Proceedings of the World Congress on Engineering 2013 Vol III,
WCE 2013, July 3 - 5, 2013, London, U.K.

Air Flow Optimization via a Venturi Type Air Restrictor

Anshul Singhal, Mallika Parveen, *Member, IAENG*

***Abstract*— The aim of this project is to create a flow restriction device to be fitted in the FSAE (Formula Society of Automotive Engineers) car being built by Team Gear Shifters of BITS-Pilani, Dubai. The car is an open-wheeled race vehicle, designed to go from 0-100 kmph in under 4 seconds and have a top speed of about 140kmph. In order to comply with the rules of the competition imposed by FSAE, a single circular restrictor of 20mm diameter must be placed in the intake system between the throttle and the engine and all engine airflow must pass through the restrictor. This is done primarily to limit the power capability from the engine. Since the maximum mass flow rate is now a fixed parameter because of the restrictor, the aim is to allow the engine to achieve the maximum mass flow with minimal pull from the engine. In short, the pressure difference between atmosphere and the pressure created in the cylinder should be minimal, so that maximum airflow into the engine at all times. From the data gathered through the numerous simulations in SolidWorks Flow Simulation, it can be observed that the optimized values for converging angle and diverging angle of the Venturi were found to be 18 degrees and 6 degrees respectively.**

***Index Terms*— Air Flow, Design Optimization, FSAE Air Restrictor, SolidWorks Flow Simulation**

I. Introduction

THE aim of this project is to create a flow restriction device to be fitted in the FSAE (Formula Society of Automotive Engineers) car being built by Team Gear Shifters of BITS-Pilani, Dubai Campus. The car is an open-wheeled race vehicle, designed to go from 0-100 kmph in under 4 seconds and have a top speed of about 140kmph. In order to comply with the rules of the competition imposed by FSAE, a single circular restrictor of 20mm diameter must be placed in the intake system between the throttle and the engine and all engine airflow must pass through the restrictor. This is done primarily to limit the power capability from the engine [10].

An Internal Combustion Engine takes in air from the environment and the air-fuel mixture is combusted inside the engine cylinders to generate the power required to run the vehicle. In a naturally aspirated engine, the engine creates a low pressure during the intake stroke, causing the air from the atmosphere to enter the cylinders [2]. The higher the rpm, the greater the pull, and the lower the pressure created inside the cylinder. According to the stoichiometric air-fuel ratio, to burn 1 gram of gasoline 14.7 grams of air is required. By reducing the diameter of the flow path from 50mm to 20mm, the flow cross-section area has reduced drastically [11]. At low rpm's of the engine when the engine requires less air, the reduction in area is compensated by the accelerated flow of air through the throat (20mm section). But since the car is designed to run at high rpm's (6,000rpm to 10,000rpm with the restrictor attached), the flow at the throat reaches sonic velocities (also known as Critical Flow condition), and therein lays the problem. Critical Flow exists when the mass flow is the maximum possible for the existing upstream conditions, and the average velocity closely approximates the local sonic velocity (speed of sound in air ≈ 330m/s, or Mach 1) [3]. Since the maximum mass flow rate is now a fixed parameter because of the restrictor, the aim is to allow the engine to achieve the maximum mass flow with minimal pull from the engine. In short, the pressure difference between atmosphere and the pressure created in the cylinder should be minimal, so that maximum airflow into the engine at all times.

Manuscript received March 24, 2013; revised April 10, 2013.

Anshul Singhal is with the Birla Institute of Technology and Science Pilani, Dubai Campus; Dubai International Academic City, PO BOX – 345055 UAE; phone: +971 555 531826; e-mail: anshulsinghal33@gmail.com

Mallika Parveen is with the Birla Institute of Technology and Science Pilani, Dubai Campus; Dubai International Academic City, PO BOX – 345055 UAE; phone: +971 508 412493; e-mail: parveen@bits-dubai.ac.ae

II. Method of Research

A. Deciding an Appropriate Kind of Obstruction Meter

Air Restrictor that is being designed is basically a kind of an Obstruction Meter. Since the aim is to optimize the Mass Flow rate we will first study different kinds of Obstruction Meters available. Broadly there exist two obstruction meters used in industries – Orifice and Venturi meters.

a. Orifice.

An orifice plate is a thin plate with a hole in the middle. It is usually placed in a pipe in which fluid flows. When the fluid reaches the orifice plate, the fluid is forced to converge to go through the small hole; the point of maximum convergence actually occurs shortly downstream of the physical orifice, at the so-called vena contracta point. As it does so, the velocity and the pressure change. Beyond the vena contracta, the fluid expands and the velocity and pressure change once again. By measuring the difference in fluid pressure between the normal pipe section and at the vena contracta, the volumetric and mass flow rates can be obtained from Bernoulli's equation. It generally has a Coefficient of drag around 0.65 [7]

b. Venturi.

The Venturi tube or simply a Venturi is a tubular setup of varying pipe diameter through which the fluid flows. It follows the same laws as the orifice meter (for compressible fluids, the velocities have to be subsonic). The Venturi effect is a jet effect; as with a funnel the velocity of the fluid increases as the cross sectional area decreases, with the static pressure correspondingly decreasing [4]. According to the laws governing fluid dynamics, a fluid's velocity must increase as it passes through a constriction to satisfy the principle of continuity, while its pressure must decrease to satisfy the principle of conservation of mechanical energy. Thus a drop in pressure negates any gain in kinetic energy a fluid may accrue due to its increased velocity through a constriction. An equation for the drop in pressure due to the Venturi effect may be derived from a combination of Bernoulli's principle and the continuity equation. It generally has a coefficient of drag around 0.85 [8].

Inference:

After analysing the two kinds of obstruction meters available it can be concluded that the appropriate obstruction meter for designing the Air restrictor would be Venturi meter with Cd (~0.85), which is greater than the Cd (~0.65) of Orifice meter.

B. Deciding the Parameter to Be Optimized

The main objective of using the Venturi design in the air restrictor is to maintain a constant mass flow rate with optimum flow of air. The mass flow rate can be maintained constant by varying any one of the parameters namely -

a. Energy
b. Velocity
c. Mach number
d. Pressure.

For practical applications calculating the Energy, Mach number and Velocity is an intricate process. Mach number is calculated using the equation:

$$M=\frac{v}{a} \qquad (1)$$

Where,
M is the Mach number,
v is the velocity of the source relative to the medium and
a is the speed of sound in the medium.

When the medium is changed the velocity of sound in the medium will change accordingly and knowing the exact value for it is difficult. Due to this using velocity and Mach number as a parameter in a fixed Venturi apparatus is not feasible. For a permanent installation of air restrictor it is tedious and impracticable to deal with the above parameters for an optimum result.

Inference:

Pressure is the most ideal parameter that can be varied to hold the flow rate constant because pressure difference across the two ends of a Venturi can be measured easily using a simple U-shaped manometer [9].

III. THEORY AND FORMULAE

For this we use theoretical data and formulae of the Venturi meter in the following way. Using Bernoulli's equation in the special case of incompressible flows (e.g. the flow of water or other liquid, or low speed flow of gas ***v***), the theoretical pressure drop $\boldsymbol{p_1-p_2}$ at the constriction is given by [13]:

$$p_1-p_2=\frac{\rho}{2}(v_2^2-v_1^2) \qquad (2)$$

Where, ρ is the density of air.

Volumetric flow rate ***Q*** is given by: -

$$Q=v_1A_1=v_2A_2 \qquad (3)$$

Where ***A*** the area of cross-section of Venturi at any point and ***v*** is the velocity of air at that point.

$$p_1-p_2=\frac{\rho}{2}(v_2^2-v_1^2) \qquad (4)$$

Then

$$Q=A_1\sqrt{\frac{2}{\rho}\bullet\frac{(p_1-p_2)}{\left(\left(\frac{A_1}{A_2}\right)\right)^2-1}}=A_2\sqrt{\frac{2}{\rho}\bullet\frac{(p_1-p_2)}{1-\left(\left(\frac{A_2}{A_1}\right)\right)^2}} \qquad (5)$$

But as we can see, all the above calculations have been made based on one assumption – Incompressible Flow. Unfortunately, the fluid under observation is Air, and air is compressible, meaning that we have to take the compressible part of air into account. To derive the equation for Mass Flow Rate of an Ideal Compressible Fluid, we have to go back to the basics [10].

The conservation of mass is a fundamental concept of physics. Within some problem domain, the amount of mass remains constant; mass is neither created nor destroyed. The mass of any object is simply the volume that the object occupies times the density of the object. For a fluid (a liquid or a gas) the density, volume, and shape of the object can all change within the domain with time and mass can move through the domain [1].

The conservation of mass (continuity) tells us that the mass flow rate through a tube is a constant and equal to the product of the density ρ, velocity V, and flow area A:

$$\mathbf{m = \rho * V * A} \qquad (6)$$

Considering the mass flow rate ***m*** equation, it appears that for a given area ***A*** and a fixed density ρ, we could increase the mass flow rate indefinitely by simply increasing the velocity ***v***. In real fluids, however, the density does not remain fixed as the velocity increases because of compressibility effects. We have to account for the change

ISBN: 978-988-19252-9-9
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

in density to determine the mass flow rate at higher velocities. If we start with the mass flow rate equation given above and use the isentropic flow relations and the equation of state, we can derive a compressible form of the mass flow rate equation. We begin with the definition of the Mach number ***M*** and the speed of sound **a** [12]-[5]:

$$V = M * a = M * sqrt\ (\gamma * R * T) \tag{7}$$

Where **γ** is the specific heat ratio, ***R*** is the gas constant, and ***T*** is the temperature.

Now substitute **(7)** into **(6):**

$$m = \rho * A * M * sqrt\ (\gamma * R * T) \tag{8}$$

The equation of state is:

$$\rho = P / (R * T) \tag{9}$$

Where ***P*** is the pressure. Substitute **(9)** into **(8):**

$$m = A * M * sqrt\ (\rho * R * T) * P / (R * T) \tag{10}$$

$$m = A * sqrt\ (\gamma / R) * M * P / sqrt(T) \tag{11}$$

From the isentropic flow equations:

$$P = Pt * (T / Tt)\string^(\gamma/(\gamma-1)) \tag{12}$$

Where ***Pt*** is the total pressure and ***Tt*** is the total temperature. Substitute **(12)** into **(11):**

$$m = (A * Pt) / sqrt(Tt) * sqrt\ (\gamma / R) * M * (T / Tt)\string^((\gamma + 1) / (2 * (\gamma - 1))) \tag{13}$$

Another isentropic relation gives:

$$T / Tt = (1 + .5 * (\gamma-1) * M\string^2)\ \string^{-1} \tag{14}$$

Substitute **(14)** into **(13):**

$$m = (A * Pt/sqrt[Tt]) * sqrt(\gamma / R) * M * [1 + .5 * (\gamma - 1) * M\string^2]\string^-[(\gamma + 1)/(\gamma - 1)/2] \tag{15}$$

This equation is shown in the red box below (NASA). It relates the mass flow rate to the flow area A, total pressure Pt and temperature Tt of the flow, the Mach number M, the ratio of specific heats of the gas γ, and the gas constant R.

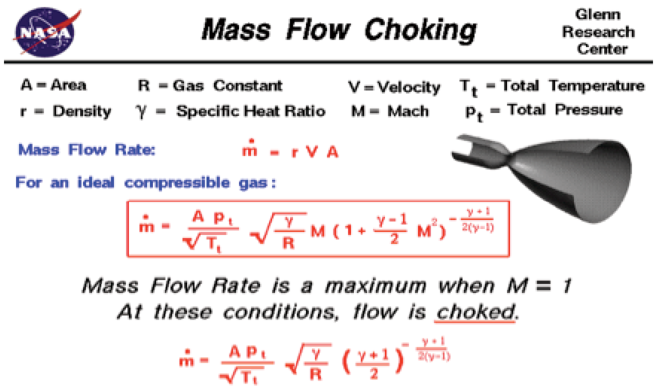

Fig 1. Formulae for Mass Flow Choking [9]

Calculations:

The values taken for substitution in Equation #10 are:
Pt = 101325 Pa
T = 300K
γ = 1.4
R (air) =0.286 kJ/Kg-K
A = 0.001256 m^2
M = 1 (Choking Conditions)

Result: Mass Flow Rate at Choking = **0.0703 kg/s**

IV. DATA ANAYLSIS

A. *Aim*

To maximise pressure recovery at outlet

B. *Variables*

a. Dependent Variables:
Delta Pressure *(= Inlet Pressure – Outlet Pressure)*

b. Independent Variables:
Converging angle and Diverging angle

c. Constants:
Inlet, Outlet and Throat diameter, Type of Fluid -Air, Temperature - 300K

C. *Boundary Conditions*

a. Inlet Face: Total Pressure = 1 bar
b. Outlet Face: Mass Flow Rate = 0.0703kg/s

D. *Software Used*

1. Modelling - CATIA V5
2. Analysis - SolidWorks 2012-2013
3. Data Tabulation - MS Excel 2007
4. Data Interpretation - MATLAB
5. Data Compilation - MS Word 2007
6. Paper Formatting - Adobe InDesign

ISBN: 978-988-19252-9-9
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

E. Data Collection and Analysis

Table I. Delta Pressure for Different Converging and Diverging Angles

	Min Value in Row	Diverging Angles						
xx,xxx	Min Value in Column	4	6	8	10	12	15	20
Converging Angles	**10**	20000	**19398**	19504	21486	22032	20044	23033
	12	19948	19700	**19326**	19679	19960	19882	23846
	14	22249	19621	20619	20576	21336	**19585**	21844
	15	**19274**	**18443**	21242	**19601**	20919	19686	22292
	16	21203	20240	20654	20835	**19242**	20095	24341
	18	19556	**18204**	20671	21604	22123	22906	**20630**
	20	20318	**18523**	20263	24719	22180	21233	24019

For the calculated Maximum Mass Flow Rate of 0.0703 kg/s, the Delta Pressure (Inlet Pressure – Outlet Pressure) was calculated for different converging and diverging angles. The minimum angle that can be manufactured with considerable precision was estimated at 4 degrees, hence the diverging angle has been considered from 4 degrees. The Diverging angle range was considered until 20 degrees because the value of Delta Pressure was increasing abruptly after 15 degrees, clearly visible by the lack of any minimum value in the 20 degrees columns. The Converging Angles have been considered from 10 degrees referring to the estimates of Venturi designs previously made [6].

In the Data collected the Minimum Delta Pressure Value was marked for every row and column as shown in the table above to find out the best combination of Diverging and Converging angle. From the above table, we can see that most of the minimum values are achieved when the Diverging Angle is 6 degrees. Therefore we shall concentrate on 6 degrees as the Diverging Angle and extend our testing to Converging Angles of 22 degrees and 25 degrees, also because a persistent trend was lacking. The data for Diverging Angle of 6 degrees is shown below [2].

Table II. Table of Delta Pressure for Diverging Angle of 6 Degrees

xx,xxx	Min Vaue	Delta Pressure
Converging Angles	10	19398
	12	19700
	14	19621
	15	18443
	16	20240
	18	**17811**
	20	17887
	22	20308
	25	20878

Inference:

From the table and graph above, it can be seen that Delta Pressure keeps increasing after 18 degrees as the Converging Angle increases. It is inferred that the *Minimum value for Delta Pressure* is achieved when the *Converging Angle is 18 degrees and the Diverging Angle is 6 degrees.*

F. Images of Data Analysis in Solid Works 2012-13

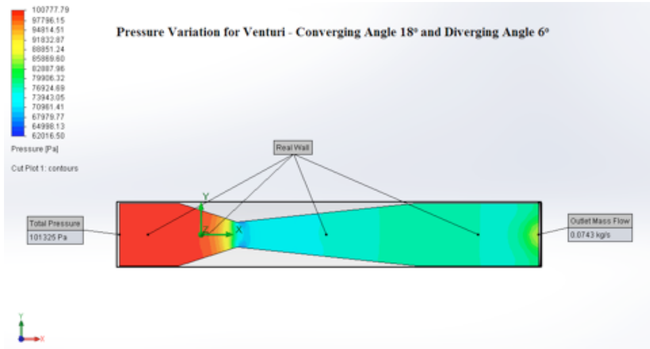

Fig 2. Pressure variation – Converging angle 18 degrees and Diverging angle 6 degrees

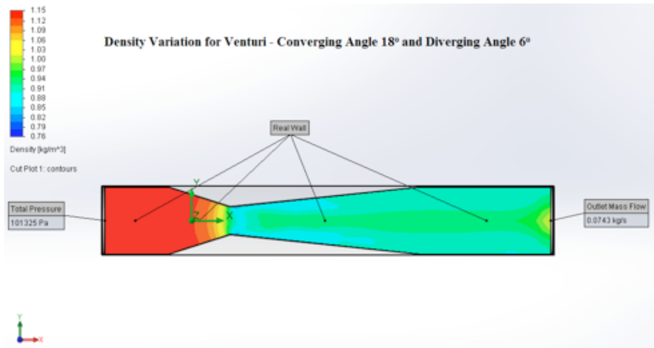

Fig 3. Density Variation – Converging Angle 18 Degrees and Diverging Angle 6 Degrees

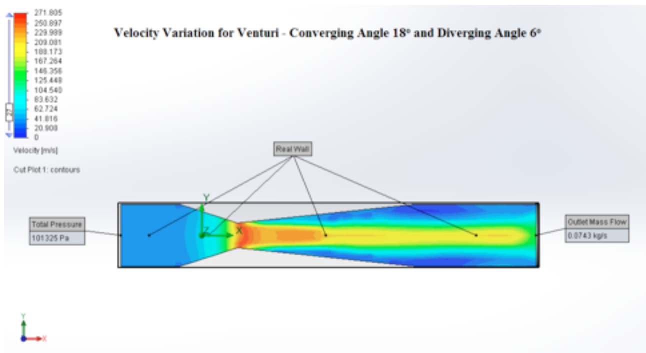

Fig 4. Velocity Variation – Converging Angle 18 Degrees And Diverging Angle 6 Degrees

V. CONCLUSION

The optimum solution to achieve maximum possible mass flow rate of air as quickly as possible is to minimize the pressure loss through the flow restriction device. The best general design for this objective is to use the Venturi design. From the data gathered through the numerous simulations, it can be observed that the values for converging angle and diverging angle of the Venturi are 18 degrees and 6 degrees respectively.

FSAE is about speed, acceleration and economy. Therefore a majority of the parts that need to be fabricated for the FSAE car are made using sheet metal. Also procuring a billet of diameter greater than 50 mm and the subsequent machining of the part to our specifications using a lathe was significantly more difficult and expensive, and so it was finalised that sheet metal will be used. Galvanised iron sheets were beaten into their respective profiles, welded, primer and spray-painted. The total cost borne was AED 15 for the primer coat and the spray paint; the sheet metal and welding services were borrowed from the Mechanical Workshop of the College.

The manufactured restrictor serves the purpose of complying with the rules with minimal compromise on power. As soon as the effective vacuum pressure created by the suction strokes of the engine cylinders reaches ≈17800Pa, maximum mass flow rate is achieved and subsequently maximum power of the engine as well (for the same rpm using any other restrictor specification).

ACKNOWLEDGMENT

I would like to thank all the people who contributed their time and helped me in writing this paper and completing in timely manner. I would also thank them for making this paper a successful one.

I would like to thank Dr. R.K. Mittal, BPD Director, for giving me the opportunity to take up this paper. I would also like to extend my gratitude to my friends and senior students of this department who have always encouraged and supported in doing my work. Last but not the least I would like to thank all the staff members of Department of Mechanical Engineering who have been very cooperative with me. I would like to make a special note of thanks to Dr. Mallika Parveen for her constant guidance and supervision in all respects of this paper.

REFERENCES

[1] Abdulnaser Syma, “Computational Fluid Dynamics,” 1st edition. Available:http://bookboon.com/en/textbooks/energyenvironment/computational-fluid-dynamics
[2] Anish Ganesh, “Students Win Using Simulation-Driven Design,” University of Waterloo Formula Motorsports, Canada. Available at: http://www.idac.co.uk/enews/articles/Students-Win-Using-Sim-Driven-Design.pdf
[3] Antonio Gomes, “Documentation of the Engine Design Process for FSAE,” Department of Mechanical and Industrial Engineering University of Toronto, March 2007. Available at: http://www.studymode.com/essays/The-Anlyse-Of-Intake-1258959.html
[4] ASME MFC-7M, Measurement of Gas Flow by Means of Critical Flow Venturi Nozzles. New York: American Society of Mechanical Engineers; 1987.
[5] Bean, H. S., ed. Fluid Meters: Their Theory and Application, 5th edition. New York: The American Society of Mechanical Engineers; 1959.
[6] Deepak Ranjan Bhola, “CFS Analysis Of Flow Through Venturi Of A Carburetor,” National Institute of Technology Rourkela, Rourkela – 769008, India, May 2011. Available at: http://ethesis.nitrkl.ac.in/2296/1/final_report.pdf
[7] Orifice Plate description – Flow through an Orifice, Available at: en.wikipedia.org/wiki/Orifice_plate, Consulted on – 2 Feb 2013
[8] Venturi description – Flow through a Venturi Available at: en.wikipedia.org/wiki/Venturi_effect, Consulted on – 20 Jan 2013
[9] www.grc.nasa.gov/WWW/k-12/airplane/mflchk.html, Consulted on – 15 Feb 2013
[10] Neal Persaud, “Design and Optimization Of a Formulae SAE Cooling System,” Department of Mechanical and Industrial Engineering, University Of Toronto, March 2007. Available at: http://docsse.com/view.php?id=856504
[11] FSAE Variable Air Intake Consulted on – 22 JAN 2013 http://poisson.me.dal.ca/~dp_06_9/concept.htm
[12] Section 8, “Sonic Flow Nozzles and Venturi — Critical Flow, Choked Flow Condition”, ASME Publication
[13] Jack Philip Holman, Experimental Methods for Engineers 6th edition, September 1993, McGraw-Hill Higher Education