Study on Performance Characteristics of Scuderi – Split Cycle Engine

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Abstract

Many decades, the internal combustion engines are the promising prime movers on the earth. Many technologies like Gasoline Direct Injection (GDI), Common Rail Diesel Injection (CRDI) have been invented to enhance the performance and to reduce the pollutants from the internal combustion engines. Many split cycle concepts have developed but could not succeed. For the first time the Scuderi group has invented a new concept of split - cycle engine, in which the air is compressed in one cylinder and transferred to the other cylinder, which is in phase with the first cylinder through the crossover passage and ignited when the piston is moving away from TDC promising to enhance the performance and reduce the pollutants. So a detailed computational study of the Scuderi split cycle engine is essential for better understanding of the concept.

In this study work, the CFD analysis is carried out for both the conventional spark ignited engine and the Scuderi split cycle engine of approximately same volume and for same amount of charge. The geometrical models required for the analysis have been modeled in CATIA V5R18 environment and the CFD analysis is carried out in the 3-D Ricardo VECTIS platform as it has inbuilt automatic dynamic mesh generation. The analyses are carried out to understand the combustion phenomena for different speeds. The performance curves of the engines are plotted and compared. The obtained results are validated by the analytical results..

From the analysis, it was found that there is increase in the indicative thermal efficiency by 5% compared to the convectional engine. From the performance curves of the Scuderi engine, it was observed that the torque is high at low speed resembling the diesel engines. For the same amount of the charge, the torque and power produced by the Scuderi engine are 15-20% higher than conventional SI engine when compared.

Key Words: I.C. Engines, Split Cycle Engines, Scuderi Engine, VECTIS, CFD.

Nomenclature

k	Turbulence kinetic energy, m^2/s^2
v	Velocity, m/s
r_c	Compression ratio
ε	Dissipation rate, m^2/s^3
m_a	mass flow rate of air kg/sec
m_f	mass flow rate of fuel kg/sec
ρ	Density, kg/m^3

Abbreviations

ABDC	After Bottom Dead Centre
ATDC	After Top Dead Centre
BDC	Bottom Dead Centre
BMEP	Brake Mean Effective Pressure
BTDC	Before Top Dead Centre
CFD	Computational Fluid Dynamics
CA	Crank Angle
IC	Internal Combustion
IVC	Inlet Valve Closing
SI	Spark Ignition
TDC	Top Dead Centre

1. INTRODUCTION

Split cycle engines are the ones which separate the four strokes intake, compression, power and exhaust of conventional engine in to two separate paired strokes to occur in two separate cylinders namely compression and power cylinders. In the compression cylinder the Intake and compression of air takes place. The compressed air is then transferred through a crossover passage from the compression cylinder in to the power cylinder. In the power cylinder the fuel is injected and burned to undergo combustion and exhausted stroke. A split cycle is an air compressor on one side and the combustion chamber on another side.

When compared to the convention spark ignition engine, the split cycle engine will be having two cylinders namely compression and power cylinders, two pistons which are connected to the crank shaft with some phase difference. The split cycle engine will produce one power stroke for every one revolution of the crank shaft when compared to the conventional engine producing one power stroke for every two revolutions of the crank shaft.

The two main drawbacks of split cycle engine are poor breathing and low thermal efficiency. The breathing problem was caused by high pressure gas trapped in the compression cylinder. This trapped high pressure gas should be re-expanded before another charge of air could be drawn in to the compression cylinder, effectively reducing the engine's capacity to pump air and resulting poor volumetric efficiency.

The thermal efficiency of the split cycle engine has always been significantly worse than a conventional Otto cycle engine. This is because they all tried to fire Before Top Dead Centre (BTDC). In order to fire BTDC in a split cycle engine, the compressed air trapped in the crossover passage is allowed to expand into the power cylinder as the power piston is in its upward stroke. By releasing the pressure of the compressed air, the work done on the air in the compression cylinder is lost. The power piston then recompresses the air in order to fire BTDC. By allowing the compressed gas to expand in the power cylinder the engine needs to perform the work of compression twice leading to low thermal efficiency.

Scuderi – split cycle engine is named after the concept proposed by Carmelo Scuderi. Scuderi - split cycle engine works same as split cycle engines with slight modifications in order to overcome the drawbacks of the ordinary split cycle engines.

Scuderi - split cycle engine consists of two cylinders namely compression cylinder and the power cylinder. Each cylinder has a piston. The air is drawn in to the compression cylinder during the suction process with inlet valve open and compressed to a very high temperature and pressure with both the valves closed. The compression and power cylinders are connected through a cross over passage to transport the compressed air in to the power cylinder. As the air is only compressed without any mixing of charge high compression ratios can be achieved. The unique valve design of the crossover passage with outwardly opening valves allow to have the compression ratios of range 75:1-100:1. These high compression ratios and unique valve design will overcome the drawback of low volumetric efficiency in the ordinary split cycle engines.

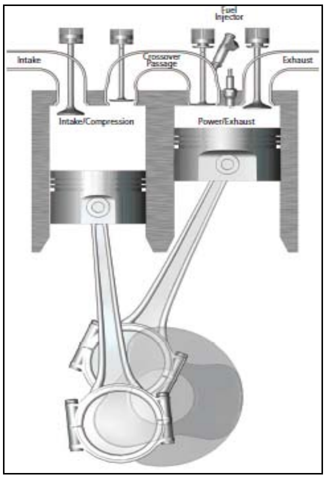

Fig.1 Scuderi split cycle engine[9]

In Scuderi engine as shown in the Fig 1.1, the two pistons are connected to the crank shaft with two different crank throws and are in phase with each other. The power piston will be leading the compression piston by 25^o of crank angle. i.e., when the power piston is at the TDC the compression piston is approaching the TDC and when the compression piston reaches TDC, power piston is moving away from the TDC.

Another modification proposed by Scuderi is firing after top dead centre. All the split cycle are fired before top dead centre which leads them to low thermal efficiency. The method of firing ATDC will reduce the work of recompressing the gas, which generally takes place in ordinary split cycle engines. Due to high compression ratio the high pressurized gas will be entering into the power cylinder with high turbulence, this result in very rapid atomization of the fuel and creates a fast flame speed or combustion rate faster than any previous methods. This will enhance the thermal efficiency of the split cycle engines.

The Scuderi engine is at the initial stages of its development and a lot of research work is being carried out to develop the prototype model. A few good studies are carried out by the researchers in this way. Scuderi Group [1] have made a comparative CFD analysis study between the conventional SI engine and the Scuderi split cycle engine in one dimensional GT power software. They claim that the compression ratio, expansion ratio, TDC phasing between the compression piston and the expansion piston , crossover valve duration and combustion duration as significant variable affecting the engine performance and efficiency of the split cycle engine.

Scuderi and Stephen[3] suggest the method to improve the part load efficiency for the Scuderi split cycle engine.. They claim that during the full load operation both the crosses over passages are utilized. That is during a single rotation of the crankshaft the crossover valves corresponding to both crossover passages are actuated, both the fuel injectors inject fuel into the exit end of their respective crossover passage. At part-load operation, the ECU will select at least one of the cross over passage to utilize. For example, at half load the compression cylinder intakes a mass of air. At half load, this mass of air can approximately match the maximum mass of air that either one of the crossover passages is designed to process during a revolution of the crank shaft. Accordingly the ECU selects only one crossover passage and injects fuel at only one crossover passage.

Scuderi Group [2]has studied on six various configurations of the helical passages, combinations of tangential or radial runner sections plus counter clockwise helical, clockwise helical or directed end sections. The directed end sections do not impart any specific rotational spin to the fuel air mixture as it enters the expansion cylinder. The computer study was carried out and a general trend was observed that the higher swirl producing passage also produced higher levels of turbulent kinetic energies. The dual tangential helical passages having end sections rotations in the same directions produced both the highest level of swirl and the turbulent kinetic energy.

Sapsford, M [4] has performed the CFD simulation studies to validate the RTFZ combustion model in VECTIS.Two representative diesel engines were chosen as subjects; a large truck engine and a small HSDI research engine. Each of the simulation programmes consisted of 6 cases forming a complete injection time swing. CFD simulation results were compared directly against engine tests. . For the large truck analysis, in all six cases both the measured and the simulated pressure curves appear smooth over the ignition period, but a slight under prediction of pressure was observed between 20^o-50^o ATDC for this they stated it may be due to the insufficient accuracy in specifying the wall boundary temperatures. The trend of NO_X decrease with the delay in the injection is perfectly predicted by the CFD simulation. From the HSDI simulation they found out that the in-cylinder pressure is matching to the simulation at the early stages of combustion but there is a slight gradient. For this they stated that this is due to more significant pre-mixed burn and quicker flame development, most likely due to the effective fuel/air mixing produced by highly swirling flow in small engine.

The literature reveals that only a few computational studies have been carried out on this engine. In the current work, CFD analysis is carried out at different speeds both on conventional SI and Scuderi engine in order to obtain performance characteristics and the results are compared.

2. GEOMETRIC MODEL

In the present study, the CFD analysis is carried out on both the convention SI engine and the Scuderi engine. The technical specifications of conventional and Scuderi engine are tabulated in the table 1 and table 2 respectively.

Parameter	Value
Engine type	4 - Stroke
Number of cylinders	2
Bore	101.6 mm
Stroke	101.6 mm
Connecting rod length	243.8 mm
Displacement volume	0.924 L
Clearance volume	0.118 L
Compression ratio	8:01
Inlet port diameter	30 mm
Exhaust port diameter	26 mm
Air- fuel ratio	18:01

Table 1. Specifications of conventional model

Parameter	Compression Cylinder	Power Cylinder
Engine type	4 – stroke	4 - stroke
Bore	112.0 mm	101.6 mm
Stroke	120.2 mm	141.1 mm
Connecting Rod Length	243.8 mm	235 mm
Displacement Volume	1.007 L	1.14 L
Clearance Volume	0.010 L	0.010 L
Compression Ratio	100:01:00	120:01:00
Intake port Diameter	30 mm	26 mm
Cross over passage Diameter	26 mm	30 mm

Table 2. Specifications of Scuderi engine model

The engine models are modelled in CATIA V5R18 environment as shown in the Fig. 2 and Fig 3. They are modelled as a closed surface in the Generative Shape Design (GSD) environment of CATIA V5R18.Two models, one with intake port geometry and the other without intake port geometry are modelled for both conventional and Scuderi engine. The CFD analysis is performed in three stages, suction, compression and combustion. The suction analysis uses the model one with intake port geometry and the model without intake port is used for both compression and combustion process.

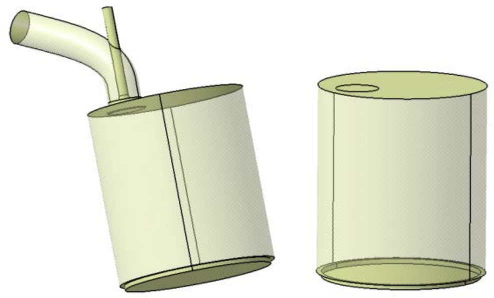

Fig. 2Conventional SI engine models

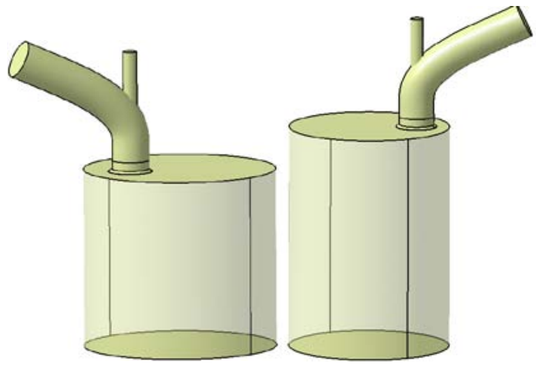

Fig. 3 Compression and power cylinder models of Scuderi split cycle engine

The CFD analysis is carried out in 3DRicardo VECTIS platform which has inbuilt dynamic mesh concept specially introduced for the engines. The methodology followed by VECTIS for CFD analysis is displayed in the Fig. 4. The mesh used for the analysis is a Cartesian hexahedralmesh. While meshing the models at different crank angles are developed and are meshed separately and fine mesh is employed at the regions of interest. VECTIS has inbuilt dynamic mesh, which uses the concept of moving mesh, in this the meshed geometries at different crank angles are linked together, so that when the piston is moving towards TDC and is

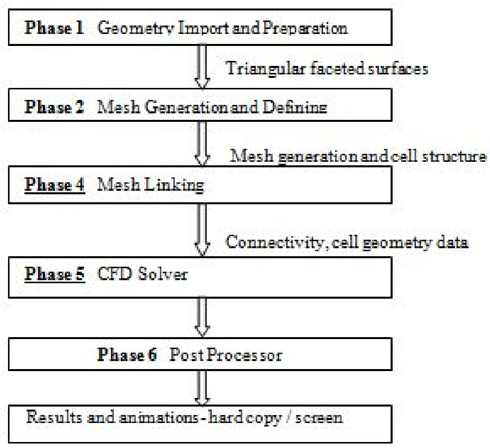

Fig. 4 CFD methodology

at 30^o BTDC, the mesh of the geometry 20^o BTDC will be distorted and is replaced their so that it will get compress. This is known as reverse mesh motion.

3. GRID INDEPENDENCE STUDY

Grid size plays an important role in both convergence and accuracy of the solution. The use of high-density mesh improves the accuracy of simulation, but is computationally expensive. Hence grid independence studies are performed to obtain an optimized mesh size. The table 3 shows the iterations carried out for different mesh size and their effect on the accuracy of the result during compression analysis.

Mesh Size	Pressure from VECTIS [bar]	Pressure from analytical [bar]	Difference in %
6.0 mm	13.24	15.88	16.6
5.0 mm	14.18		10.7
4.0 mm	14.92		6.04
3.0 mm	15.56		2.01

Table 3. Grid independency study details

From the grid independency study we found that the grid with 3 mm is producing accurate results and it was chosen for the analysis.

4. BOUNDARY CONDITIONS

The boundary conditions for different regions define their functionality with respect to other regions. Improper boundary conditions will results in errors during the analysis. The default boundary condition is wall boundary in VECTIS, which acts like a separating-surface between the inside and outside of the geometry. The table 4 show the various boundary conditions and their functionality. The boundary conditions are same for the conventional and Scuderi engine.

No:	Boundary Name	Type	Function
1	Cylinder Head	Wall	Stationary wall
2	Liner	Interpolated motion	Moves with the piston
3	Piston	Motion	Motion as per con rod length
4	Intake port	Wall	Stationary wall
5	Intake opening	Input/output	Allows the fluid
6	Valve	Motion	Moves as specified

Table 4. Boundary conditions for the models

5. SOLUTION PROCEDURE

The analysis for the conventional engine is carried out in three stages, suction analysis, compression analysis and combustion analysis. The suction analysis is carried out from the inlet valve opening to inlet valve closing i.e., from 17^{o} ATDC to 210^{o}ATDC with the model having intake port geometry. The restart file generated by the suction analysis is used to start compression analysis. The compression analysis starts from IVC to 360^{o} ATDC. This produces the motoring curve. The combustion analysis starts just before the ignition and ends at 540^{o} ATDC. The restart file generated by the compression analysis is used to start combustion analysis. The restart file will model the conditions when it was written. All the inputs required for the analysis are entered in phase-5.Various inputs for the combustion analysis of conventional engine are drawn in the table 5.

Parameter	Value
Time base	4 – stroke
Equation coupling	Pressure Correction Solver
Solution algorithm	PISO
Iteration per step	5
Speed	3000 rpm
Turbulence model	k-ε model
Turbulence velocity	4.57 m/sec
Length scale	0.0025 m
Wave fluid property	Gasoline
Ignition	333^{o} – 336^{o}
Combustion model	RTFZ
Initial pressure	9.86 bar
Initial temperature	540 K

Table 5. Solver inputs for conventional engine model

The analysis for the Scuderi split cycle engine model is carried out in four stages, suction analysis, compression analysis, fuel spray analysis and combustion analysis. The suction analysis is carried out from the inlet valve opening i.e., around 7^{o} ATDC to IVC at 186^{o} ATDC.The compression analysis from IVC to 360^{o} ATDC. The suction and compression takes place in compression cylinder and the compressed air is transported to power cylinder. The fuel spray analysis starts at 6^{o}ATDC to 20^{o} ATDC in power cylinder. The combustion analysis starts at 20^{o} ATDC to 540^{o} ATDC.

The analysis of the Scuderi split cycle engine is critical job as it has two pistons moving at different direction for some point of time and in same direction for some point of time and at different speeds. So modelling the Scuderi engine with two cylinders at a time and giving different motion to the two pistons is difficult in the VECTIS environment because the VECTIS uses the automatic dynamic mesh concept in which the cylinder model is for different crank angles are generated and meshed them separately. During the analysis in order to generate the dynamic mesh it calls upon the mesh at different crank angles at their respective times. So while generating the geometry at different crank angels it considers only one piston motion. So in order to overcome this, the two cylinders of the Scuderi split cycle engine is modelled separately and analysed separately. That is the suction and compression analysis is done in the compression cylinder, the combustion and the exhaust analysis is done in the power cylinders.

The compression ratio of the compression cylinder is 100:1 as a result the clearance between the piston and the cylinder head is about one mm, this results in the distortion and damage of the cells at that instance leading to mesh related errors. So the compression cylinder is modelled without the cross over passage and made some assumptions looking through the process of the compression analysis of the Scuderi engine.

In the compression stroke of the Scuderi engine, the piston of the compression cylinder starts from the BDC towards the TDC. During this motion, when the piston is at 30^o before TDC of the compression cylinder the cross over passage valve opens and the compressed air is pumped into the power cylinder. As a result there will be drop in the pressure in compression cylinder. At the same point when the cross over valve opens in the power cylinder, the piston of the power cylinder is at 5^o BTDC of power cylinder and is moving toward the TDC. This will cause a slight pressure rise in the compression cylinder compensating the drop occurred before. Once the power piston reaches the TDC and starts its down travel then both pistons are moving in opposite directions sweeping out the same amount of volume. During this process we can assume that there will be no pressure rise in the compression cylinder and it is merely pumping the compressed air from compression cylinder to the power cylinder at almost same pressure.So by considering the assumption made above, the compression analysis is carried out from the IVC position to the point of opening of the cross over passage valve, which is at 30^o BTDC. From the analysis the pressure at this point is 27.3 bar at 780 K as shown in the Fig. 5.

In the power cylinder, the combustion process is analysed in two stages. One is fuel spray analysis and the other is combustion analysis. In the fuel spray analysis the fuel is injected in to the power cylinder when the piston is at 6^o ATDC, moving away from TDC by maintaing constant pressure in the power cylinder. This process continues up to 20^o ATDC. The restart file is generated at this point and is used to restart the combustion analysis. In the combustion analysis the charge is ignited at 20^o ATDC and there will be a rapid combustion due to the high turbulence developed. The solver inputs for the combustion in power cylinder of Scuderi engine model is shown in table 6.

Parameter	**Value**
Time base	4 – stroke
Equation coupling	Pressure Correction Solver
Solution algorithm	PISO
Iteration per step	5
Speed	3000 rpm
Turbulence model	k-ε model
Fuel injected	4.82×10^{-5} kg
Ignition	380^o – 383^o
Combustion model	RTFZ
Initial pressure	30 bar
Initial temperature	780 K

Table 6. Solver settings for combustion analysis of Scuderi split cycle engine model

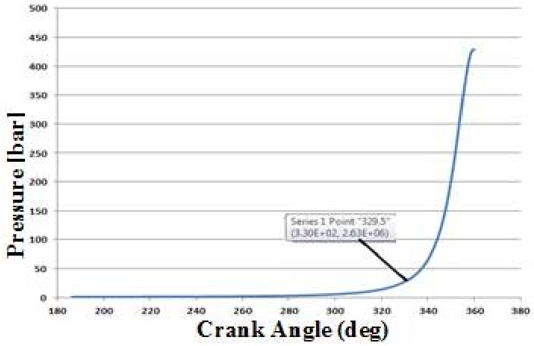

Fig. 5 pressure rise in compression cylinder of Scuderi engine

6. VALIDATION OF RESULTS

The validation of conventional engine is carried out by two approaches. One is comparing the results obtained from VECTIS by Wave model and the other by analytical relations. The Fig. 6 shows the wave model of the conventional engine. Various inputs for the conventional engine in the wave model are shown in the table 7.

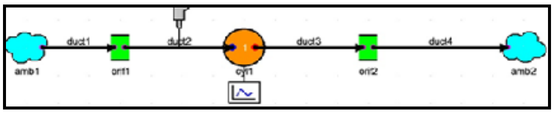

Fig. 6 Wave model of conventional engine

The peak pressure obtained from the VECTIS for the conventional engine is 40bar. The peak pressure obtained from the wave model for the conventional engine of same dimensions is 42 bar. The difference between the two peak pressures is 2 bar, a small difference which is acceptable. The peak pressure from wave model is high compared to 3-D VECTIS model because the VECTIS model considers the effects in all three directions unlike the wave 1-D model. The peak pressure obtained from the analytical relations is calculated as

Parameter	**value**	**Parameter**	**value**
Bore	101.6 mm	combustion	SI Wiebe
stroke	101.6 mm	Engine speed	3000 rpm
Connecting rod	243.8 mm	Air fuel ratio	18:01
Compression ratio	8:01	50% burn point	10^o ATDC
		10% - 90%	24^o CA

Table 7. Solver settings for conventional engine wave model

Volume of air sucked (V_a) $= \frac{(\eta_v \times V_{cyl} \times rpm)}{2 \times 60}$

$= 0.8 \times 9.4 \times 10^{-4} \times 3000/120$

$V_a = 0.0199$ m^3/ sec

Density of inlet air $(\rho) = ({}^{P}/_{R.T})$

$= (1 \times 10^5 / (0.287 \times 320))$

$= 1.088$ kg / m^3

$m_a = V_a \times \rho = 0.0199 \times 1.088 = 0.2173\ kg/sec$

$m_f = (m_a/A/F) = \frac{0.002173}{18} = 0.0012\ kg/sec$

$m_f = (m_f \times 60 \times 2)/3000 = 4.82 \times 10^{-5}\ kg$

$Q = m_f \times CV = m \times C_v \times \Delta T$

$0.9 \times 4.82 \times 10^{-5} \times 42000 = 10.94 \times 10^{-4} \times 1.135 \times \Delta T 1464.1 = \Delta T$ (K)

$T_3 = \Delta T + T_2 = 1464.1 + 654.8 = 2119.2$ K

$P_3 = \left(\frac{T_3}{T_2}\right) \times P_2 = \left(\frac{2119.2}{654.8}\right) \times 15.88 = 48.12$ bar

The peak pressure obtained from the analytical relations is 48.12 bar, this will be high because the analytical calculations are carried out for the fuel air cycle, which does not consider the losses due to many factors like ignition time delay, incomplete combustion of the fuel, progressive combustion, blow down losses, heat transfer into the walls of the combustion chamber on the other and the analysis represents the actual cycle of the Otto cycle. The difference is about 8.12 bar which is 16% falls in the acceptable range which is about 15-20%.

7. RESULTS AND DISCUSSIONS

The Fig 7 and Fig 8 represents the variation of the in cylinder pressure with respect to the crank angle for 3000 rpm in conventional and Scuderi engine respectively.

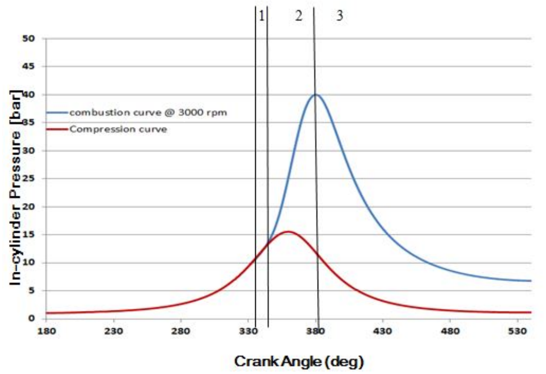

Fig. 7 In cylinder pressure in conventional engine model

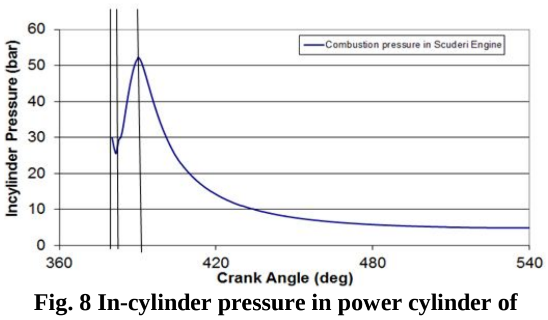

Fig. 8 In-cylinder pressure in power cylinder of Scuderi split cycle engine model

The peak pressures obtained from the conventional and the Scuderi engine models are 40 bar and 51.8 bar respectively. The Fig 7 show the three different stages of combustion, the first stage is the ignition lag or preparation stage, the second stage represents the flame propagation and the third stage represents the after combustion process**.** From the Fig 7 we see that though the charge is ignited at 332^{o} CA, the combustion curve follows the compression curve without any sudden change and we can notice that there is no gap which implies the correct combustion process and the results obtained are correct. We can also see that there are no irregularities during the pressure rise which indicates there is no knocking taking place during the combustion process. We also observe that there is no rapid increase in pressure indicating the normal combustion. The peak pressure is occurring just after TDC resembling the actual combustion process.

From the Fig 8, we can see that the ignition lag or preparatory stage has reduced as the initial temperature, pressure, and the turbulence of the charge is high compared to the conventional engine which are the prime factors influencing the flame growth. We also observe that the flame propagation stage is reduced and there is a rapid rise in the pressure. This is due to the high turbulence in the cylinder which aids in the propagation of the flame front rapidly. As the combustion is happening rapidly there will be huge forces developed, in order to sustain those forces the compression and the power cylinders are off-set and also the engine is made bulkier compared to the conventional engine.

From the Fig8, we notice that there is a pressure drop at 380^{o} in start of the combustion process. This is obvious because when the charge is ignited, there will be a time lag between the ignition point to the point of flame development. During this stage the piston is moving down which increases the volume and so there is a drop of pressure unlike in conventional engine during this stage of flame development the piston is moving towards the TDC which decreases the volume of combustion chamber.

The Fig. 9 and Fig.10, display the combustion progress taking place in the cylinders at two planes in conventional and Scuderi engine respectively.

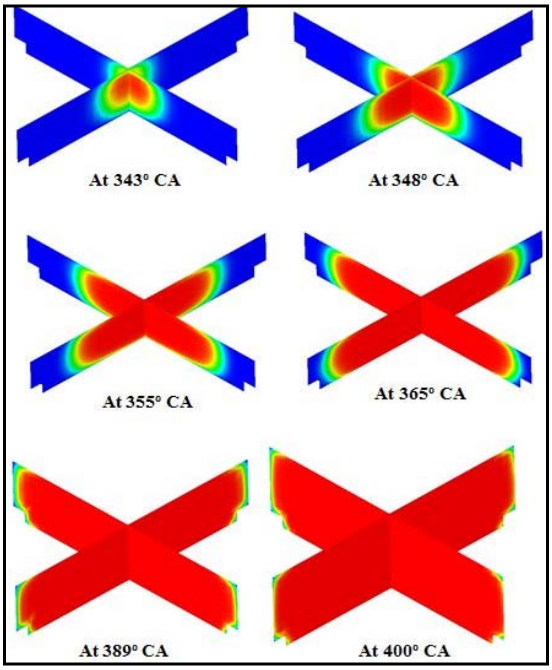

Fig. 9 Combustion progress in conventional engine

The combustion in the conventional engine starts at 334^{o} CA and the flame touches the walls of the combustion cylinder at 400^{o}CA. whereas in the combustion progress in Scuderi engine starts at 380^{o} CA and the flame touches the walls of the combustion cylinder at 401^{o} CA. This indicate that the combustion process in the Scuderi engine is rapid compared to the conventional engine.

This may be due to the initial high temperature, pressure and the turbulence in the combustion cylinder.

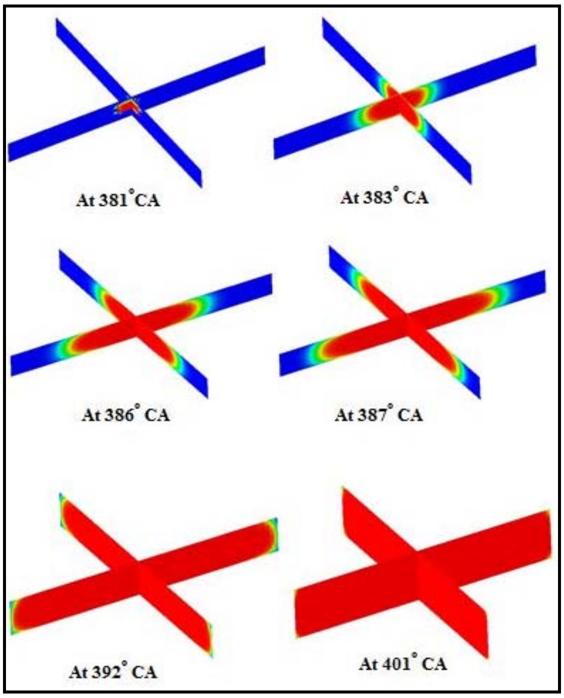

Fig. 10 Combustion progress in power cylinder of Scuderi engine

We also notice that the combustion chamber volume of the Scuderi engine is very small compared to the conventional engine. The combustion duration in the Scuderi engine from the literature is 23^o CA and from the analysis it is 21^o CA, this justifies that the analysis is correct.

7.1 p – V curves

The Fig. 11shows the variation of in cylinder pressure with respect to the cylinder volume.

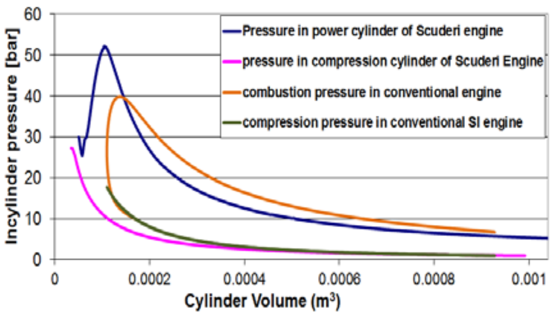

Fig. 11 p-V curves of conventional engine and Scuderi engine

The areas under the curves represent the work done on the piston. It is the total amount of work done in the system. From the above plot the work done in the compression stroke and the power stroke of the Scuderi engine is more compared to the conventional engine. The areas under the curves are calculated by using sketch tracer concept in CATIA environment as shown in the Fig. 12. From the Fig. 12, it is clear that the area under the curve for the Scuderi engine is 18% more compared to the conventional engine. Hence, the work done by the Scuderi engine is more than the conventional engine.

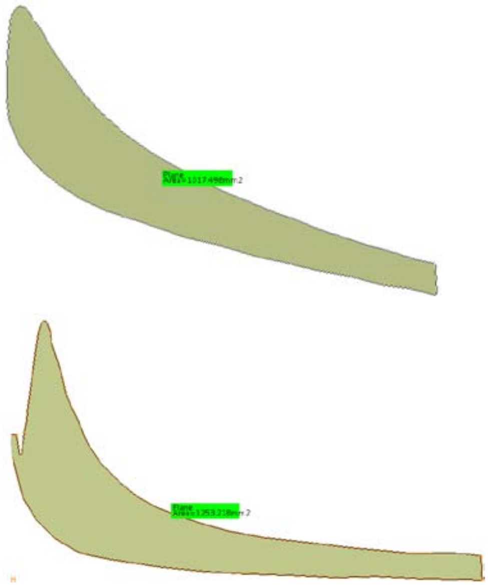

Fig. 12 Area under the p-V curves

7.2 Power and Torque curves

The analysis is carried out for different speeds of the engine varying from 1500 rpm to 5000 rpm in order to plot the performance of the engines**.** The Fig.13 shows the power and torque curves of both the conventional and the Scuderi engine

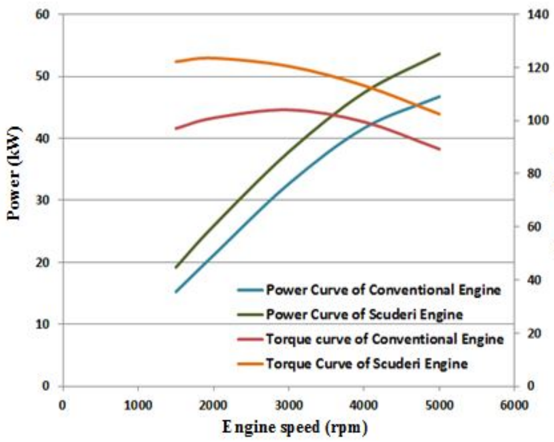

Fig. 13 Power and torque curve for conventional and Scuderi engine

From the Fig. 13, we see that the torque curve of the conventional engine is increasing with the speed up to 3500 rpm and drops gradually. This is because the volumetric efficiency drops down at higher speeds and also the friction losses will be high at higher speed. When we observe the torque curve of the Scuderi engine it is high at the initial stages, remains flat for small range of speed and drops down. This is because at low speed there will be good amount of time for the combustion process and this reduces along with increase in speed. This resembles the characteristics of the diesel engine, having the initial torque higher. From the Figure.8.9, it is clearly visible that the torque and power of the Scuderi engine is higher than that of the conventional engine and it is about 15% higher than the conventional torque and power.

8. CONCLUSIONS

From the functional point of view the Scuderi split cycle engine is having an inbuilt turbo charger. The peak pressure obtained in the Scuderi engine is 11 bar more than the conventional engine at 3000 rpm. The combustion in the Scuderi engine is rapid when compared to the conventional SI engine due to high initial pressure, temperature and turbulence. The turbulence in the Scuderi engine is high compared to the conventional engine which aids in rapid propagation of flame and rapid combustion. There will be difference between the peak pressure developed in the fuel air cycle and the analysis result, because the fuel air cycle cannot take the losses into consideration. In p-V plots the area under the curve of Scuderi engine is more. Hence the work done on the piston is more in this case when compared to the conventional SI engine. The torque of the Scuderi is more at low speed and gradually decrease as it reaches high speed, which is the characteristic nature of diesel engines. The thermal efficiency of the Scuderi engine is 5% more than the conventional SI engine. Finally, from the power and torque curves, we can see that the performance of Scuderi engine is 15-20% more compared to conventional engine.

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