice is found with:

$$R_d = \frac{48 * \dot{m}}{\pi * d * \mu} \quad \textbf{(2.2.13)}$$

Through an iterative process, a guess value for the discharge coefficient is chosen (0.62 is typically a good starting point) and a guess value of $\dot{m}$ is obtained. This guess value of mass flow rate is then used to calculate the Reynolds number for the orifice (R_d). From here, the value of λ can be found and used to calculate the value of K. K can then be used to find a new, more accurate value of C using the relation C=K/$\sqrt{1-\beta^4}$. The new value of C is then used to find a new mass flow rate and the process is repeated until the new value of C is very close to the previously used value. The final value of C is used to find the final value of $\dot{m}$ which when divided by the density of the air and multiplied by 60 sec/min yields the volumetric flow rate through the orifice in cubic and thus the cylinder head in feet per minute.

CHAPTER 3: DESIGN

A concentric square-edge orifice plate was chosen as the metering element primarily for ease of home manufacture. Granted, the edge condition and concentricity of the orifice plate is critical to obtaining accurate readings, the ease of manufacture means multiple plates can be made quickly if any issues arise. A venturi meter would provide more consistent readings, but the difficulty of manufacture is the limiting factor.

The overall dimensions of the flow bench piping was based on Fig II-II-1 (G) "Recommended Minimum Lengths of Pipe Preceding and Following Orifices, Flow Nozzles and Venturi Tubes" located in *Fluid Meters* (6^{th} ed.) [11]. Based on these recommendations, with 4" PVC sewage pipe as the main tubing for the flow bench and a β of .5 or less, the dimensions preceding a straightening vane was found to be twice the diameter of the pipe (8"), the straightening vane as twice the diameter of the pipe (8"), length of pipe following the vane and preceding the orifice plate as roughly six times the diameter of the pipe (24"), and the length of pipe following the orifice plate as roughly four times the pipe diameter (16").

The material selected for the orifice plate and flanges was 1/8" mild steel [12]. The instrumentation selected for the flow bench was a Pyle PDMM01 digital manometer. This device was chosen because the manufacture of an accurate and calibrated inclined manometer would have been time consuming and most likely resulted in larger error.

The cabinet of the flow bench was designed purely for simplicity and practicality; thus ½" plywood was chosen as the material and the rectangular notches on either side of the intake port are simply to allow for easier clamping of the work piece. The straightening vane was, like most of this project, designed with minimum cost in mind and regular drinking straws were selected as the vane material and glued together with spray adhesive; the vane his held in place by a small metal brad protruding from the outside of the pipe in to ensure the vane stays put. The 4”x2” reducer was used to allow shop-vacs with 2” diameter hoses to be attached and used as the vacuum source. The 2” ball valve located after the reducer is to adjust the test pressure. Pressure tap locations were chosen to be D&½ D. This choice was made because the flange material is too thin to accommodate flange taps, and over the central range of β ratios, the difference between D&½ D taps and vena contracta taps is negligible [13]. The taps themselves were chosen to be ¼” NPT male x 3/16” barbed fittings.

Detailed drawings of key flow bench parts, as well as a bill of materials, can be found in APPENDIX A.

CHAPTER 4: FABRICATION AND CONSTRUCTION.

4.1 Orifice Plate and Flanges

All pieces were cut from the same piece of 12”x24” piece of 1/8” mild steel. The pieces were cut by drilling a 1/8” locating hole in the center and using a tracing tool with a plasma cutter pivoting around this hole and a 1/8” drill bit acting as a pin. Once all the flanges and orifice blanks were cut, they were aligned using the 1/8” center hole and tacked together so that the outer edges could be cleaned up and bolt holes drilled to ensure that the holes would line-up with each other.

Due to the inaccuracy of their manufacture, a clocking notch was cut into all the pieces after the holes were drilled and the tack welds broken (bolts were used to obtain proper alignment) to ensure that the orifice plate(s) would bolt between the flanges accurately and be concentric with the pipe. After the plates had been separated from each other and notched, locating tabs were welded onto the flange rings. A piece of 4” PVC pipe was used to ensure concentricity with the flange ring while attaching the tabs. The orifice plate orifice was cut using a 2” hole-saw. The plate blank was screwed to a piece of 2”x4” and clamped down to the drill press work surface. The work piece was leveled to reduce the inaccuracy of the cut. Unfortunately, the hole-saw proved to be rather crude and resulted in a jagged and uneven orifice as seen in Figure 9. To remedy this issue and create an appropriate edge that would result in accurate data, a Dremel tool and grinding stone were used to remove and rough features and make the edge of the orifice as square as possible. The “cleaned up” orifice can be seen in Figure 10. The flanges on either side of the orifice plate, once attached to the 4” PVC, were given a small bead of silicone to create a gasket and prevent leaks.

4.2 Cabinet

That cabinet was simply cut from a 4’x8’x1/2” piece of plywood using a table saw and hole saws were necessary to cut the required openings. The cabinet was assembled using brads and wood glue to ensure adequate bonding and strength.

4.3 Piping

The PVC piping was simply cut to length, according to the designs, and the edges of the cuts were cleaned of all rough and haggard material to reduce any flow restrictions or mating issues. The flanges were attached to the appropriate ends of the pipes using general purpose white caulk manufactured by DAP. The inside edge were the pipe and flange meet was also given a thin coating of caulk to prevent any leaks.

The pressure tap holes were drilled at the appropriate locations and taped using a ¼” NPT tap. Once fitted, the bottom of the nylon taps were ground down flush with the inner surface of the pipe using a Dremel tool to reduce and unwanted turbulence that would affect readings. Note: the excess caulk seen on the inside surface of the pipe in Figure 11 was removed before final assembly and testing.

The 90^{o} elbow was glued to the 12” vertical section of PVC using ABS/PVC cement, the ABS 4”x2” reducer was also the 12” post meter PVC pipe using the same adhesive. The 36” piece of PVC was not glued to the elbow but instead fitted into the elbow which was already a tight fit, and sealed using a piece of string around the circumference along with more caulk. This was done not only to ease attaching the vertical piece to the underside of the work surface, but in case the straightening vane ever needs to be replaced or modified, the string can be removed, the caulk along with it, and the piece can be removed from the elbow.

4.4 Straightening Vane

The manufacture of the straightening vane is rather straight forward, the straws were glued together in a rough jig to get a fairly consistent size and then then other straws were attached as necessary to fill out the dimensions and fit the PVC snuggly. A spare brad left over from constructing the cabinet was used to secure the van in the pipe; because the brad passed through the pipe wall, caulk was used to seal the brad in position.

CHAPTER 5: TESTING

5.1 Test Prep

The head used for testing was a stock Chevrolet 350 head, casting numbers 3782461. The head was first cleaned with a Demel tool and a stainless steel wire-wheel attachment. The head surface and combustion chamber was cleaned of all carbon deposits and debris in order to reduce likelihood of debris being sucked into the flow bench and to promote sufficient sealing with the flow bench surface. The intake port to be tested was also cleaned of carbon buildup using the Dremel tool and carbon build up on the intake valve was removed as well, all this was done to prevent debris being sucked in and to achieve reliable results; i.e. carbon build up would cause inaccuracies in the data and data would be different from results obtained on professional flow bench which requires very clean test pieces.

The stock valve spring was replaced with a low tension test spring thus allowing for easy manipulation of the valve during testing. A piece of .120” wall 1”x1” steel tubing was drilled and fitted to one of the rocker studs and secured with nuts and tie-wire. The square tubing provided a surface to mount the magnetic base of the dial gauge as well as providing a leverage point for adjusting the valve height Figure 13.

The head was then centered over the intake of the flow bench and plumber’s putty was used to create a seal between the head and the test surface and clamped down using four C-clamps. Plumber’s putty was also applied around the intake port to be tested and secured with blue painter’s tape, Figure 12. The putty reduces the turbulence otherwise created by sucking air through the head and the sharp edge of the intake port; the tape keeps the putty from being sucked into the head itself. The two shop-vacs ere also attached to the 2” Tee fitting at the end of the flow bench.

5.2 Testing Procedures

The testing procedures are rather straight forward. The Pyle digital manometer was first manually calibrated to atmospheric pressure using the built-in feature. With both vacuums running the valve operator depressed the valve to the maximum valve height to be tested, 0.6”. With the 2” ball valve completely closed and the digital manometer attached to the test pressure port, the minimum test pressure value of -15.5 in.H2O was determined.

From this point on, the desired valve height (i.e. .05”, .1”, etc.) was obtained by leveraging the valve down using a long screwdriver by the valve operator; the data collector then attached the Pyle digital manometer to the test pressure port and adjusted the test pressure via the 2” ball valve to the desired -15.5 in.H2O. Once the ball valve had been adjusted, the intake valve was closed and the test pressure port line closed using a golf tee. The digital manometer was then connected to the metering pressure tap lines (previously sealed with golf tees). At this point with the intake valve closed, the differential pressure across the orifice plate is zero. The intake valve was closed for this pressure tap switch because earlier tests were the intake valve was held open, and the pressure tap switch made, resulted in the digital manometer re-calibrating itself to make the differential pressure reading its new “zero”, thus ruining the data.

With the manometer connected, the valve operator opened the intake valve to the previous height at which the ball valve had been adjusted for, and the differential pressure was observed on the manometer and manually written down. This process was repeated throughout the desired valve height range (0.05-0.6” in increments of 0.05”) and two complete sets of data were obtained back-to-back.

CHAPTER 6: RESULTS

The data obtained was then entered into an Excel spreadsheet that incorporates the equations derived and listed in *Fluid Meters*. A screen shot of the spreadsheet can be found in Appendix C and the step-by-step calculations made in the spreadsheet are shown in Appendix D for one valve height. During the acquisition of data there was a fair amount of oscillation in the displayed pressures on the digital manometer. Typically the pressure would fluctuate by roughly +/- 0.2 in.H2O and the values recorded as data were more or less the observed average. Over the two runs, the data is fairly consistent; most data points vary by only 0.1 in.H2O, the exception being the final three test points which vary by as much as 0.3 in.H2O at 0.5 inches of valve lift. The exact reason for this is unknown, but being that it occurred at the upper end of valve lift with the highest volumetric flow rates there may have been a little too much turbulence around the pressure taps.

After the recorded data was entered into the spreadsheet, the following table was produced:

Table I

Valve Lift (in.)	ΔP (inH2O) (Run 1)	ΔP (inH2O) (Run 2)	$\dot{V}$(cfm) (Run 1)	$\dot{V}$ (cfm) (Run 2)
0.05	0.2	0.3	27.5	33.6
0.10	0.6	0.6	47.3	47.3
0.15	1.3	1.4	69.3	71.9
0.20	2.4	2.4	93.9	93.9
0.25	3.6	3.5	114.8	113.2
0.30	4.8	4.7	132.3	130.9
0.35	5.6	5.5	142.8	141.5
0.40	6.2	6.2	150.1	150.1
0.45	6.6	6.5	154.8	153.6
0.50	6.3	6	151.3	147.7
0.55	6.4	6.2	152.5	150.1
0.60	6.2	6	150.1	147.7

The volumetric flows were then plotted against the valve height producing Figure2.

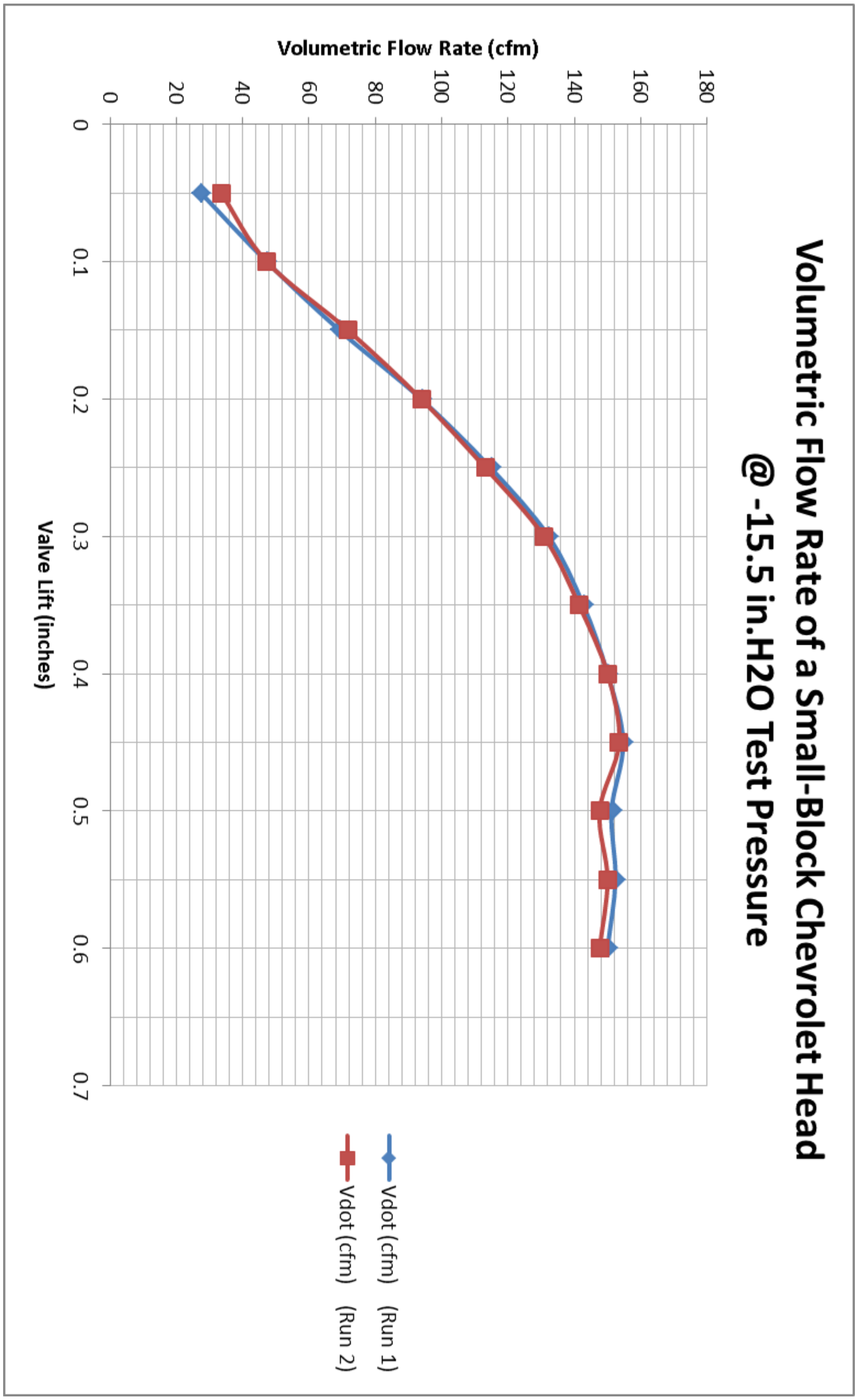

Figure 2, Flow Rates of Two Runs on Project Flow bench

CHAPTER 7: DISCUSSION OF RESULTS

The results obtained from the two runs seem to be right in line with data published in other sources. The increase in volumetric flow throughout most of the valve lift range and the volumetric flow reaching a peak before maximum valve lift and decreasing slightly afterward correlates well published data on a variety of heads. In David Vizard's *How to Port & Flow Test Cylinder Heads*, he published a graph comparing the volumetric flow, as calculated by Audie Technologies Flow Quick instrumentation, of a Holley 23-degree, high performance street small-block Chevrolet head.

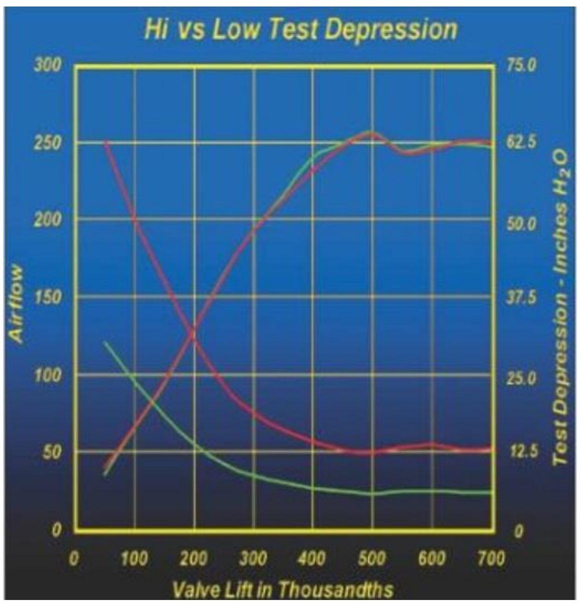

Figure 3, Reference Flow Data [13]

Noting that the head Mr. Vizard tested is a performance head, thus meaning it will have higher volumetric flow than the head tested in this report, but the characteristic rise, mild plateau, and even dip around 0.55" and 0.6" of lift looks remarkably similar to the graph of the data obtained on my flow bench, indicating that my results are, on brief inspection, in the ballpark.

CONCLUSIONS

The same head and intake port was tested by Pettis Performance in Hesperia California using a SuperFlow SF-600 flow bench; the data obtained from their testing at test pressure of 15 inH2O are listed in Table II as well as the averaged cfm values from the project flow bench and the percent difference between the two. Figure 3 shows the averaged flow rate plotted against the SF-600 flow rates from 0.1 to 0.6 inches of valve lift in 0.1 inch increments. By finding the percent difference between the data obtained using the flow bench and the SF-600 data, it was found that at lower valve lifts the percent error is around 6 and this error decline as valve lift increases ultimately resulting in a percent error of less than one at 0.5 and 0.6 inches of valve lift. These results are better than expected; despite the 6% error at the lower valve lifts, this error is only results in differences of 3-7 cfm, which for a non-racing engine is negligible. The error is most likely less than what the math shows because the tests performed on the project flow bench were conducted at a test pressure of 15.5 in.H2O and the SF-600 data was obtained a test pressure of 15 in.H2O; the slightly higher vacuum of 15.5 would result in slightly higher flow rate numbers, which is observed in then data.

In conclusion, the accuracy is more than enough justification to build the flow bench detailed in this project for around $140 as opposed to commercially available units like the SF-600 which can be sold second hand for between $2500 and $4000 [15].

Table II

Valve Lift (in.)	$\dot{V}$ avg, cfm (project)	$\dot{V}$, cfm SF-600	% diff
0.1	47.3	44.6	6.053811659
0.2	93.9	88.5	6.101694915
0.3	131.6	124.5	5.702811245
0.4	150.1	146	2.808219178
0.5	149.5	149	0.33557047
0.6	148.9	149	-0.06711409

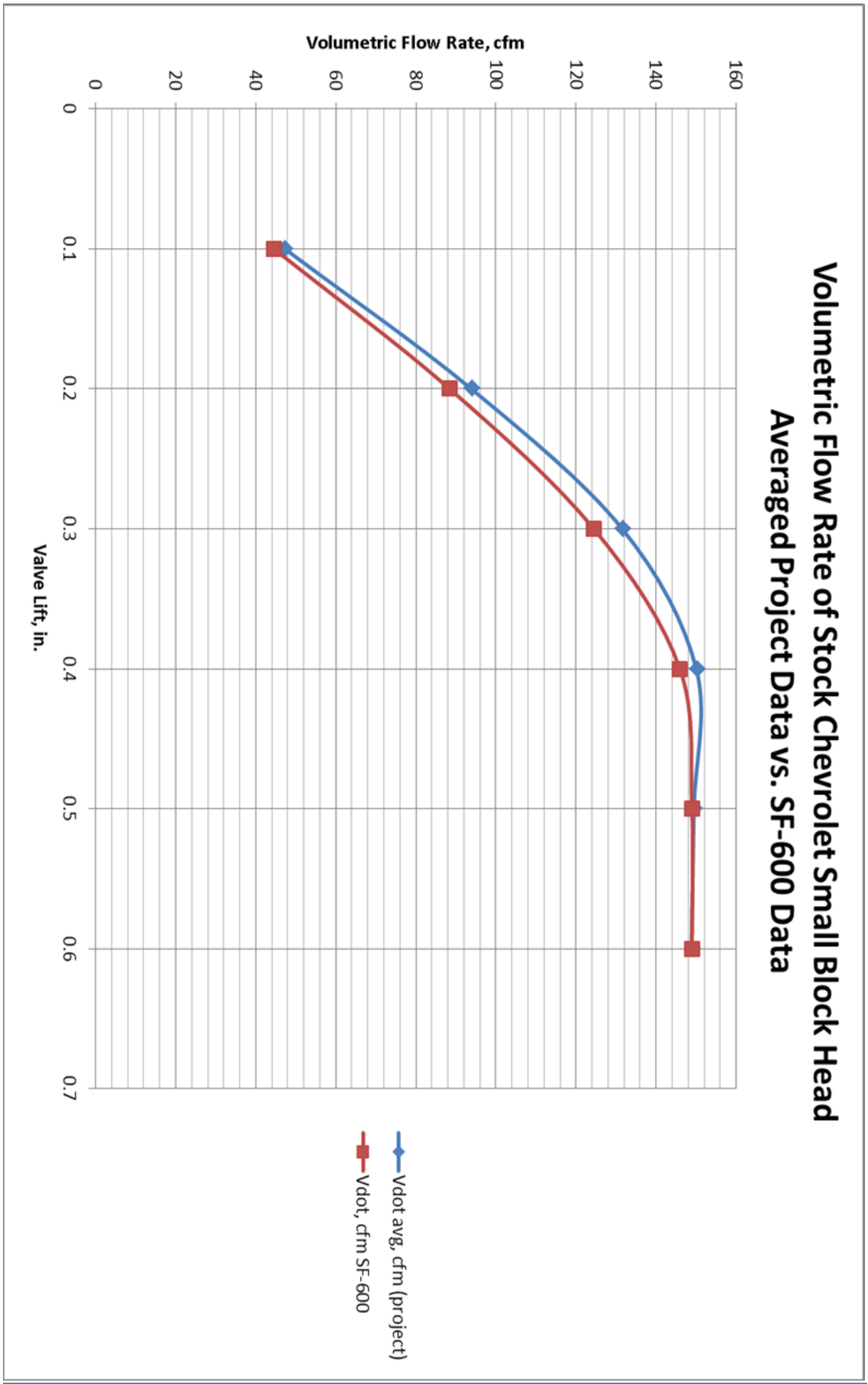

Figure 4, Comparison of Averaged Project Data vs. SF-600 Data

RECOMMENDATIONS

Some issues with the flow bench that should be addressed include: the fabrication of steel parts, pressure tap internal diameters, and vacuum source and pressure sensors. The steel parts made (flanges and orifice plate) should be precision cut in any future version of this flow bench with emphasis placed on the orifice plate. Laser or water jet cut flanges and orifice plate would ensure proper concentricity with the flow bench piping and have a much better surface finish on the cuts which is critical (for the orifice plate) in obtaining truly accurate data. *Fluid Meters* recommends that the center of the orifice should be close to 1/32" of the center of the pipe [14]. The pressure taps used had rather large internal diameters; although the taps were made to be flush with the inner diameter of the pipe, there could still be turbulence or pressure pulses inside the taps which resulted in the +/- 0.2 in.H2O oscillations seen on the digital manometer. The two shop-vacs used for this project aren't capable of as much flow as commercial benches; together they were only able to achieve -15.5 in.H2O at full valve lift for a stock cylinder head, just over half of what commercial benches can achieve for much more performance oriented cylinder heads. Lastly, the digital manometer was a great alternative to either making and calibrating an inclined manometer, or buying a commercially available unit, but if price was not an issue, electronic pressure transducers would result in much more accurate data.

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NOMENCLATURE

Symbol	Description	Units
a	Area of an orifice	in^2
C	Coefficient of Discharge	ratio
D	Diameter of pipe or meter tube	in
d	Diameter of orifice	in
E	Velocity of approach factor $=1/\sqrt{1-\beta^4}$	number
F_a	Area thermal expansion factor	ratio
g_c	Proportionality constant in the force-mass-acceleration equation=32.147	number
h_w	Effective differential pressure	in. H2O
K	Flow coefficient = CE	ratio
$\dot{m}$	Mass flow rate	${lb_m}/{sec}$
p	Pressure absolute	psia
$\dot{V}$	Volumetric flow rate	${ft^3}/{min}$
R_d	Reynolds number based on d	ratio
T	Absolute temperature	$°R$

V	Velocity	ft/sec
x	Ratio of differential pressure to inlet static pressure = $\Delta p / p_1$	ratio
Y	Expansion factor for a gas	ratio
Z	Compressibility factor for a real gas	ratio
β	Ratio of diameters = d / D	ratio
γ	Ratio of specific heats for a gas (ideal) = c_p / c_v	ratio
Δp	Differential pressure = p_1-p_2	psi
λ	A Reynolds number reciprocal = $1000/\sqrt{\beta R_d}$	ratio
μ	Absolute viscosity of a fluid	$lb_m / ft * sec$
ρ	Density	lb_m / ft^3

APPENDIX A: Bill of Materials and Drawings

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.	PRICE/ UNIT	PRICE TOTAL
1	Flowbench bottom panel	1/2"x12"x72"	1	$3.75	$3.75
2	Flowbench left panel	1/2"x12"x18"	1	$0.94	$0.94
3	Flowbench right panel	1/2"x12"x18"	1	$0.94	$0.94
4	Flowbench gusset	Cut from left over plywood, 6" triangle	4	$0.08	$0.32
5	Flowbench mid support	1/2"x12"x18"	1	$0.94	$0.94
6	Flowbench top	1/2"x12"x72"	1	$3.75	$3.75
7	Flowbench tab flange		3	$0.85	$2.55
8	4" 90 degree PVC elbow		1	$8.54	$8.54
9	2" PVC pipe piece	SCH 40, 4" long	2	$0.07	$0.14
10	Flowbench main tube	32" long, 4"dia PVC Sewage Pipe	1	$0.10	$0.10
11	Flowbench post meter tube	10" long, 4"dia. PVC Sewage Pipe	2	$0.10	$0.20
12	4"x2" ABS reducer reducer		1	$12	$12.00
13	2" PVC Tee Tee - 4880K48	SCH 40	2	$2.77	$5.54
14	Flowbench 2.05 in orifice		1	$0.85	$0.85
15	2" PVC pipe piece	SCH 40, 10" long	1	$0.07	$0.07
16	CR-PHMS 0.25-20x1x1-N		4	$0.15	$0.60
17	MSHXNUT 0.250-20-S-N		4	$0.15	$0.60
18	Flowbench pipe support	Cut from left over plywood,1.75"x6"	1	$0.05	$0.05
19	Pressure Tap Fittings	Nylon 1/4" NPT male x 3/16" barbed fitting	3	$0.35	$1.05
20	HLSCREW 0.2500x0.625		4	$0.15	$0.60
21	2" PVC ball valve		1	$17	$17.00
22	PYLE PDMM01 Digital Manometer		1	$68	$68.00
23	3/16" Vacuum Line	1.67 ft per piece	3	$1.60	$4.80
24	Precision Brand M6S Micro Seal, Miniature All Stainless Worm Gear Hose Clamp, 5/16" - 7/8"		6	$0.54	$3.24
25	DAP White Caulk		1	$2.28	$2.28
		TOTAL PRICE			$138.85

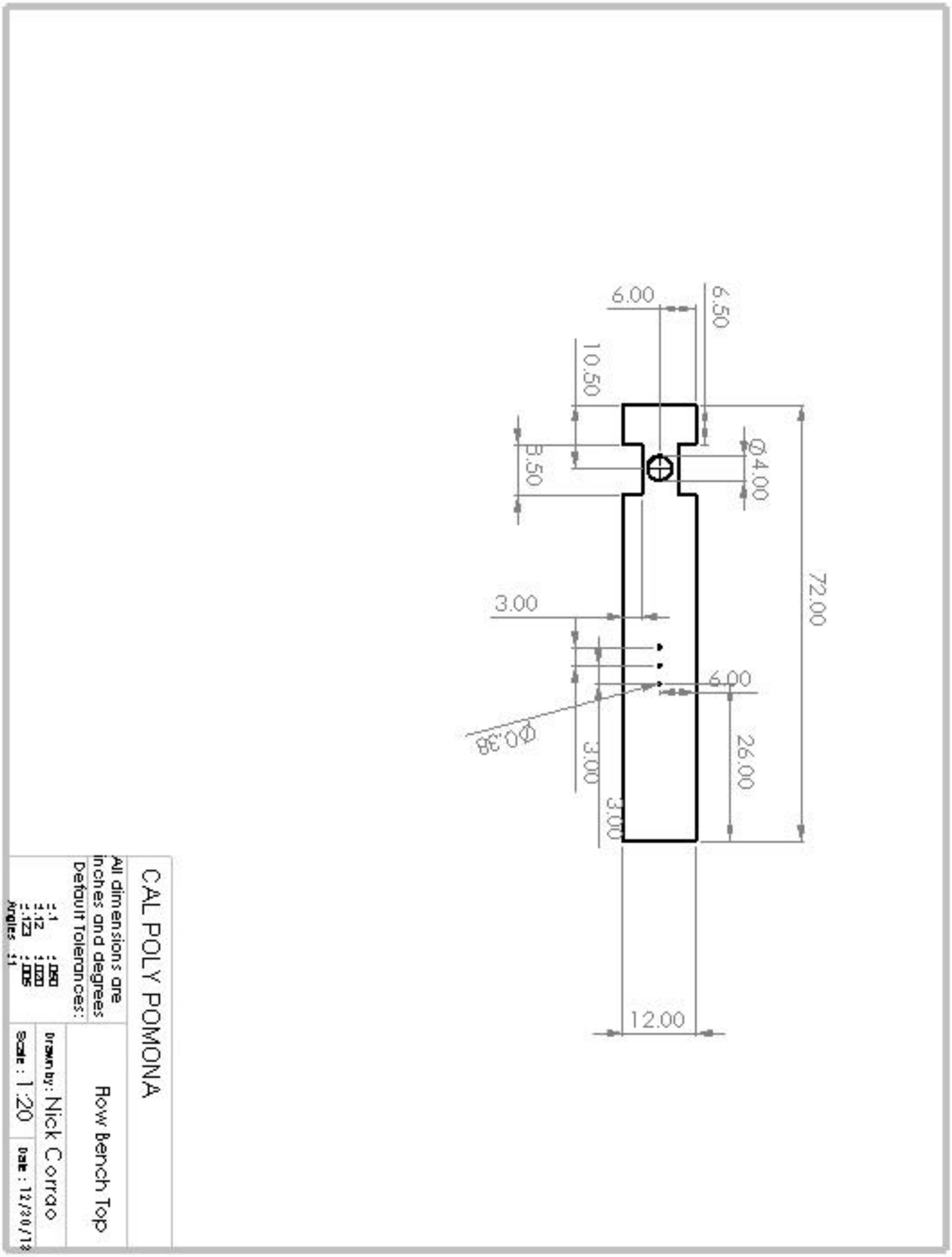

Figure 5, Top Panel Drawing

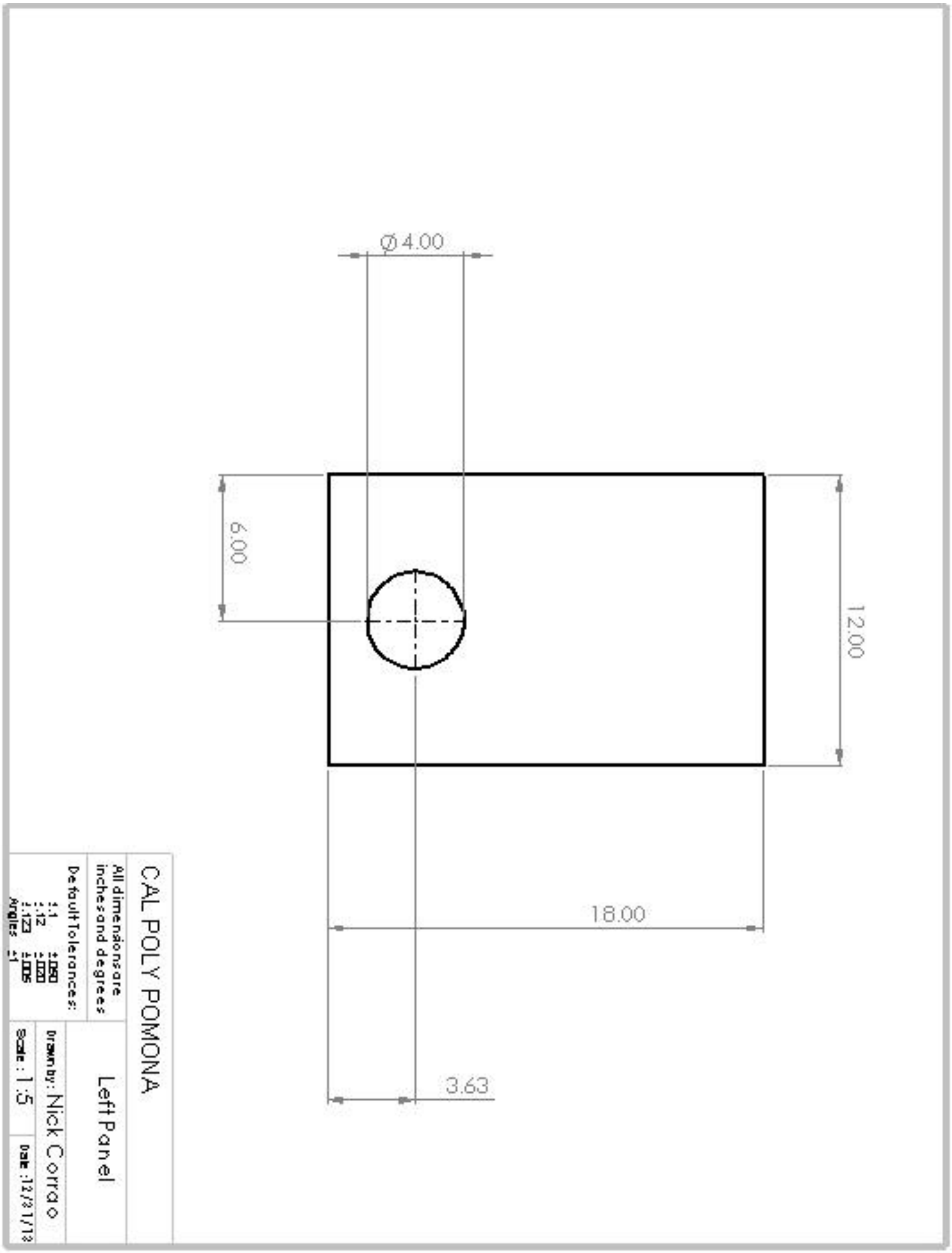

Figure 6, Left Panel Drawing

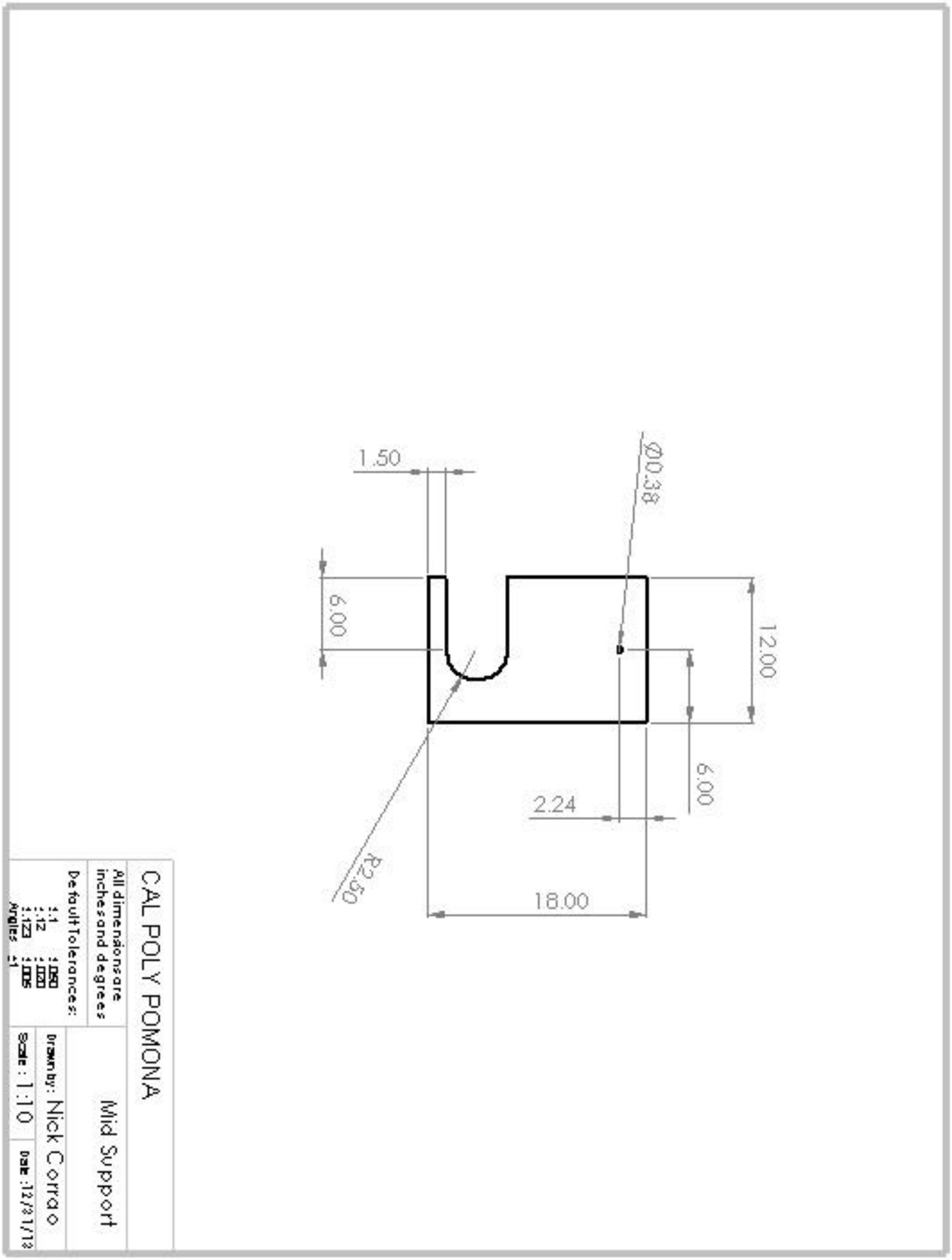

Figure 7, Mid Support Drawing

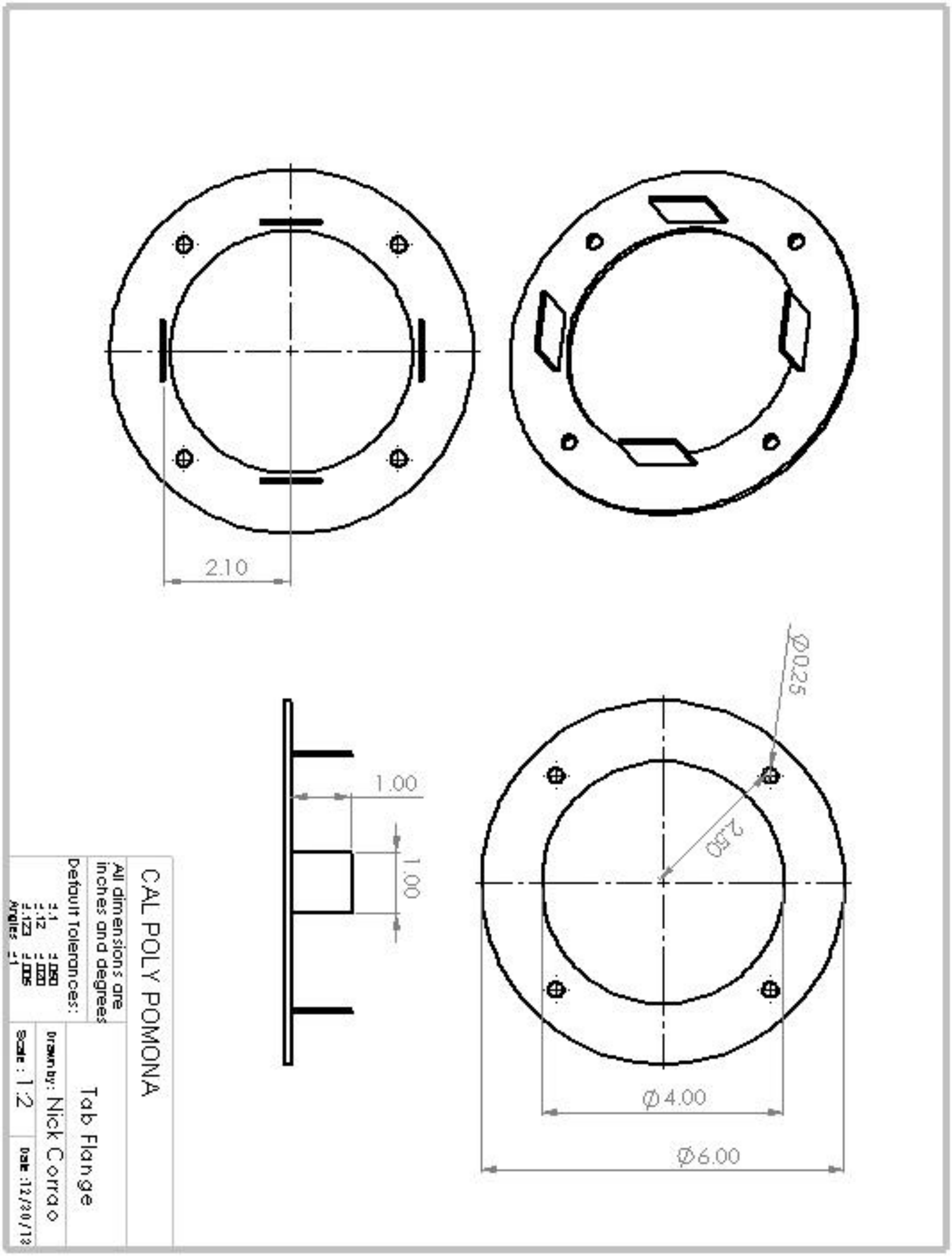

Figure 8, Tab Flange Drawing

APPENDIX B: Excel Spreadsheet Used for Data Reduction

d (in.)	2.05			
D (in.)	3.965			Entered Value
β	0.5170			Calculated value
E	1.03776			
Fa	1			
p line	13.5			
p atm (psi)	14.1			
p test (in.H2O)	-15.5			
Δp (in. H2O)	6	<===		
T (^{o}R)	531			
p sat (psi)	0.38			
x	0.0160			
γ	1.4			
x/γ	0.01141			
Y	0.995			
Z	1			
W	0.01796			
ρ (lbm/ft^3)	0.0681			
mdot guess (lbm/sec)	0.1715			
mdot new1 (lbm/sec)	0.1676			
mdot new2 (lbm/sec)	0.1676			
C guess	0.6200			
C new1	0.6060			
C new2	0.6061			
Rd guess	105619			
Rd new1	103240			
μ	0.0000121			
K guess	0.6289		K new1	0.6290
Ko	0.6224			
λ guess	4.2793		λ new1	4.3283
b	0.001523882			
q (cfm)	147.7			

APPENDIX C: Sample Calculations

Diameter of orifice; d=2.05"   Diameter of pipe; D=3.965"

$$\beta = \frac{d}{D} = \frac{2.05"}{3.965"} = 0.5170$$

$$E = \frac{1}{\sqrt{1-\beta^4}} = \frac{1}{\sqrt{1-0.5170^4}} = 1.03776$$

$$F_a = 1.000\ (for\ 0.2 - 1.1\%\ C\ Steel\ @\ 71°\mathrm{F})[4]$$

$$P_1 = P_{Test} - P_{atm} = (-15.5\ in.H2O) \times (0.03606) - 14.1\ psig) = 13.5\ psia$$

$$x = \frac{\Delta P}{P_1} = \frac{6\ in.H2O \times 0.03606}{13.5\ psia} = 0.016$$

γ= 1.4 (for air)

$$\frac{x}{\gamma} = \frac{0.016}{1.4} = 0.01141$$

$$Y = 1 - (0.41 + 0.35\beta^4) * \left(\frac{x}{\gamma}\right) = 1 - (0.41 + 0.35 * 0.5170^4) * 0.01141 = 0.995$$

$$Z = 1\ (air\ @\ near\ atmospheric\ pressure\ and\ 71°\mathrm{F})[15]$$

$$P_{sat} = .38\ psia$$

$$W = .622 \times \frac{P_{sat}}{P_1 - P_{sat}} = .622 \times \frac{.38\ psia}{13.5\ psia - .38\ psia} = 0.01796 \frac{lbm_w}{lbm_a}$$

$$\rho_1 = 2.6991 \times (1 + W) \times \frac{P_1 - P_{sat}}{T * Z} = 2.6991 \times \left(1 + .01796 \frac{lbm_w}{lbm_a}\right) \times \frac{13.5psia - .38psia}{(71°\mathrm{F} + 460) * 1}$$

$$= 0.0681\ \frac{lbm}{ft^3}$$

$$\dot{m} = .099702 \times \left(\frac{C*Y*d^2*F_a}{\sqrt{1-\beta^4}}\right) * \sqrt{\rho_1 * \Delta P} = 0.099702 \times \left(\frac{C*0.995*(2.05in)^2*1}{\sqrt{1-0.5170^4}}\right) * \sqrt{.0681\,\frac{lbm}{ft^3} * 6\ in.H2O} =$$

$$0.2766C\,\frac{lbm}{sec}$$

$$C\ value\ needs\ to\ be\ guessed, so\ .62\ is\ used$$

$$\dot{m}_{guess} = 0.2766 * .62 = 0.1715\ \frac{lbm}{sec}$$

$$\mu = 0.0000121\ \frac{lbm}{sec - ft}\ (air\ @71°\text{F})[16]$$

$$R_d = \frac{48 * \dot{m}_{guess}}{\pi * d * \mu} = \frac{48 * 0.1715\,\frac{lbm}{sec}}{\pi * 2.05\ in * 0.0000121\,\frac{lbm}{sec - ft}} = 105637$$

$$C = K/E$$

For D&1/2D taps:

$$K = K_o + b * \lambda$$

$$\lambda = \frac{1000}{\sqrt{\beta * R_d}}$$

$$K_o = \left(0.6014 - 0.01352 * D^{\frac{-1}{4}}\right) + \left(0.3760 + 0.07257 * D^{\frac{-1}{4}}\right) * \left(\frac{0.00025}{D^2\beta^2 + 0.0025D} + \beta^4 + 1.5 * \beta^{16}\right)$$

$$b = \left(0.0002 + \frac{0.0011}{D}\right) + \left(0.0038 + \frac{0.0004}{D}\right) \times [\beta^2 + (16.5 + 5 * D) * \beta^{16}]$$

Substituting in values for D, β and R_d

$$\lambda_{guess} = \frac{1000}{\sqrt{0.5170 * 105637}} = 4.279$$

$$K_0 = .622$$

$$b = 0.00152$$

$$K_{guess} = .622 + 0.00152 * 4.279 = 0.6285$$

$$C_{new\ 1} = \frac{K_{guess}}{E} = \frac{.6285}{1.03776} = .6056$$

Taking this new C value and plugging it into the mass flow rate equation;

$$\dot{m}_{new\ 1} = 0.1715 * C_{new\ 1} = 0.2766 * .6056 = 0.1675 \frac{lbm}{sec}$$

$$R_{d\ new\ 1} = \frac{48 * 0.1675 \frac{lbm}{sec}}{\pi * 2.05\ in * 0.0000121 \frac{lbm}{sec - ft}} = 103225$$

$$\lambda_{new\ 1} = 4.329$$

$$K_{new\ 1} = .622 + 0.00152 * 4.329 = .6286$$

$$C_{new\ 2} = \frac{K_{new\ 1}}{E} = \frac{.6286}{1.03776} = .6057$$

Since just a very small change occurred from $C_{new\ 1}$ to $C_{new\ 2}$, this is the value of C which will be used.

Final mass flow rate;

$$\dot{m}_{final} = 0.2766 * 0.6057 = .1675 \frac{lbm}{sec}$$

$$\dot{V} = \frac{\dot{m}_{final}}{\rho_1} * 60 \frac{sec}{min} = \frac{.1675 \frac{lbm}{sec}}{0.0681 \frac{lbm}{ft^3}} * 60 \frac{sec}{min} = \boldsymbol{147.6\ \frac{ft^3}{min}\ or\ 147.6\ cfm}$$

APPENDIX D: Figures

Figure 9, Rough Cut Orifice Plate

Figure 10, Cleaned up Orifice Plate

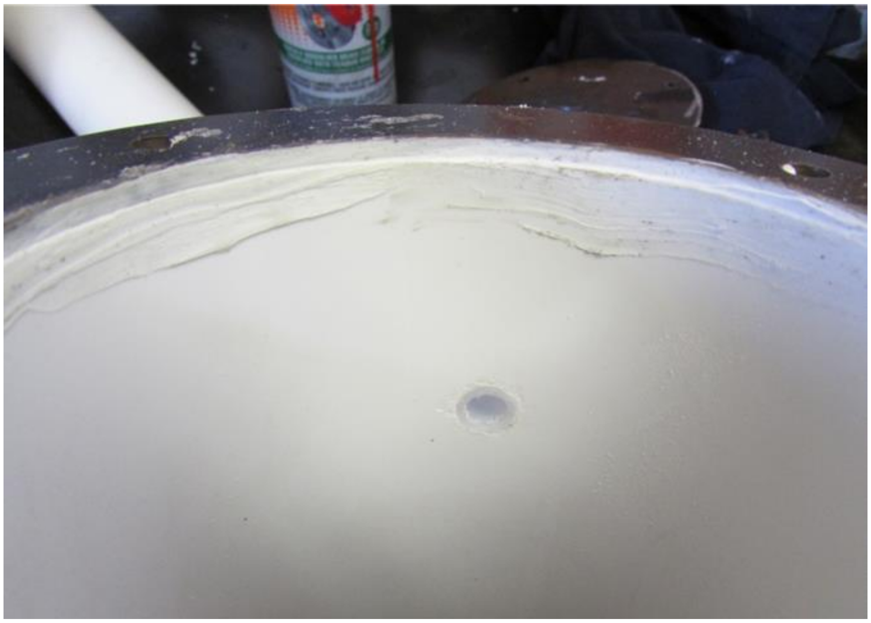

Figure 11, Pressure Tap

Figure 12, Head Set Up with Putty

Figure 13, Head Set Up with Valve Depressing Apparatus

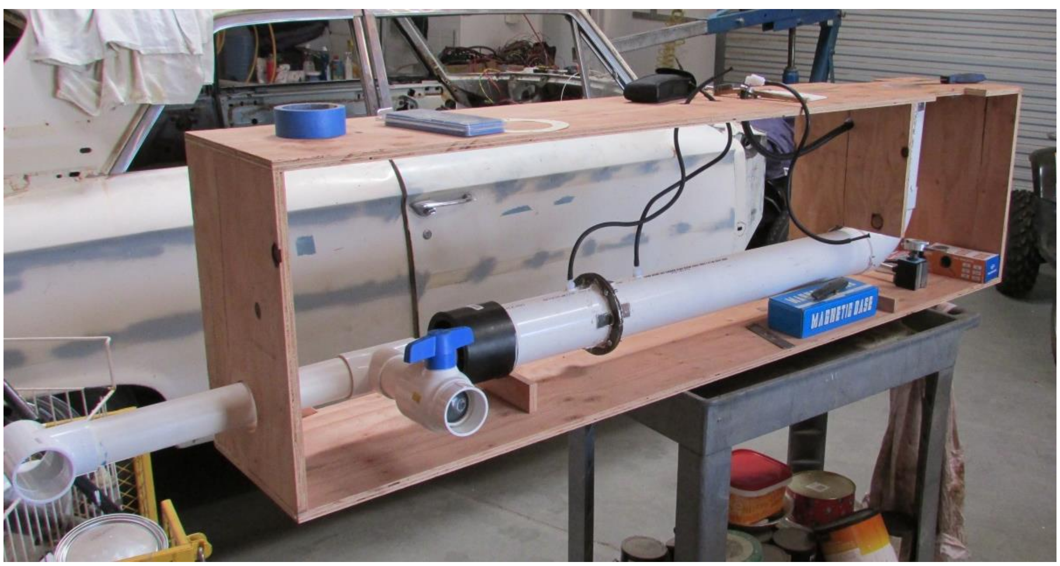

Figure 14, Completed Flow Bench

APPENDIX E: Original Data Sheet

$T = 71°F$

$\phi = 28\%$

$P_{atm} = 28.8$ in Hg

Test: 5-6 pm

Value lift (in)	ΔP (in H_2O) 1	ΔP (in H_2O) 2
.05	.2	.25
.1	.6	.6
.15	1.3	1.4
.2	2.4	2.4
.25	3.6	3.5
.3	4.8	4.7
.35	5.6	5.5
.4	6.2	6.2
.45	6.6	6.8
.5	6.3	6
.55	6.4	6.2
.6	6.2	6

2.05" orifice $P_{Test} = 15.5$ in H_2O

APPENDIX F: Pettis Performance Data Sheet

PETTIS PERFORMANCE

RACING ENGINES <> MACHINE SHOP <> DYNO

17585 LEMON ST. HESPERIA, CA. 92345 <> 760-244-5239 or 760-244-4415 <> fax: 760-244-1572

www.PettisPerformance.com <> PettisPerformance@Verizon.net

FLOW BENCH RESULTS

Project: NICK CORRAU SB CHEVY 461

Test pressure: 15 & 28 Bore size: 4.155

PORT: 3/6 INT Valve: 1.940 STOCK TYPE

Valve seat: AS RECIEVED

15"

scale:	71.4	150	150	298	298	298
meter:	62.5	59	83	49	50	50
LIFT:	.100	.200	.300	.400	.500	.600
CFM:	44.6	88.5	124.5	146.0	149	149
gain:						

Notes:

PORT: 3/6 INT Valve: 1.940 STOCK TYPE

Valve seat: AS RECIEVED

28"

scale:	71.4	150	298	298	298	298
meter:	87	80.5	56	65	68	68
LIFT:	.100	.200	.300	.400	.500	.600
CFM:	62.1	120.7	166.8	193.7	202.6	202.6
gain:						

Notes: