A PROJECT REPORT ON

EFFECT OF PENTROOF INCLINATION ANGLE ON INDUCED TUMBLE IN 4-VALVE S.I. ENGINE

SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENT

OF

NATIONAL INSTITUTE OF TECHNOLOGY, RAIPUR (C.G.)

FOR

THE DEGREE OF BACHELOR OF TECHNOLOGY

(2014-18)

UNDER THE GUIDANCE OF

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ABSTRACT

The present study has been conducted to observe the effect of the pentroof inclination angle of a 4-valve S.I. engine on the induced tumble motion and the turbulence generated inside the combustion chamber. The objective behind the study is to focus on enhancing the in-cylinder charge motion for achieving better combustion. It has been found in [2] that higher turbulence intensity at the end of compression stroke resulted in a faster flame speed during ignition which further resulted in the combustion having lower percentages of unburnt hydrocarbons and lesser soot particles. One way of achieving higher turbulence levels at the end of compression stroke is by increasing the swirl or tumble induced in the charge flowing inside the combustion chamber as shown in [3]. As the induced tumble is found sensitive to the combustion chamber geometry[4], its variation with changes in the pentroof inclination angle have been studied. Different combustion chamber geometries have been developed and attention has been focussed on the in-cylinder flow behaviour during intake and compression strokes. A CFD analysis has been carried out using the k-ε turbulence model to generate the flow field.

The velocity field obtained has been used to compute the tumble ratio for the different chamber geometries, which have then been compared and discussed. This is followed by observing the variation of turbulence intensity during the intake and compression strokes to understand the different stages of turbulence. Finally, the values of the turbulence intensities near the spark plug at the end of compression stroke for the different geometries have been compared and discussed.

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Declaration

This Project work is a presentation of our original research work. Whatever contributions of others are involved, every effort is made to indicate this clearly, with due reference to literature and acknowledgment of collaborative research and discussion.

The work was done under the guidance of Dr. S Bhowmick (Assistant Professor) and Dr. V.K. Gaba (Assistant Professor) of National Institute of Technology, Raipur.

I, Devraj Singh Gaidu (14119017) and Vikas Verma(14119089), certify and ensure that the above statements are true to the best of our knowledge.

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Declaration

In my capacity as supervisor of candidates of the project work, I certify that above statements are true to the best of my knowledge.

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Certificate of Examiner

This is to certify that the project work titled EFFECT OF PENTROOF INCLINATION ANGLE ON INDUCED TUMBLE IN 4-VALVE S.I. ENGINE is submitted by Devraj Singh Gaidu and Vikas Verma, students of B. Tech 8th Semester, Mechanical Engineering Department is a bona-fide work done by them. This certificate does not necessarily endorse or accept any statement made, opinion expressed or conclusion drawn as recorded in the report. However it only signifies the acceptance of the report for the purpose for which it is submitted.

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Last but not the least we would than our parents and friends without whose constant support, achieving what we achieved would not have been possible.

DEVRAJ SINGH GAIDU
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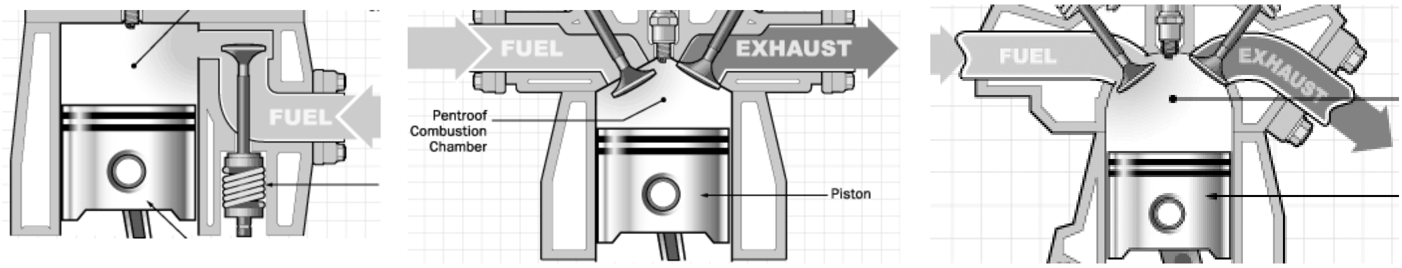
Notations

CAD	*Crank Angle in degrees*
ρ	*Density*
TDC	*Top Dead Centre*
BDC	*Bottom Dead Centre*
k	*Turbulent kinetic energy*
ε	*Turbulent dissipation*
P	*Pressure*
deg	*Degrees(Angular displacement)*
V	*Velocity*
x,y,z	*Rectangular coordinate*

Introduction

1.1. Pentroof Engines

The pentroof engine (also termed penta-engine) is the most common engine design used today by many manufacturers for engines producing relatively high horsepower for displacement, for both racing and passenger car engines.



Flathead Pentroof Hemi

Figure 1

The pentroof engine was developed in 1911 with the purpose of accommodating the benefits of both the flathead and the hemi engine into a single engine. For instance, the flathead provides good clearance volume but has a poor volume to surface area ratio whereas the hemi engine provides high volume to surface area ratio(lesser heat lost to the walls) but has a poor clearance volume. Moreover a hemispherical cylinder head is limited to only two valves per cylinder.

1.2. Multivalve Engines

A Multivalve engine is one where each cylinder has more than two valves. A multivalve engine has better breathing capacity and is able to operate at higher revolutions per minute than a two valve engine, delivering more power.

Cylinder head having 4-valves

Figure 2

Adding more valves increases valve area and improves the flow of intake and exhaust gases thereby enhancing combustion, volumetric efficiency and power output. Some engines are designed to open each intake valve at a slightly different time which increases turbulence improving the mixing of air and fuel at low engine speeds. Further they allow the same breathing capacity for a reduced valve lift which helps in obtaining a smaller clearance volume thereby improving the engine efficiency.

1.3. An introduction to the present study

Cleaner emission and improved fuel consumption are main environmental issues for automotive manufacturers. To achieve these goals at a low cost, engine emissions must be improved. One way to achieve stable combustion of lean mixtures is to generate turbulence during the induction process .

It is very important for the development of the ICE with high compression ratio to realize higher turbulence intensity and lean burn combustion. The in-cylinder flows of Internal Combustion Engine (ICE) that the flow structure generated by intake flows is related closely to the design and performance of the ICEs. One of the most important factors for stabilizing the ignition process and fast propagation of flame is the production of high turbulence intensity.

In this study, three different engine geometries having the inclination angle on the inlet side as 10 degree, 15 degree and 20 degrees have been created. The in-cylinder tumble motion during intake and compression strokes is then analyzed. This is followed by measuring and comparing the values of tumble ratio obtained in each of the three geometries. Finally, the computed turbulence intensities near the spark plug are compared and discussed.

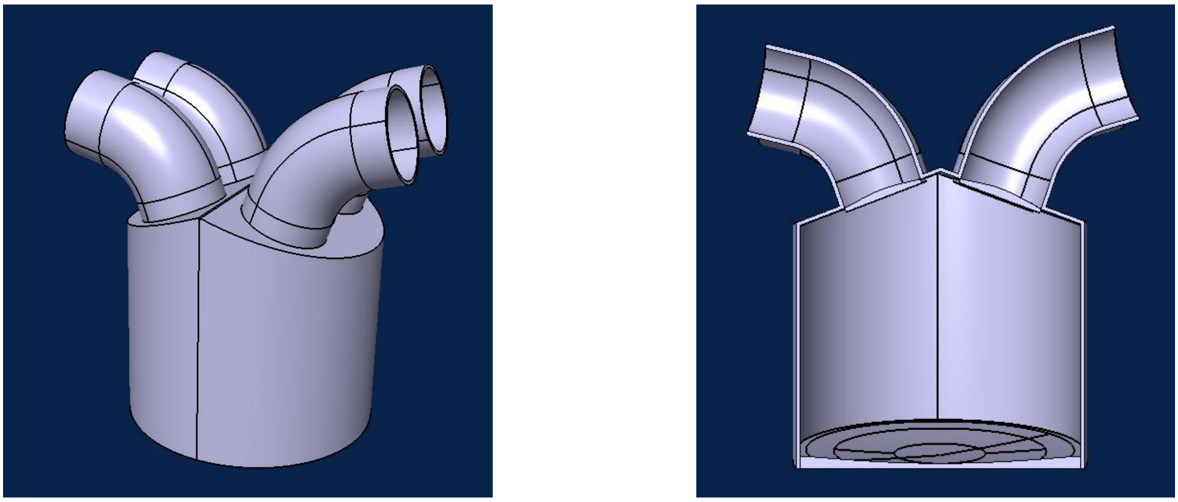


Figure 3

Literature Review

2.1. O. Bailly et al. (1999)

O. Bailly et al. (1999) carried out comparisons between computation and experiments of the intake and compression strokes of a four-valve engine. He described the formation of different vortices , in the form of both swirl and tumble, at different crank angle positions throughout the intake and compression stroke. It was shown that with the commencement of intake stroke two counter rotating vortices are setup in the combustion chamber, one below the inlet valve and the other diagonally opposite towards the exhaust valve. The variation of turbulence intensity is also presented with growth and degeneration of the vortices. It was shown that towards the end of compression stroke, the tumble motion decays into a large number of smaller eddies which are responsible for increased turbulence intensity.

2.2. A. Torres and S. Henriot (1994)

Torres and Henriot (1994) studied the combustion taking place in lean-burn 4-valve engines with different valve configurations. It was found that the valve configuration producing higher turbulence intensity at TDC during compression stroke was associated with a higher flame speed during combustion.

2.3. K. Lee et al. (2007)

K. Lee et al. (2007) investigated the effects of swirl and tumble flows on flame propagation in a 4-valve S.I. engine. Swirl was induced with the help of swirl control valves whereas tumble was induced by intake ports having different entry angles. It was observed that the size of the initial flame front was largest for the case having the highest induced tumble flow. Further, both the swirl and tumble flows were linked with flame convection resulting in improved flame development.

2.4. M. Achuth and P.S. Mehta (2001)

Achuth and Mehta developed a predictive model for tumble charge motion and turbulence in 4-valve pentroof engines. The mechanism behind turbulence generation was shown by explaining different stages involved. It was shown that the first peak in tumble ratio is achieved during the intake stroke which is followed by the development of a counter-rotating vortex which is responsible for a second peak occuring during compression stroke.The dependence of the tumble charge motion on the combustion chamber geometry was also studied. It was shown that although the intake system design mainly influenced the generation of tumble motion, the combustion chamber geometry plays a significant role in the stability and retention of the tumble motion during the compression stroke.

2.5. S. Falfari et al. (2014)

Recently S. Falfari et al. (2014) performed a study on the influence of geometrical engine parameters on induced tumble flow in small PFI engines. The geometrical parameters that were varied were : the intake duct angle and the piston shape. It was found that the more significant parameter influencing the peak tumble velocity was the intake duct angle. It was suggested that creating obstruction in the intake port could lead to a more favourable charge inlet angle at the expense of reduced flow rate. The effect of compression ratio on the tumble flow was also studied and it was found that it only influenced the degree of deformation of tumble.

2.6 P. Baker et al. (2014)

P. Baker et al. (2014) studied the variation in tumble ratio by modifying the intake port of a 4-valve pentroof engine. The modification involved insertion of tubes into the induction tracts which block a portion of the curtain area around the inlet valve. It was shown that for low operating speeds there was negligible reduction in flow rate and flow reversal. The inserted tubes prevented the formation of two intervening counter-rotating vortices and instead resulted in the formation of a single vortex having higher peak tumble ratio.

2.8. Conclusion

From these studies, few conclusions may be drawn. The first is that inducing more tumble flow in the combustion chamber is directly linked with better combustions with fewer emissions. Second, the induced tumble is greatly dependant on the cylinder geometry. From this we can infer that a combustion chamber geometry possessing higher values of induced tumble ratio and turbulence intensity at the end of compression stroke is likely to achieve better combustion during ignition. These conclusions will be used as guidelines for this study, and the objectives will be outlined in the following section.

Chapter 3

Objectives

This study investigates the effect of pentroof inclination angle on induced tumble in 4-valve S.I. Engine. The objective behind performing this study is to focus on enhancing the in-cylinder charge motion for achieving better combustion with fewer emissions. Since the flame propagation inside the chamber is highly reliant on the turbulence present at the end of compression stroke, this study focuses on observing the induced tumble flow and monitoring the corresponding variation in turbulent intensity inside the combustion chamber. In this study , the in-cylinder flow is inspected at small intervals of crank angle positions and the formation of different eddies & their transition from one state to other is recorded and discussed. The inferences obtained from the computational results of this study could be utilized for suggesting modifications in engine cylinder head designs.

This study also ensures that there is no flow separation occurring at the cylinder or piston walls. It also ensures that no local eddies are setup in the combustion chamber that could have adverse effect on the induced tumble flow.

Chapter 4

Problem Modelling

4.1. Problem Setup

The engine is a research four-valve-per-cylinder spark ignited configuration. This engine has a pent roof combustion chamber with central spark plug, separated intake ports and a flat top piston. Some of the engine specifications are summarized in Table.

Bore	80 mm
Compression Ratio	8.0
Combustion Chamber	Pentroof
Valves per cylinder	4
Engine Speed	2000 RPM
Intake Valve lift	8.035 mm
Intake Valve opening	0 CAD
Intake Valve closing	199 CAD

The Fig. 3 shows the combustion chamber geometry. Three such engine geometries having the inclination angle on the inlet side as 10 degree, 15 degree and 20 degrees have been created. The inclination angle of the pentroof at the exhaust side is common in all the three cases and is fixed at 20 degree. The other dimensions are similar to that of the combustion chamber considered by O. Bailly et al. [1]

The fluid domain enclosed inside the combustion chamber has the piston as one of its boundaries. Since the piston is subject to reciprocating motion, the boundary of the control volume is a moving boundary and requires to be modelled using special mesh motion techniques.

Apart from the piston, the inlet valves are also subject to motion which again causes the boundaries of the control volume to move. For this purpose it is required to breakdown the combustion chamber geometry into different meshing zones based on the type of distortion experienced by the zones.

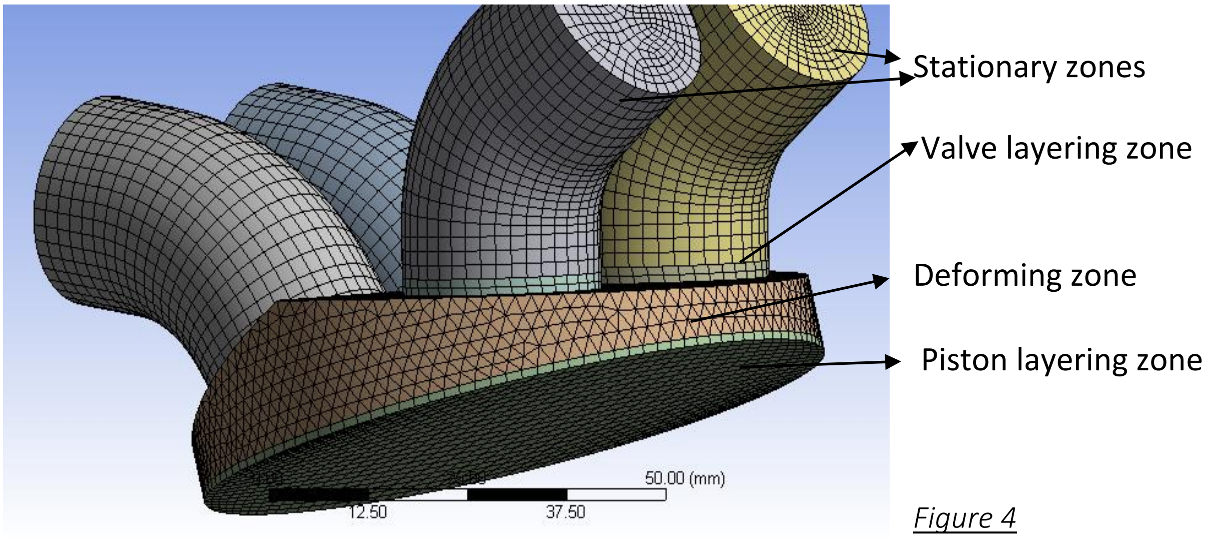


Figure 4

The region just above the piston is meshed with a layering zone. The purpose of this zone is to create new layers of cells as the piston moves away while expanding the control volume. A similar zone is created just above the inlet valves to accommodate the motion of the valves.

The upper region of combustion chamber is a deforming zone. This region encounters a lot of distortion due to motion of the valves and requires new cells to be created while eliminating the previous cells. The rest of the zones, including the inlet and exhaust ports are modelled with stationary mesh as there is no boundary deformation in the vicinity of these zones and unnecessary computation is supposed to be avoided

The piston is given the motion by creating a velocity profile wherein the velocity is a function of the crank angle .This enables to accurately model the motion of the piston. Similarly the inlet valves also have a separate velocity profile and the valve timing is dependant on the crank angles specified in the profile .

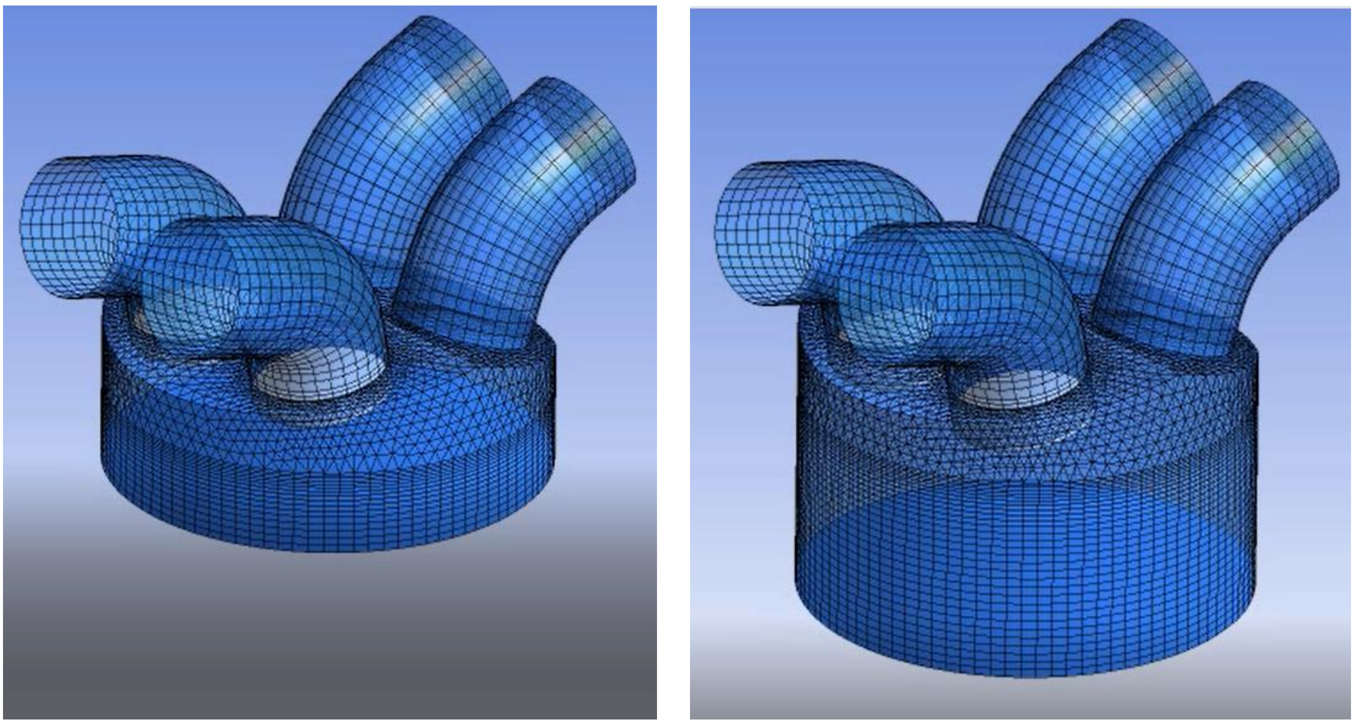


Figure 5

4.2 CFD Modelling

4.2.1 CFD in internal combustion engine

CFD analysis is capable of solving various case study including compressible or incompressible fluid flow, isothermal fluid flow, and also chemical reaction occurred in the flow. In the case of Internal Combustion (IC) engine, there are four methods available which differ from each other depending on the demanded results of users.

The analyses mentioned are :

1. Port Flow analysis
2. Cold Flow analysis
3. In-Cylinder Combustion simulation
4. Full Cycle simulation

In this study, we focus our attention on the Cold Flow analysis wherein we are investigating the tumble motion setup in the combustion chamber prior to ignition i.e. during the intake and compression strokes.

4.2.2 Flow-Governing equations

The basic approach is to calculate the velocity field inside the flow domain by solving the Navier-Stokes equations. In our present study, however, the control volume (enclosed by the cylinder walls and piston) does not have a constant volume as assumed while deriving the Navier-Stokes equations. Therefore, we use a more generic form of the conservation equations as derived from Reynolds Transport Theorem -

$$\frac{dM_{sys}}{dt} = \overset{\underline{\text{Storage}}}{\frac{d}{dt}\int_{cv} \rho d\forall} + \overset{\underline{\text{Transport}}}{\int_{cs} \rho V \cdot dA}$$

(Mass conservation equation from Reynolds Transport Theorem)

$$\frac{D}{Dt}\int_{sys} \mathbf{V}\rho \, d\forall = \frac{\partial}{\partial t}\int_{cv} \mathbf{V}\rho \, d\forall + \int_{cs} \mathbf{V}\rho\mathbf{V} \cdot \hat{\mathbf{n}} \, dA$$

(Momentum conservation equation from Reynolds Transport Theorem)

4.2.3 Properties of fluid

Air is considered as the fluid in the flow domain and is assumed to follow ideal gas law. Since we are required to model turbulence in our problem, it becomes essential to treat the viscoscity of air as a function of temperature instead of assuming it to be constant. The viscoscity of air is obtained using Sutherland's law. Sutherland's Law can be expressed as-

$$\mu = \mu_{ref} \left(T/T_{ref}\right)^{3/2} \frac{T_{ref} + S}{T+S}$$

For air at moderate temperatures and pressures :

$\mu_{ref} = 1.7984 \times 10^{-6}$ kg/m-s

$T_{ref} = 273.11$ K

S= 110.56 K

4.2.4 Turbulence Model

To close the system of mean flow equations, the model chosen is the k-ε turbulence model. The k-ε turbulence model has been shown to be suitable for complex shear flows involving rapid strain, moderate swirl, vortices and locally transitional flows (e.g. boundary layer separation, vortex shedding , stall in wide angle diffusers, etc.).This makes it the turbulence model of choice when modelling in-cylinder charge motion [1,2,4].

Model equation for *k :*
The equation for *k* contains additional turbulent fluctuation terms, that are unknown. Again using the Boussinesq assumption, these fluctuation terms can be linked to the mean flow. The following (simplified) model equation for *k* is commonly used.

$$\underbrace{\frac{\partial(\rho k)}{\partial t}}_{\text{Rate of increase}} + \underbrace{div(\rho k\mathbf{U})}_{\text{Convective transport}} = \underbrace{div\left[\frac{\mu_t}{\sigma_k}\, grad\ k\right]}_{\text{Diffusive transport}} + \underbrace{2\mu_t E_{ij}.E_{ij}}_{\text{Rate of production}} - \underbrace{\rho\varepsilon}_{\text{Rate of destruction}}$$

The Prandtl number σ_k connects the diffusivity of *k* to the eddy viscosity. Typically a value of 1.0 is used.

Model equation for *ε* :
A model equation for ε is derived by multiplying the *k* equation by(ε/*k*) and introducing model constants. The following (simplified) model equation for ε is commonly used.

$$\underbrace{\frac{\partial(\rho \varepsilon)}{\partial t}}_{\text{Rate of increase}} + \underbrace{div(\rho \varepsilon\mathbf{U})}_{\text{Convective transport}} = \underbrace{div\left[\frac{\mu_t}{\sigma_\varepsilon}\, grad\ \varepsilon\right]}_{\text{Diffusive transport}} + C_{1\varepsilon}\underbrace{\frac{\varepsilon}{k} 2\mu_t E_{ij}.E_{ij}}_{\text{Rate of production}} - C_{2\varepsilon}\rho\underbrace{\frac{\varepsilon^2}{k}}_{\text{Rate of destruction}}$$

The Prandtl number $\sigma\varepsilon$ connects the diffusivity of ε to the eddy viscosity. Typically a value of 1.30 is used. Typically values for the model constants $C_{1\varepsilon}$ and $C_{2\varepsilon}$ of 1.44 and 1.92 are used.

The turbulent viscosity is calculated from:

$$\mu_t = C_\mu \frac{k^2}{\varepsilon} \qquad C_\mu = 0.09$$

4.2.5. Boundary Conditions

The two inlet hoses leading to the combustion chamber have their open end modelled as pressure inlet zone. The entry condition is specified as constant pressure inlet of 1 atm having a temperature of 27 degree Celsius. The initial velocity is set to zero and the initial temperature and pressure are uniform in both the chamber and the intake port cell zones. The analysis is carried out for intervals for 3 degree crank angle rotation. Time step corresponding to this interval is .00025s.

4.2.6 Computation of flow parameters

1. Tumble Ratio

After solving the system of conservation and turbulence equations, we get the velocity field in the flow domain. It is then required to quantify the tumble inside the cylinder, for which we compute the tumble ratio as follows-

$$R_t = \frac{L.ta}{I.ta} \Big/ \frac{2\pi N}{60}$$

where L.ta is magnitude of fluid angular momentum with respect to tumble axis, and I.ta is moment of inertia of fluid mass about tumble axis.

By comparing the values of tumble ratio for different cases we are able to assess the tumble produced by the in-cylinder charge motion.

2. Turbulence Intensity

The turbulence intensity, also often referred to as turbulence level, is defined as:

$$I \equiv \frac{u'}{U},$$

where u' is the root-mean-square of the turbulent velocity fluctuations and U is the mean velocity .

If the turbulent energy, k, is known u' can be computed as:

$$u' \equiv \sqrt{\frac{1}{3}\left(u_x'^2 + u_y'^2 + u_z'^2\right)} = \sqrt{\frac{2}{3}k}$$

U can be computed from the three mean velocity components U_x, U_y and U_z as:

$$U \equiv \sqrt{U_x^2 + U_y^2 + U_z^2}$$

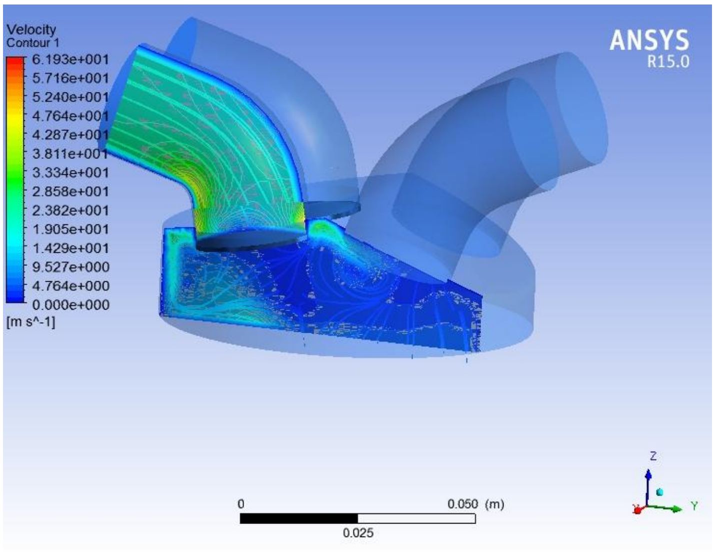
Chapter 5

Results and Discussion

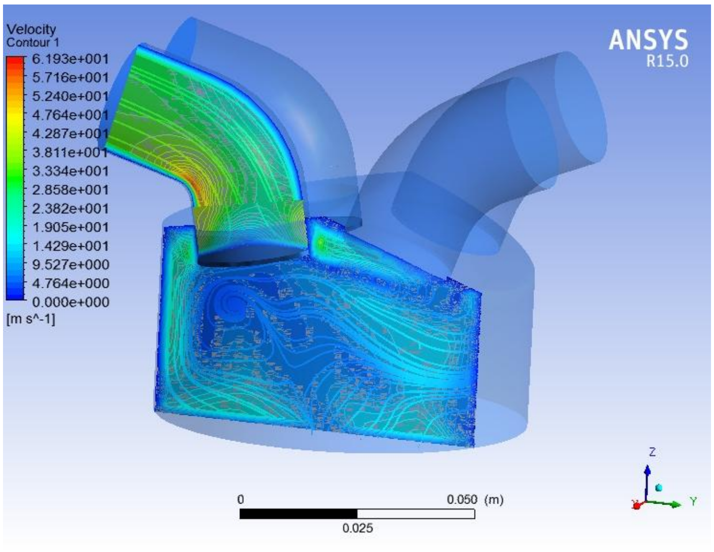
5.1. Different stages of turbulence generation

Phase 1 :
This phase extends from intake TDC to the instant of max valve lift (90 CAD). A small vortex begins to form which gradually grows in size having its centre below the intake valves. An initial peak value of tumble ratio is attained in this phase. This growth in size is accompanied by a subsequent reduction in angular velocity.

Phase 2 :
This phase exists between max valve lift instant(90 CAD) and BDC In this phase, a weaker secondary vortex of counter-rotating nature starts developing below the exhaust valves. This vortex interferes with the growth of the initial vortex causing a reduction in the overall tumble ratio.



36 CAD



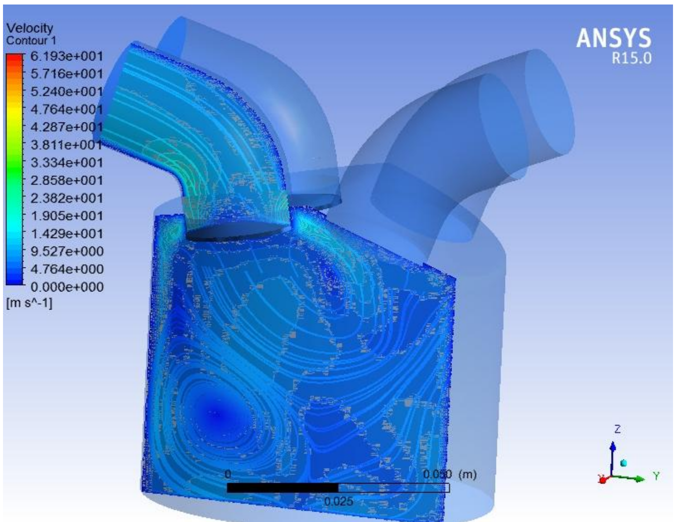
78 CAD

Phase 3 :

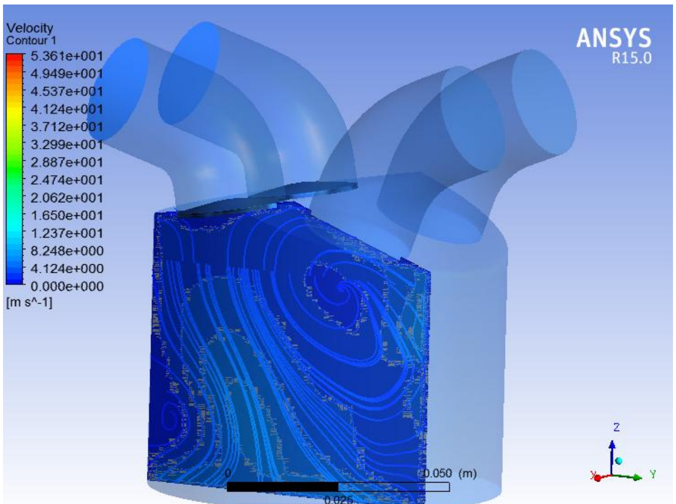
This phase exists between BDC and the instant of next peak tumble(300 CAD). The initial vortex continues to fade away whereas the counter-rotating vortex continues to grow becoming more prominent.. This results in a increase in tumble ratio that culminates in its peak value. It has been suggested that this phenomena is assisted by the compressive forces exerted by the upward moving piston[4]

Phase 4 :

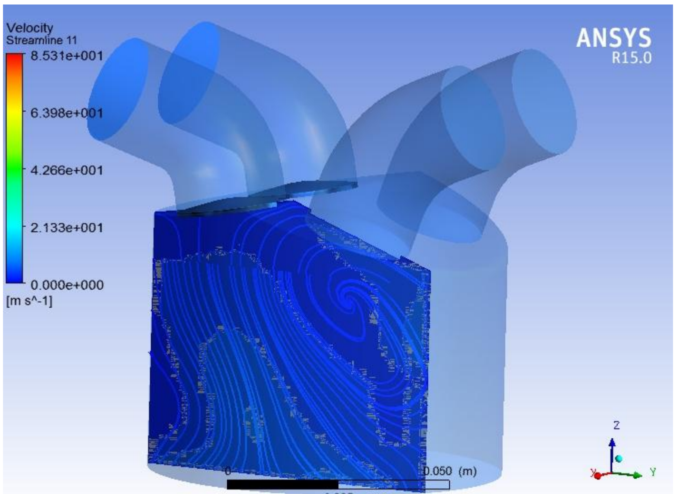
The occurrence of peak tumble ratio marks the onset of this phase, which extends until compression TDC. The vortex is compressed continuously between the cylinder head and the piston head which causes it to breakdown into a number of small vortices. This is responsible for an increase in the turbulence intensity towards the end of the compression stroke.



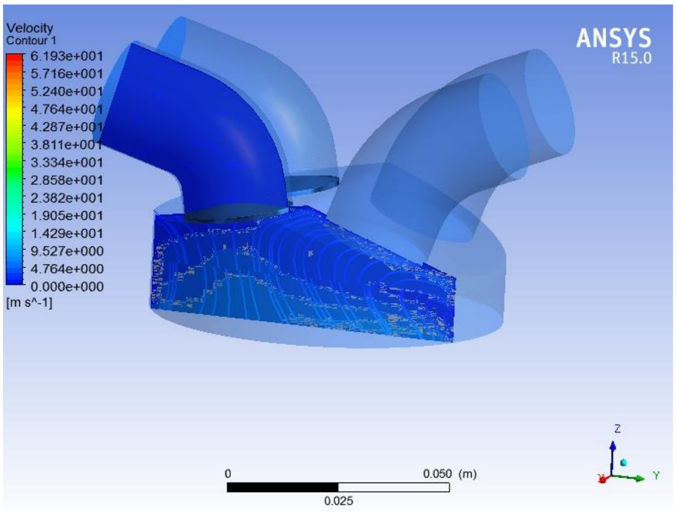
141 CAD



231 CAD



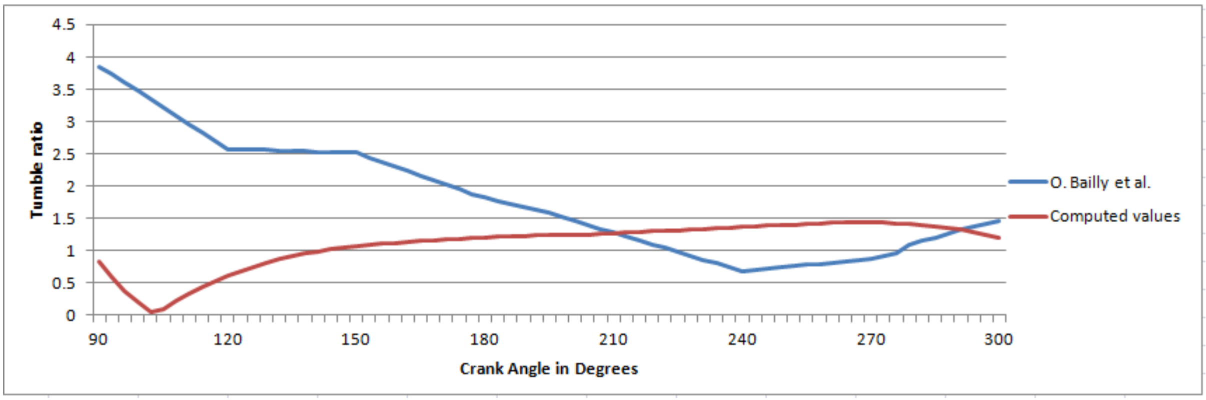
252 CAD



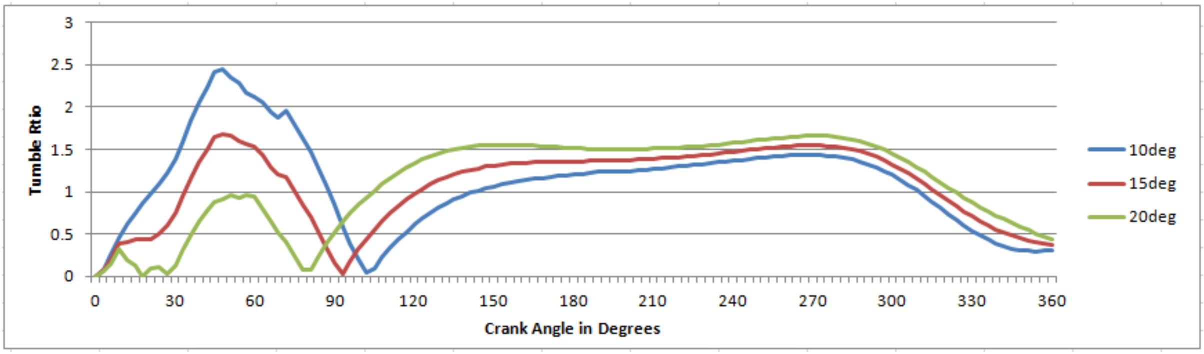
318 CAD

5.2. Tumble ratio characteristics

The obtained values of tumble ratio for different crank angles are shown in the graph below. The computed values were validated against the values of tumble ratio measured by O. Bailly et al.[1]



Another graph was plotted showing the variation of tumble ratio during compression stroke (180 CAD to 360 CAD) across the three different combustion chamber geometries having different pentroof inclination angles.



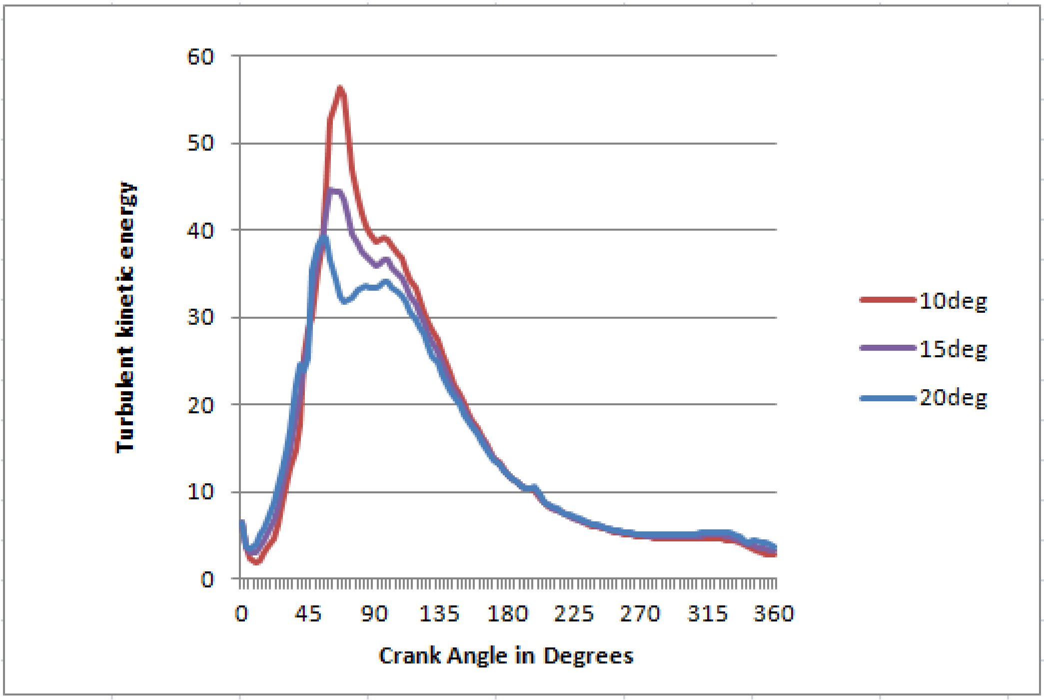
The following inferences were made by observing the above graph-

1. The curve attains two peak values - the first during the intake stroke and the second during compression stroke. This behaviour is in agreement with growth the growth and decay of vortices occurring in the combustion chamber.

2. The peak value of tumble ratio during the compression stroke increases with the pentroof inclination angle. An increase in the value from 1.435, for the geometry having 10 degree inlet angle, to 1.661,for the geometry having 20 deg inlet angle was observed.

3. The peak value of tumble ratio during the intake stroke decreases with the pentroof inclination angle. However this phenomenon is of little significance as it has negligible effect on the turbulence produced towards the end of compression stroke.

5.3. Turbulent kinetic energy characteristics



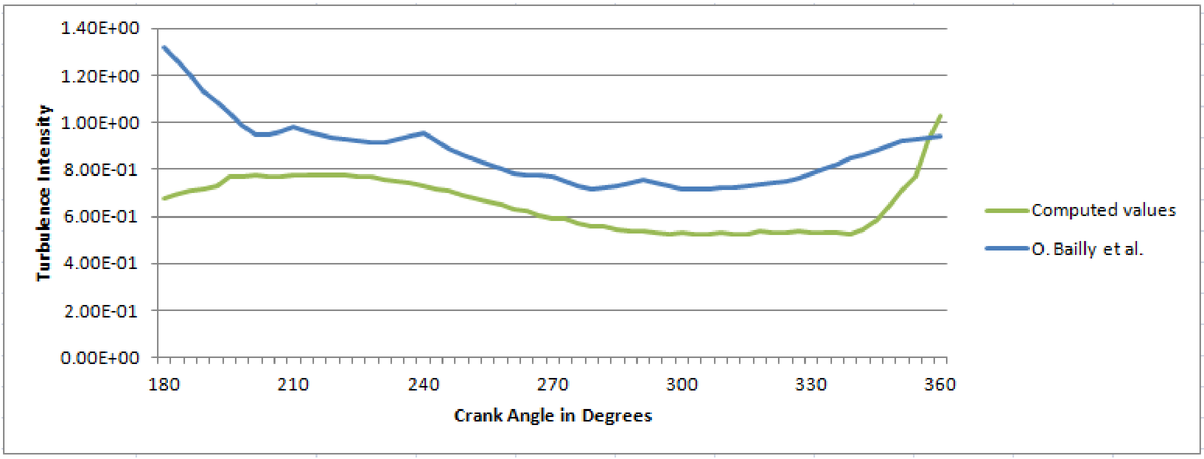
The variation of turbulent kinetic energy with the varying positions of crank angle is shown in fig. The trend followed by the curve is in agreement with the expected behaviour.

The curve attains peak value slightly before 90 CAD. This can be attributed to the fact that the destruction of the mean vortex gives rise to turbulence enhancement, by supplying the mean flow energy contained in the vortex . Moreover, it is seen in previous fig that there is a decline in the tumble ratio for the region near 90 CAD, which further supports this behaviour.

It is also observed that towards the end of the compression stroke, at about 30 degree before TDC, there is a small increase in the turbulent kinetic energy, due to breakdown of vortices post vortex spin-up phase.

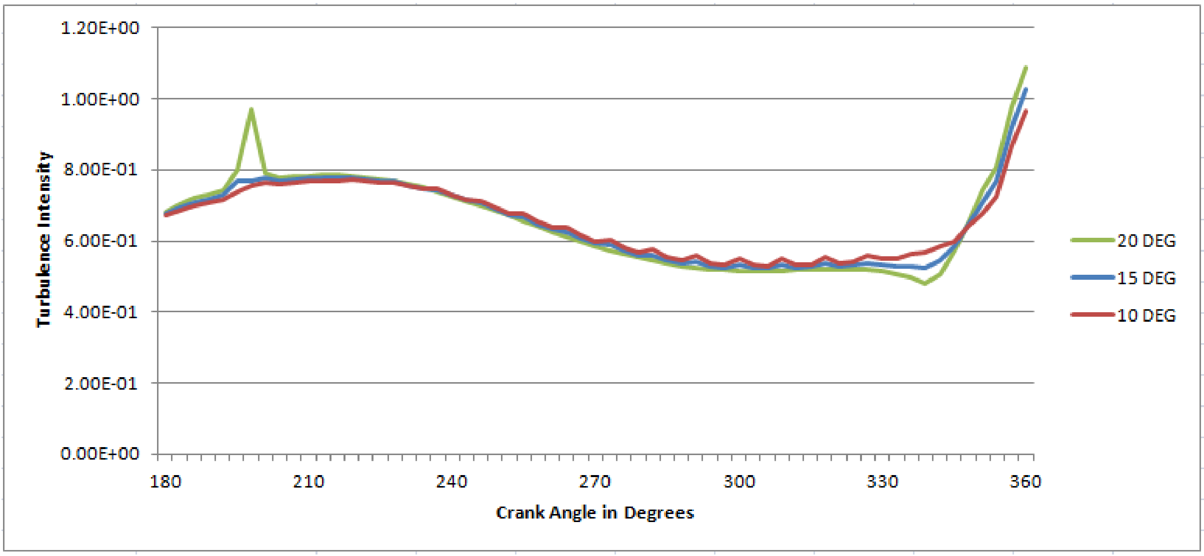
5.3. Turbulence intensity characteristics

To ascertain the level of turbulence in the combustion chamber during the compression stroke , the turbulence intensity has been computed. The computed values have been validated against the values obtained by O. Billy et al.[1]



From the graph it is seen that towards the end of the compression stroke there is an increase in turbulence intensity which peaks at the instant when ignition is to take place.

Since this increase in turbulence right before ignition is the desired characteristic, we have compared the turbulence intensities among the three different geometries having different pentroof inclination angles to assess which geometry produces highest turbulence at compression TDC.



5.7. Conclusion

The work performed depicts how modification in the inclination angle of pentroof of an S.I. engine affects the in-cylinder flow characteristics leading to conditions favourable for better combustion. Results show better turbulence characteristics were obtained for the cylinder having higher pentroof inclination angle. The variation in flow parameters having influence over impending combustion has been shown with the help of computational resources and the factors leading to such behaviour of these parameters have been discussed in this study

The conventional approach of designing the cylinder geometry focused on establishing a trade-off between minimum surface area to volume ratio of the cylinder and minimum size of the clearance volume, but the outcomes of this study add to the latest developments made in this field which focus on the impact of the cylinder head geometry on the in-cylinder flow characteristics, suggesting the importance for its consideration. The outcome of this study shows that an increase in inlet angle of pentroof is associated with the production of higher amount of induced tumble flow. The higher tumble produced is further linked with a higher turbulent intensity towards the end of the compression stroke. This high turbulent intensity, right before ignition, is significant as it results in a higher value of flame speed, thereby ensuring stable combustion with fewer emissions.

References

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