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# Analytical Investigation to Predict the Intake pipe Diameter in Naturally Aspirated Internal Combustion Engine

Mahmoud A. Mashkour
Department of Mechanical Engineering, University of Technology, Baghdad, Iraq

**Abstract:** An analytical solution to estimate the effective diameter of a naturally aspirated internal combustion engine intake pipe is presented in this paper. A formula to predict the effective diameter for the geometry of the intake pipe as a function of cylinder bore, stroke, engine speed, pipe length, ambient temperature and aerodynamic losses, is created. In this analysis, the kinetic energy of a charge column in the period of a filling cylinder and the maximum kinetic energy are taken into consideration. A comparison between the results of the novel suggested formula and the actual published work to determine the intake pipe diameter is done for different engines. The results indicated that a good agreement to estimate the intake pipe diameter, especially in high-speed engines is obtained.

**Key words:** Internal combustion engine, induction system, intake pipe, volumetric efficiency

## INTRODUCTION

The basic function of the engine intake pipe is to provide the engine with a fresh charge for combustion in cylinders to take place every cycle. It is therefore important to do a proper intake pipe design. The design of the intake pipe represents a key area in which the automobile engineer can significantly influence engines performance. For both spark and compression ignition engines, the fluid flow processes with intake pipe play an essential role in determining an engine's overall operating characteristics. The power output of an internal combustion engine is directly proportional to the amount of the charge (air or mixture) that can be forced into the cylinder per cycle. This amount is most effectively increased by means of a mechanical supercharger or a turbocharger. These devices have several disadvantages. First, they add to the mechanical friction or pumping work of the engine and second they raise the inlet charge temperature of the engine, increasing the sensitivity of the engine to knock. However, the power output may also be increased by the selection of a suitable diameter and length for the intake pipe. Many researchers have found that the amount of charge forced into the cylinder can be increased almost 15-20% by this method (Heywood, 2002).

The intake pipe is considered as a device for storing kinetic energy. It is argued that as the charge flows toward the cylinder, the kinetic energy is stored in the charge column by virtue of its velocity. Some of this energy can be recovered toward the end of the inlet process because of the ramming effect of the charge column which tends to build up the pressure at the intake port and forces more charge into the cylinder. The length of the intake pipe should provide a frequency (oscillation) of flow process of charge column movement conformity with the engine, while, the diameter of the intake pipe should provide a sufficient store in kinetic energy for this charge column at a minimum losses of energy that required to overcome various types of aerodynamic resistances (the air filter, major and minor losses, losses in valves mechanisms) (Heywood, 2002; Taylor *et al*., 1955). From this point of view, it is important therefore to design an intake pipe geometry (its length and diameter) that will reduce the impact of this flow restriction.

Research has focused (either experimentally or numerically) on the effect of flow behavior in the intake pipes which can be used to force additional air (or mixture) into the engine making it more efficient. The level of the influence is mainly determined by the pipe diameter and length. Until recently, the design of such problem has been based on a trial and error which is costly at time consuming. Early intake pipes were based on any commercially available pipes without regarding the behavior to interior flow as shown in Fig. 1 (Eberhard, 1971). Many theories were built forward regarding the behavior of the flow in the intake pipe. The numerical techniques and methods (such as method of characteristics, Lax-Wendroff method, finite element method, etc.,) were used to determine the optimum intake pipe length and diameter to achieve improved volumetric efficiency. A number of papers have been published on this subject, from 1-D, 2-D to 3-D like (Ferguson and Kirkpatrick, 2001). In addition, there are a number of commercial codes available today (like KIVA, LOTUS)

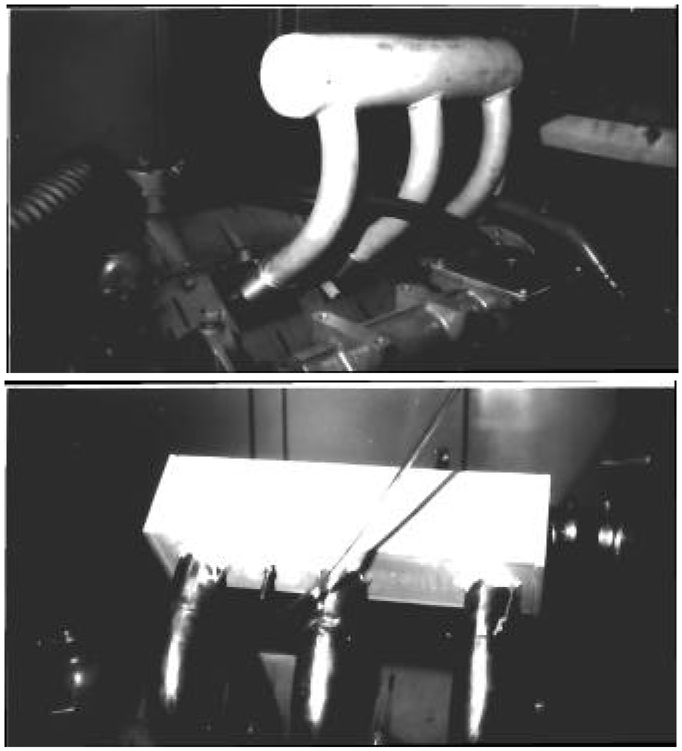

Fig. 1: Early intake pipe configurations (Taylor *et al*., 1955)

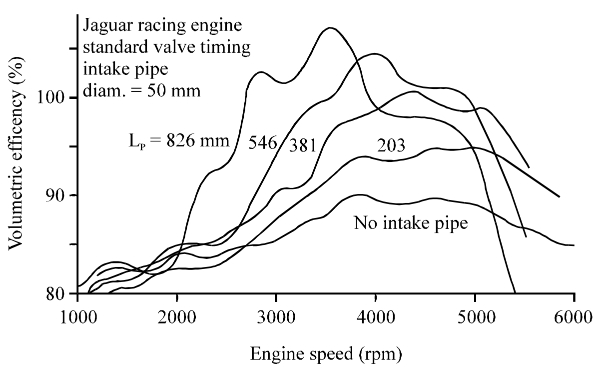


Fig. 2: Measured volumetric efficiency versus engine speed and intake pipe length (Tabaczynski, 1982)

(Pearson *et al*., 2006). Tabaczynski (1982) showed experimentally the effect of intake pipe length on the volumetric efficiency with different engine speeds as shown in Fig. 2. Analytical model was suggested to predict the effective length of the intake pipe (Ferguson and Kirkpatrick, 2001; Tabaczynski, 1982).

The main object of this study is concerned with the mathematical description of such intake pipe diameter. An analytical solution to estimate the effective diameter of a naturally aspirated internal combustion engine intake pipe is presented. A formula is created to predict the effective diameter for the geometry of the intake pipe as a function of cylinder bore, stroke, engine speed, pipe length and aerodynamic losses. A comparison between the results of the novel suggested formula and the actual published work to determine the intake pipe diameter is done for different engines.

## MATHEMATICAL ANALYSIS

The work can be divided into the following main steps. Firstly, is to find an algebraic expression for the kinetic energy of a charge column in the intake pipe during the period of filling the cylinder. This analysis is taken into considering the pressure loss that required to overcome the major and minor resistances. Secondly, is to obtain the maximum kinetic energy by differentiating the equation obtained in the first step relative to the intake pipe diameter then equating to zero. The resulting solution of this equation leads to predicate the best formula to design the diameter of the intake pipe. Finally, the diameter estimation of this form is validated with the actual model.

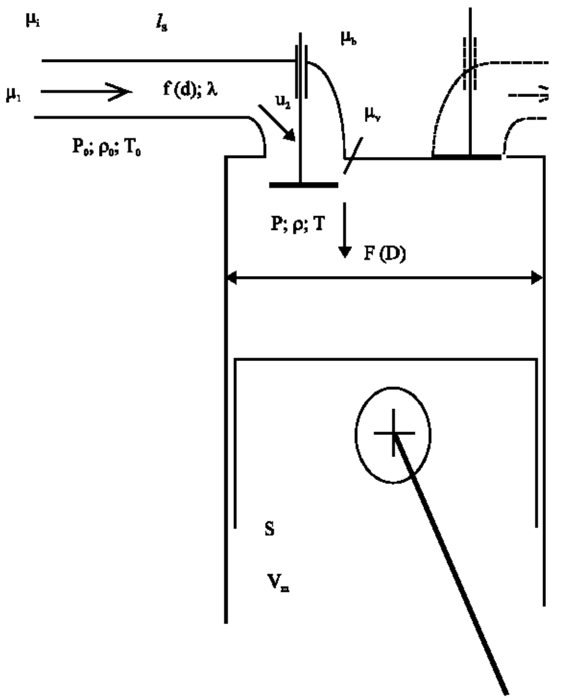


Fig. 3: Representation of intake pipe for four stroke aspirated internal combustion engine (Tabaczynski, 1982)

The approach for providing the maximum charge flow to engines as shown in Fig. 3, has to build intake pipes with large diameter which yield small charge losses due to friction owing to small charge velocities in the pipes. The opposite situation that of smaller diameter and higher velocities may also aid the charge flow to the engine, due to use of the kinetic energy of the charge for the purpose of "ram-charging" the cylinder, to compress the cylinder charge prior to intake valve closure. The "tuned" pipe makes use this phenomenon to increase the pressure in the cylinder at valve closure, thereby increasing the mass of charge taken in by the cylinder (Heywood, 2002; Ferguson and Kirkpatrick, 2001).

The intake pipe $l_c$ in which all local resistances are included by increasing the origin intake pipe length $l_S$ by an additional virtual length pipe $l_{eq}$.

Hence:

$$l_C = l_S + l_{eq} \tag{1}$$

The total pressure loss of the pipe is the same as that produced in a straight pipe whose length is equal to the pipe of the original system plus the sum of the equivalent length of all the components in the system.

Using the famous formula for Darcy-Weisbach, the pressure loss considering the major and minor losses can be calculated:

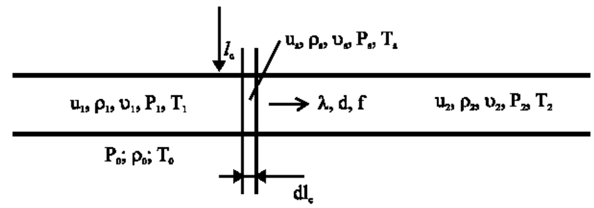


Fig. 4: Representation model solution of the corrected intake pipe

$$h = \lambda \frac{l_{eq}}{d_S}\frac{u^2}{2} = (\mu_i + \mu_b + \mu_v)\frac{u^2}{2} \tag{2}$$

From that, the equivalent length is:

$$l_{eq} = \frac{d_S}{\lambda}(\mu_i + \mu_b + \mu_v) \tag{3}$$

Thus, the new intake pipe length is the greater length $l_c$ possesses aerodynamic properties of the real intake pipe length $l_p$, however, this kind is more convenient for the mathematical solution of this problem.

According to Fig. 4, the kinetic energy per unit volume in the outlet of pipe:

$$K.E. = \frac{\rho_2 u_2^2}{2} \tag{4}$$

Moreover, from continuity equation can be obtained:

$$u_2 = \xi \frac{F V_m}{f} = \xi \left(\frac{D}{d_S}\right)^2 V_m \tag{5}$$

where, $\xi$ is a coefficient of corrected velocity factor.

Then, substituting Eq. 5 in Eq. 4 results:

$$K.E. = \xi^2 \frac{\rho_2}{2}\left(\frac{D^2}{d_S^2}\right)^2 V_m^2 \tag{6}$$

The law of movement for one dimensional in the differential form (Benson, 1982) is:

$$\frac{dP_S}{\rho_S} + d\left(\frac{u_S^2}{2}\right) = -\lambda \frac{dl_c}{d_S}\frac{u_S^2}{2} \tag{7}$$

It seen that:

$$d\left(\frac{P_S}{\rho_S}\right) = \frac{\rho_S dP_S - P_S d\rho_S}{\rho_S^2}$$

Alternatively, in other form:

$$\frac{dP_S}{\rho_S} = d\left(\frac{P_S}{\rho_S}\right) + P_S \frac{d\rho_S}{\rho_S^2} \tag{8}$$

The work done per unit mass of a gas in the pipe can be calculated as in the following:

$$\begin{aligned} W &= \int_{P_1}^{P_2} v_S \, dP_S \\ &= P_2^{\frac{1}{k}} v_2 \int_{P_1}^{P_2} P_S^{-\frac{1}{k}} dP_S \\ &= P_2^{\frac{1}{k}} v_2 \frac{k}{k-1}\left(P_2^{\frac{k-1}{k}} - P_1^{\frac{k-1}{k}}\right) \\ &= \frac{k}{k-1}\left(P_2 v_2 - P_2^{\frac{1}{k}} v_2 P_1^{\frac{k-1}{k}}\right) \\ &= \frac{k}{k-1}(P_2 v_2 - P_1 v_1) \end{aligned}$$

The total work done by the gas with considering its velocity (Estop and McConkey, 2009):

$$W = \frac{k}{k-1}(P_2 v_2 - P_1 v_1) + \frac{u_2^2 - u_1^2}{2}$$

In a differential form:

$$dW = \frac{k}{k-1} d(P_S v_S) + d\left(\frac{u_S^2}{2}\right) \tag{9}$$

Here, dW = 0, that is:

$$\frac{k}{k-1} d(P_S v_S) + d\left(\frac{u_S^2}{2}\right) = 0$$

And, as: $v_s = \frac{1}{\rho_s}$

$$d\left(\frac{P_S}{\rho_S}\right) = -\frac{k-1}{k} d\left(\frac{u_S^2}{2}\right) \tag{10}$$

Now from the law of conservation of mass ($M_S = \rho_S u_S f$):

$$\begin{aligned} d\rho_S &= -\frac{M_S}{f}\frac{du_S}{u_S^2} \\ &= -\rho_S u_S \frac{du_S}{u_S^2} = -\rho_S \frac{du_S}{u_S} \end{aligned}$$

And, hence:

$$\frac{d\rho_S}{\rho_S} = -\frac{du_S}{u_S}$$

Thus:

$$d\left(\frac{u_S^2}{2}\right) = u_S \, du_S = -u_S^2 \frac{d\rho_S}{\rho_S}$$

In addition, that at steady flow:

$$\rho_2 u_2 = \rho_S u_S = \rho_1 u_1$$

From that, can be obtained:

$$u_S = \frac{\rho_2 u_2}{\rho_S} \tag{11}$$

$$d\left(\frac{u_S^2}{2}\right) = -(\rho_2 u_2)^2 \frac{d\rho_S}{\rho_S^3} \tag{12}$$

Substituting the Eq. 8, 10, 11 and 12 in the Eq. 7 yields:

$$\frac{k-1}{k}(\rho_2 u_2)^2 \frac{d\rho_S}{\rho_S^3} + P_S \frac{d\rho_S}{\rho_S^2} - (\rho_2 u_2)^2 \frac{d\rho_S}{\rho_S^3} = -\lambda \frac{dl_c}{d_S} \frac{(\rho_2 u_2)^2}{2\rho_S^2}$$

The replacement of $P_S$ by an equivalent value in the equation of state of gas and making transformations and simplifying, gives:

$$-\frac{2}{k}\frac{d\rho_S}{\rho_S} + \frac{2RT_S}{(\rho_2 u_2)^2}\rho_S \, d\rho_S = -\lambda \frac{dl_c}{d_S} \tag{13}$$

Now, equating equation of state for gases with Eq. 10:

$$d\left(\frac{P_S}{\rho_S}\right) = RT_S = -\frac{k-1}{k} d\left(\frac{u_S^2}{2}\right)$$

After integration, results:

$$R(T_S - T_2) = -\frac{k-1}{k}\frac{u_S^2 - u_2^2}{2}$$

Substituting by equivalent value in Eq. 11:

$$RT_S = \frac{k-1}{k}\left(\frac{u_2^2}{2} - \frac{1}{2}\left(\frac{\rho_2 u_2}{\rho_S}\right)^2\right) + RT_2$$

Substituting value $RT_S$ in Eq. 13, after transformations, becomes:

$$-\frac{k+1}{k}\frac{d\rho_S}{\rho_S}+\left(\frac{k-1}{k\rho_2^2}+\frac{2RT_2}{(\rho_2 u_2)^2}\right)\rho_S d\rho_S=-\lambda\frac{dl_c}{d_S} \tag{14}$$

Integrating this equation along the pipe length results in:

$$\begin{aligned}&-\frac{k+1}{k}\int_{\rho_1}^{\rho_2}\frac{d\rho_S}{\rho_S}+\left(\frac{k-1}{k\rho_2^2}+\frac{2RT_2}{(\rho_2 u_2)^2}\right)\int_{\rho_1}^{\rho_2}\rho_S\, d\rho_S=-\frac{\lambda}{d_S}\int_0^{l_c}dl_c\\ &-\frac{k+1}{k}\ln\frac{\rho_2}{\rho_1}+\left(\frac{k-1}{k\rho_2^2}+\frac{2RT_2}{(\rho_2 u_2)^2}\right)\frac{\rho_2^2-\rho_1^2}{2}=-\frac{\lambda l_c}{d_S}\end{aligned} \tag{15}$$

In Eq. 15 the first term can be neglected because it is small. Also, assume that the accepted limit velocity of flow is about 40 m $sec^{-1}$, so, the first part in the second term is small comparing with the second part so that can be neglected. Hence, from the last equation, we can get an expression for the density as follow:

$$\rho_2=\rho_1\sqrt{1-\frac{\lambda l_c}{RT_2}\frac{u_2^2}{d_S}} \tag{16}$$

So, the expression for the kinetic energy (Eq. 6) becomes:

$$K.E.=\frac{1}{2}\xi^2\rho_1\left(D^2V_m\right)^2\frac{1}{d_S^4}\sqrt{1-\frac{\lambda l_c}{RT_2}\xi^2\frac{\left(D^2V_m\right)^2}{d_S^5}} \tag{17}$$

The maximum kinetic energy can be obtained by differentiating the above equation relative to the pipe diameter and equating the resulted equation to zero.

$$\frac{dK.E.}{dd_S}=\xi^2\rho_1\frac{\left(D^2V_m\right)^2}{2}\left(\frac{1}{d_S^4}\frac{\frac{5}{2}\xi^2\frac{\lambda l_c}{RT_2}\frac{\left(D^2V_m\right)^2}{d_S^6}}{\sqrt{1-\xi^2\frac{\lambda l_c}{RT_2}\frac{\left(D^2V_m\right)^2}{d_S^5}}}-\frac{4}{d_S^5}\sqrt{1-\xi^2\frac{\lambda l_c}{RT_2}\frac{\left(D^2V_m\right)^2}{d_S^5}}\right)=0 \tag{18}$$

After equating to zero and some transformations, Eq. 18 becomes:

$$\frac{13\lambda l_c}{RT_2}\xi^2\frac{\left(D^2V_m\right)^2}{d_S^5}=8 \tag{19}$$

The average speed of the piston is:

$$V_m=\frac{2SN}{60}=\frac{SN}{30}$$

(N-revolution per minute of the engine), the following equation is obtained:

$$d_S=\sqrt[5]{\frac{13\lambda l_c}{8RT_2}\xi^2\frac{D^4(SN)^2}{900}} \tag{20}$$

The variation in the temperature in the intake pipe is closely equal to the ambient temperature. Also, substituting in last formula the following:

$$\xi_0=\sqrt[5]{\xi^2}$$

and R = 287 J/kg.K results:

$$d_S=0.09115\,\xi_0\sqrt[5]{\frac{1}{T_0}\lambda l_c D^4(SN)^2} \tag{21}$$

From the experimental data, $\xi_0$ = 1.48 in aspirated internal combustion engines can be used (Ferguson and Kirkpatrick, 2001):

$$d_S\approx 0.135\sqrt[5]{\frac{1}{T_0}\lambda l_c D^4(SN)^2} \tag{22}$$

In addition, for high-speed engines especially, it is possible to set and accept the equivalent length simply equal to 12-14 m and $\lambda$ = 0.018 practically. Thus with the aim of this correction factors, the following expression can be obtained:

$$d_S\approx 0.06045\sqrt[5]{\frac{1}{T_0}l_c D^4(SN)^2} \tag{23}$$

Therefore, the calculations of $d_S$ can be carried out corresponding to a rough estimate of the effective length $l_S$ under the formula given by Tabaczynski (1982):

$$l_S=7.5\frac{a}{N}$$

where, a is a speed of sound: $a=\sqrt{kRT}$.

Equation 23 is the required relationship to estimate the diameter of the inlet pipe as a function of cylinder bore, stroke, engine speed and pipe length when considering the flow losses.

## VERIFICATION OF RESULT

Verification is best achieved by means of comparison with the actual values for the intake diameter of an aspirated internal combustion engines. The following data was obtained from the published work. Many applications to examine the resulting equation were done and some of these as in the following cases:

**Case 1:** The first application of obtained equation on the Dorman engine (Data from Benson and Baruah work) (Benson, 1982), with the following specifications: D = 125 mm, S = 130 mm, N = 1500 rpm, $l_p$ = 609 mm

$$d_S \approx 0.06045\sqrt[5]{\frac{1}{T_0}l_c D^4 (SN)^2}$$
$$\approx 0.06045\times\sqrt[5]{\frac{1}{304}\times 12.609\times(0.125)^4\times(0.130\times 1500)^2}$$
$$\approx 0.049945\,m \approx 50\,mm$$

In Benson and Baruah work, the diameter is 50.8 mm.

**Case 2:** A Volkswagen engine with the following specifications: D = 82 mm, S = 110 mm, N = 1500 rpm

$$d_S \approx 0.06045\sqrt[5]{\frac{1}{T_0}l_c D^4 (SN)^2}$$
$$\approx 0.06045\times\sqrt[5]{\frac{1}{304}\times 15.7\times(0.082)^4\times(0.11\times 1500)^2}$$
$$\approx 0.03483\,m \approx 35\,mm$$

On other hand, the actual diameter of intake pipe is about $d_S \approx$ 33-36 mm.

**Case 3:** Another application of the obtained equation on the engine of automobile BMW, with the following specifications: D = 80 mm, S = 6.6 mm, N = 6000 rpm

$$l_S = 7.5\frac{340}{6000} = 0.425\,m;\ l_{eq} = 12\,m;\ l_c \approx 12.43\,m$$

$$d_S \approx 0.06045\sqrt[5]{\frac{1}{T_0}l_c D^4 (SN)^2}$$
$$\approx 0.06045\times\sqrt[5]{\frac{1}{304}\times 12.43\times(0.08)^4\times(0.066\times 6000)^2}$$
$$\approx 0.0463\,m \approx 46.3\,mm$$

Actually: $d_S$ = 46 mm

Clearly, it is noted that the difference between the value of the intake pipe diameter which is calculated from the above formula and the actual value, is small.

## CONCLUSION

The theoretical scheme described in this paper could successfully predict the intake pipe diameter of a naturally aspirated internal combustion engines.

Algebraic expression for the diameter of the intake pipe in the naturally aspirated internal combustion engines has been obtained as a function of cylinder bore, stroke, engine speed, ambient temperature, pipe length and losses.

The design equation shows a good correspondence with the values obtained from the actual engines. Clearly, depending on the obtained calculation, the technique offers a choice of an effective diameter of the intake pipe to provide essentially a desirable result, especially in aspirated high-speed piston engines by inertia supercharging as compared with previous results (Benson, 1982; Eberhard, 1971) and that can be recommended for using in the engineering practical field.

Obviously, the equation capable of accurately predicting the intake pipe diameter would be a powerful tool for the engine designer.

## NOMENCLATURE

a : Speed of sound
$V_m$ : Mean piston speed
D : Diameter of the cylinder
d : Pipe diameter
F : Cross-section area of the cylinder
f : Cross-section area of the pipe
k : Ratio of specific heats
l : Pipe length
N : Engine speed
P : Pressure
u : Velocity
R : Gas constant
S : Stroke
T : Temperature

**Greek letters:**

$\xi$ : Correction factor of velocity
$\lambda$ : Coefficient of friction losses (major losses)
$\mu$ : Coefficients of local aerodynamic losses (minor losses)
$\rho$ : Density
$\upsilon$ : Specific volume

**Subscripts:**

0 : Ambient conditions
1 : Upstream conditions
2 : Downstream conditions
i : Inlet
b : Bend
c : Corrected
eq : Equivalent
p : Pipe
v : Valve
s : Current conditions

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