Free-Piston Engine Design and Evaluation with Different Compression Ratios

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Abstract: *Free-piston heat engines are being examined by a number of international research groups as an option for conservative technology, as they are not limited by the movement of the crankshaft as in conventional engines. The free-piston engines are employed in applications like power generation using conventional methods. The higher compression ratios provide higher cycle efficiencies and also boost in-cylinder temperatures increasing mechanical stress, pressures, and heat transfer losses. This paper presents different in-cylinder flows by computational fluid dynamics for various compression ratios ranges from 7 to 15 and estimates were made to find best compression ratio for the optimal engine operating performance and characteristic.*

Key Words: *two stroke SI engine, computational fluid dynamics, Compression Ratios, piston engines, heat transfer losses*

1. INTRODUCTION

High dependence on fossil fuels as an energy resource plays an important role, particularly in relation to land, and sea transport which contributes to the creation of large amounts of CO_2 and other emissions. Most of the research work, particularly in the automobile sector, has been done to develop more environmentally acceptable fuel chains, and the fuel cell motor vehicle stands out as a capable skill for future needs. Most of the research has been carried out mainly in the automotive industry, which produces more environmentally friendly fuel chains, and the fuel cell vehicle stands out as a skill capable of future needs. Even if greater in motor vehicle performance, the execution barriers for such essential skill

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modification are huge, and the overall fuel chain performances are not nevertheless fine to individuals of conservative skill [1]. Hybrid electrical vehicles power-driven through conservative interior combustion heat engines contain the potential to understand bulky discharge reductions inside an extensively shorter timescale. A year ago, and especially in the middle of the 20th century, free-piston heat engines are being processed through a high number of universal research groups as an original to conservative engine and generator setup used for producing hydraulic power in off-highway motor hybrid vehicles.

Likely the reward of the free-piston heat engine which embrace optimized ignition during unpredictable solidity ratio, and leading to superior part of cargo efficiency, clear multiple-fuel process and concentrated frictional losses owed to an effortless layout by means of a small number of movable parts [4].

This paper delivers the history of the free piston heat engine with significance on the latest applications, and studies of forthcoming of free-piston heat engines act as a substitute to conservative skill.

Basics of Free-piston heat engine

Free piston engines are called so because the pistons in the engines move linearly inside their cylinders without any mechanical constraint such as a crankshaft. This simplifies engine design and the engines are potentially more efficient because they do not have frictional and other losses associated with more normal mechanical linkage designed to convert the reciprocating motion into rotary motion.

The piston moves freely because its activity is not governed by the location of a spinning crankshaft, as in conventional heat engines, but rather by the interaction between gas and the load forces acting on it.

This produced the free-piston heat engine with some dissimilar features, which include (a) the requirement for dynamic control of piston action changeable stroke length. (b) Erstwhile significant quality of the free-piston heat engine is potential diminutions in frictional losses and promise toward optimize heat engine procedure by means of the unstable compression ratio.

The original

R. P. Pescara [2] is typically accredited by the innovation of the free-piston heat engine with its reference patent from 1928, Junkers in Germany, was also dealing with free-piston heat engines at that time.

Large numbers of patents have been available describing moving parts with free pistons or associated with such machines.

In addition to the patent, the elite Pescara copyright detailed in a single piston spark ignited air compressor attempts to keep a large number of applications utilizing the free-heat engine piston concept.

In 1932, Pescara continued his discover on free-piston heat engines and produced a model with spark ignition (1936) and diesel ignition (1938).

The most recent led to the progress of the Pescara free-heat engine piston air compressor [3]. In 1941 Pescara [7], [10] invented a very multi-stage free-piston air compressor engine.

Piston configuration

Free-piston heat engines are commonly classified into three types based on the container/ piston planning. A fourth type, free-piston gas generators, categorizes engines where the weight is taken out simply from a fatigue turbine and not from a consignment device which is routinely joined to the heat piston engine. The followings are the explanation of various types of free-heat piston engines.

Single piston

This engine essentially consists of three parts: a recover device to accumulate the energy necessary to clutch the next combustion container, load devices and cylinder charge shown in figure 1 the hydraulic container provided as jointly rebound device and load, while in other designs these might be two entity devices, for in case a gas filled bound chamber and an electric generator. An easy proposes with elevated controllability is the key authority of the single piston plan compared to the other free-piston heat engine configurations [8]. The return device could tender the chance to precisely control the quantity of energy place into the solidity process and thereby adaptable the compression ratio and stroke length.

Dual piston

The design of the free double piston heat engine is shown in the example that was a recent survey of free heat piston engine machines. A number of twin-piston models were designed and small prototypes appeared, along with hydraulic power generation and electric power. The double piston engine preparation omits the persevere for a bounce-back device, as the working piston provides the work to make the solidity process in the other container. This allows a more compact and simple device with a weight ratio to a higher power.
Some issues with the dual piston layout have, however, been reported [9]. The control of piston movement, in exacting stroke length and solidity ratio, has proved difficult. This is owing to the information that the burning procedure in one container drives the solidity in the other, and tiny variations in the incineration will have high pressure on the subsequent solidity. This is a control examination if the incineration procedure is to be inhibited precisely in charge to optimize competence or emissions. Investigational learning with double piston heat engines has reported high consideration to high cycle-to-cycle dissimilarity and load nuances.

Opposed-piston

An opposed-piston free-piston heat engine basically consists of two-single piston units with an ordinary combustion chamber. All pistons need a return tool and a loading mechanism might be attached to one or others of the heat engine pistons [5], [6]. The conflicting piston principle was worn roughly completely in the early on free-piston heat engine designs and perfunctory linkages associated the two pistons to confirm symmetric piston movement, as established in the form.

These engines served efficiently as atmosphere compressors and later on as gas generators in very large-scale plants, regularly with a number of units feeding a solo power turbine. The major benefit of the opposed piston planning is the totally impartial and vibrations free plan.

This aspect is not shared by any of the other free-piston heat engine arrangements which necessitate alternative means of controlling such vibrations. An additional benefit of the opposite piston plan is compact heat transmit losses suitable to the opposite piston container, and this too permits uniflow scavenging to be worn, open-handed high scavenging competence.

The complete requirement for piston bringing together machinery is the mainly noteworthy difficulty of the opposed piston heat engine work. This, jointly with the requirement for a double set of the major components, makes the heat engine complicated and large.

Considers the conflicting piston heat engine design and discards this, and the only latest contrasting piston free-piston heat engine design reported is the hydraulic machine industrial by Ito and Hibi.

2. VARIABLE COMPRESSION RATIO CALCULATION

A spark-ignited engine is frequently alleged to perform similar to an ideal Otto cycle. This is of path far from the truth, but it gives a tool for irregular estimates.

Table 1: Simulation for the engine with varying compression ratio

Bore	Stoke Length	Stoke Volume	Compression ratio(r)	Clearance Volume	Clearance Length
m	mm	mm^3		mm^3	mm
35	37	35598.26	7	5933.04	6.17
35	37	35598.26	9	4449.78	4.63
35	37	35598.26	11	3559.83	3.70
35	37	35598.26	13	2966.52	3.08
35	37	35598.26	15	2542.73	2.64

This means that if the solidity is high then the efficiency is high. The highest competence is thus achieved at countless compression ratios. In an actual engine cycle, energy is gone to heat transfer. The rate of heat transfer increases with compression, and consequently the highest efficiency is originating at an incomplete compression ratio.

$$\text{Compression ratio } (r) = (Vs + Vc)/Vc \text{ also } Vc = Vs/(r+1)$$

where:

Vs = Swept volume for one cylinder
= 3.1416 x (Radius of bore)2 x stroke length

Vc = Clearance volume for one cylinder
= 3.1416× (Radius of bore)2 x Clearance Length Also, Clearance length = Vc/Vs

In the current work, simulation was performed to visualize the combustion performance, temperature rise, pressure rise within the engine all through ignition with the compression ratio of 9. Forth coming/Forthcoming work is to run CFD, Simulation for the engine with varying compression ratios as shown in Table 1, comparison of various performances of the engine.

Model Construction

Three-dimensional representation of the two-stroke free piston engine with elliptical shaped exhaust as shown in figure 1 was initial created in Solid Works 2012 and exported as IGS files. The IGS or IGES file format is a third party file format most probably supported in all commercially available CFD tool. The IGS files were then imported into ANSA, the mesh generator. In ANSA, the imported geometry is checked for topology. Then the flow domain is extracted from the imported model.

Figure 1: Model of the Engine Cylinder with elliptical shaped exhaust port

Meshing

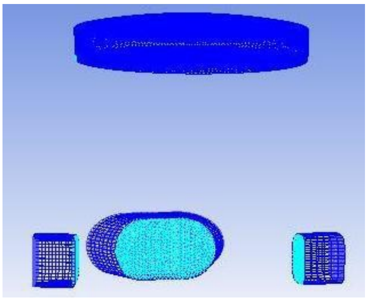

Figure 2: Volume mesh of static region of the engine extracted fluid domain

Table 2: Meshing details of the engine Surface mesh details

Element Type	Triangular [TRI]
Maximum Quality	0.6
Surface mesh count	13613
Volume mesh details	
Element Type	Tetrahedral [TET] &
Prism	
Maximum Quality	0.87
Volume mesh count	322500

Solver Set Up and Methodology

In this case, ANSYS-FLUENT is used as the resolver. Temporary choice is activated.

Table 3: Parameter Dimension

Volume (cc)	40
Stroke (mm)	37
Bore (mm)	35
Connecting rod(rigid) length (mm)	34
Admission ports	2
Exhaust ports	1
Exhaust port opening/ closing	27.9 mm from TDC/ 9.1 mm from BDC
Scavenged port opening/ closing	28.9 mm from TDC/8.1 mm from BDC
Compression ratio (r)	7:1

TDC-Top Death Centre BDC-Bottom death Centre

- In-cylinder gas is assumed to be 3-D, turbulent, incompressible in nature.
- Energy equation is activated to solve thermal distribution.
- Species transport with volumetric reaction is enabled to model combustion inside the engine.
- Flash explosion is enabled to start on aflame when the piston reaches TDC
- Turbulence is formed according to K-ε model
- SIMPLE (Semi Implicit Pressure Linked Equations) algorithm is used to resolve the difficulty.
- Segregated solver is worn for pressure-velocity coupling.
- Active meshing method is worn to net the quantity of container using fluent dynamic mesh choice.
- Layering methodology is worn to mould the movement of the piston.

Boundary Conditions

The initial conditions for this case are:

- Linear piston velocity is assumed to be 5.59 m/s. The Boundary conditions for this case are:
- Inlet is assumed to be the velocity inlet with velocity of 4m/s,
- Air-fuel ratio octane fuel is assumed to be 12:1 (the mass fraction of octane is 0.0769230 and the mass fraction of air is 0.923076) at both the transfer port 1 and transfer port 2,
- Temperature of the inlet at the transfer port is given as 303K,
- Interface boundary condition is given and interfacing is done between the sliding interface zone and the gas exchanging ports,
- Wall boundary condition is given to remaining part of the engine,
- Outlet is presumed to be the pressure outlet with 0 Pascal (static pressure).

3. RESULTS AND DISCUSSIONS

Numerical simulations for the modelled two-stroke free-piston engine were carried out with keeping the compression ratio as 7:1 to evaluate the combustion performance, pressure rise and temperature rise.

CFD-fluent results and discussion

The result of the combustion simulation of the two-stroke free-piston engine with compression ratio 7:1 is obtained with the air fuel mixture velocity/speed set to 4m/s at the transfer port inlet and the linear piston velocity/speed to 5.59 m/s.

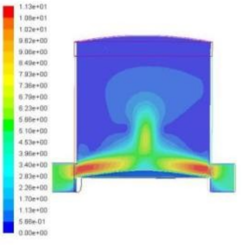

Figure 3: Contour of velocity (m/s) of air - octane mixture at mid plane of the engine with a piston 26.10 mm from TDC

The inlet velocity which is given as 4m/s at the inlet is reached to the maximum value of 12 m/s when the transfer port is opened with the movement of piston towards Bottom death Centre. The maximum velocity at the exhaust has reached up to 28 m/s of burned gas with exhaust port half opened condition. The velocity contour at the air-octane mixture getting into the cylinder is shown in figure 3.

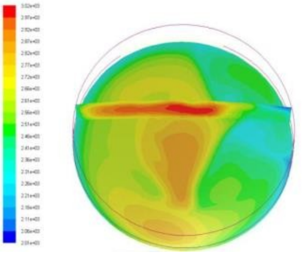

Figure 4: The Contour of total temperature during combustion of the air-octane mixture when the piston is at TDC

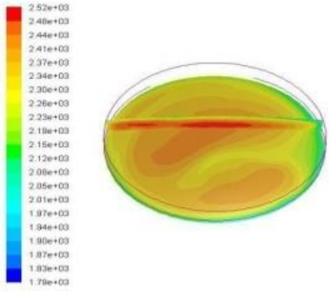

Figure 5: The Contour of total temperature during combustion of the air-octane mixture when the piston is at 0.8 mm TDC

The peak temperature value obtained during combustion when the piston is at TDC is around 3000K (Figure 4 and 5) which roughly equates to the actual combustion of octane fuel.

There is a notable downturn in temperature inside the cylinder to around 1200K when the exhaust port is opened with the piston motion towards BCD.

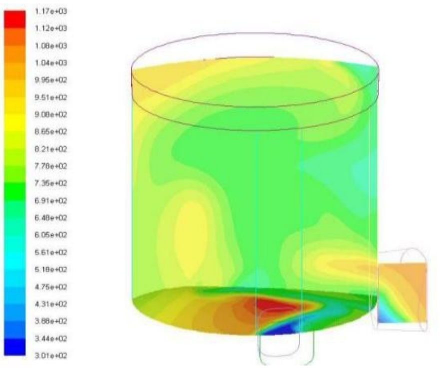

Figure 6: The Contour of total temperature at mid plane inside the cylinder, when the piston is at 33 mm from TDC

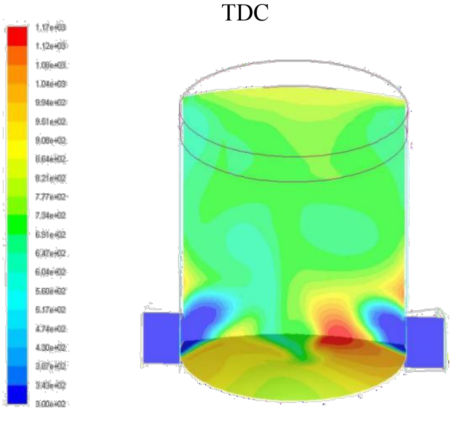

Figure 7: The Contour of total temperature at mid plane inside the cylinder when the piston is at 33 mm from TDC (At Exhaust)

Figure 7 illustrates the hotness sharing inside the container at the ending of the stroke with the warm gases separation of the exhaust port.

High temperature flanked by 800K and 900K was still seen inside the container near TDC even at the ending of the expansion stroke.

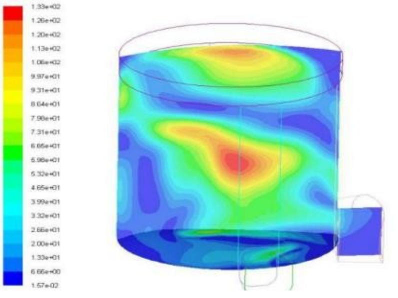

Figure 8: The Contour of total pressure at mid plane inside the cylinder when the piston is at 33 mm from TDC

The high pressure of about 12.5 bar is achieved during combustion inside the modeled engine when the piston is at TDC. The difference of pressure throughout solidity stroke and turn over combustion and the incineration reaction's peak pressure of the air-fuel mixture with respect to piston arrangement is illustrated in fig. 9.

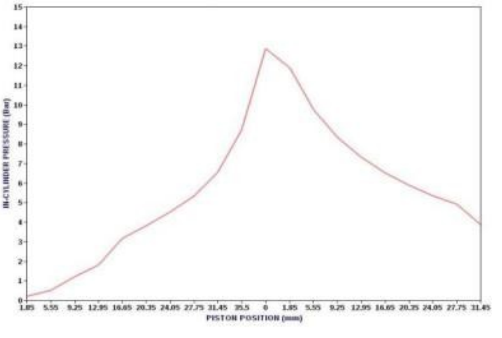

Figure 9: In-cylinder pressure vs piston position

At present, the manner in which the combustion occurs in small container, at high speed engines are largely unknown. Nevertheless, combustion has been originated to be the major drawback in extending the functioning restrictions for economized applications. More than a few important differences may be predictable as a result of the smaller capacity and increased engine speed when compared to superior bore engines. These contain: enlarge in-wall belongings on fall over and whirl rakishness, decrease in the turbulent length scale and increased heat losses from the enlarged surface to volume ratio enlarged fire extinguish area consequential from lesser combustion chamber authorization distances. Therefore, combustion analysis was finished to measure ignition effects for the compression ratio of 7 with a constant piston velocity of 5.5m/s.

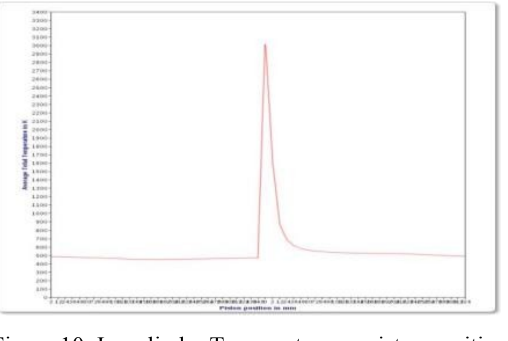

Figure 10: In-cylinder Temperature vs piston position

Fig. 10 illustrates that there is a maximum peak in hotness in the in-cylinder section at the peak quiet center position of the container piston movement and also due to superior flash timing.

The burning procedure in the free-piston engine has been established and also the main drawback, below high peak temperature in extending the operating limits for downsized applications. Due to the smaller capacity and increased engine speed when compared to larger bore engines, the temperature distribution rate and frequent process of combustion take place. These include the heat distribution on the walls of the cylinder, the heat flux generated during the combustion operation, etc. Hence, a combustion analysis for temperature distribution was also completed to quantify the combustion effects for the compression ratio of 8 with a constant piston velocity of 5.5m/s.

4. CONCLUSIONS

Most of the fuel in the free piston engine burns in the pre-mixed phase, resulting in a very high rate of heat discharge with a high force slope, according to the computational burning examination of two-stroke free-piston engine. An improper scavenging process takes place inside the engine which might be the cause of the present low compression ratio configuration of the engine. Additional investigations are necessary to improve the general performance which includes the variation of compression ratio and its effect on engine speed (piston velocity), combustion, heat release rate, maximum temperature rise, maximum pressure rise and scavenging performance.

FUTURE SCOPE

- The combustion chamber can be planned in all axes as a Rankine profile by consistently changing the sizes.
- The number of pintle hole changes in the changed profile combustion chamber.
- The effect of chamber profile variability in engine performance parameters and vibrations can be studied with a variety of fuels.
- EGR with turbochargers can be used with the above combination to test engine performance and vibration.
- The influential factor / parameters can be identified using the Artificial Neural Network, Genetic Algorithms and Surface Response Method.

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