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Design of Four Stroke Engine Simulation Model to Optimize Volumetric Efficiency at Variable Stroke Length and Compression Ratio

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ABSTRACT

Modelling and simulation tools are very essential supports in today's technology that reduce the physical prototypes in the field of developing multidomain systems. Virtual prototyping also decreases the product development time. This paper presents a virtual prototype model of a four stroke I.C. engine in which an incorporated mechanism will vary the stroke length during suction stroke and allow more air to be drawn into the cylinder instead of compressing the intake air to increase the air density. It has a larger stroke length during the suction and compression strokes. The stroke length during the power and exhaust strokes is identical to corresponding conventional engine. As a result, larger volume of air is taken in during suction and is compressed. The volumetric efficiency thus increases. It has been seen that a maximum of 2 cm extra stroke length may be obtained during suction and compression which yields a 12.5% increase in volume for the proposed design of the model. Also, from dynamic simulation curve, a maximum of 34% increase is observed. There is also an increase in the thermal efficiency due to increased compression ratio. The rest of the strokes are performed in the same way as in a conventional engine. This increases the engine efficiency. This concept introduces an economical advantage over the turbocharger and supercharger.

NOMENCLATURE

D	Cylinder Bore (mm)
r_1	Base circle radius of cam (mm)
L	Cylinder Length (mm)
r_2	Nose radius of cam (mm)
d	Piston diameter (mm)
α	Angle of ascent of cam (degree^0)
l	Piston length (mm)
Φ	Angle of contact on circular cam (degree^0)
D_c	Crank shaft diameter (mm)
δ	Follower Lift (mm)
R	Crank radius (mm)
ds	Spring wire diameter (mm)
B	Distance between main bearings (mm)
D_s	Mean coil diameter of spring (mm)
L_1	Connecting rod length (mm)
N	No. of active spring coils
L_2	Piston rod length (mm)
N_t	Total no. of spring coils
L_3, L_4	Control lever lengths (mm)
L_5	Free length of spring (mm)
p	Pitch of spring (mm)
D_1	Crank shaft pulley diameter (mm)
k	Stiffness of spring (N/mm)
D_2	Cam shaft pulley diameter (mm)
L_6	Belt length (mm)
x	Centre distance, Belt (mm)
ω	Angular velocity (rad/s)

Keywords

Variable Stroke Length, Volumetric Efficiency, Variable Compression Ratio, Simulation, CREO.

INTRODUCTION

The sole objective of the project lies in increasing the volumetric efficiency of the engine by having a variable stroke length of the engine by introducing a cam-controlled lever mechanism. With the evolution of IC engine an effort has always been made to increase the efficiency of the Engine. Increasing the volumetric efficiency of the engine by proper combustion of the fuel is one of the successful methods discovered.

For the first time in 1878, supercharger was invented by Dugald Clerk [1, 2] in order to increase the volumetric efficiency by increasing density of the intake which is driven by shafting it with the output shaft. Due to a major disadvantage of supercharger, as it consumes a part of the output power, turbocharger has been developed with the same principle to increase volumetric efficiency with a difference that it is powered by a turbine driven by the engine's exhaust. However, turbocharger also has certain disadvantages. Due to increase in the temperature of the intake air, it may cause detonation especially in petrol engines. Also, turbocharger is very expensive.

So, looking at these disadvantages our aim is to develop an engine such that the same task can be done with comparative low cost in an effective manner. D. Osorio, Julian et al. [3] have proposed a Continuous Variable Valve Timing (CVVT) system for load control in spark-ignition engines, analyzed and compared with a conventional Throttle-controlled Engine. A fuel economy increment of up to 4.1% is observed from the analysis, for a CVVT Engine with reference to a Throttled Engine at a 20% – 30% load that is typical of a real vehicle engine operation. Satyanarayana K. et al. [4] predicted the design performance based on stress-strain behavior for three different variable compression ratios of a diesel engine. Pournazeri Mohammad et al. [5] proposed and designed hydraulic variable valve actuation system which significantly improves the engine performance in terms of volumetric efficiency, fuel consumption etc. Wakode Vaibhav R. et al. [6] investigated a modification on single cylinder diesel engine for three different values of compression ratios and fuel injection pressures. They found that fuel injection pressure at 220 bar and compression ratio of 18 yields optimum engine performance and emissions. Ganji; Rao Prabhakara et al. [7] analyzed numerically using CONVERGE™ Computational Fluid Dynamics code to optimize compression ratio and other related parameters at 100% engine load to test its performance and was found satisfactory. Radivoje B. Pesic et al. [8] did both theoretical and experimental investigation of the impact of automatic variable compression ratios on working process parameters in experimental diesel engine. They illustrated and critically examined alternative methods of implementation of variable compression ratio. Doric, Jovan Z., et al. [9] presented a simulation of variable movement of piston for obtaining variable compression ratio, variable displacement and combustion during constant volume. Yamin, Jehad A. A. et al. [10] presented a theoretical study on the effect of variable stroke length technique on the emissions of a water-cooled four-stroke, direct injections diesel engine with the help of experimentally verified computer software designed mainly for diesel engines. The emission levels were studied over the speed range in between 1000 rpm and 3000 rpm and stroke lengths in between 120 mm and 200 mm, which were compared with those of the original engine design. Yamin, Jehad A.A. et al. [11] developed a simulation model and verified with experimental results from the literature for both constant and variable-stroke engines which shows the advantages of utility of variable stroke engines in fuel economy issues. G, Abhishek Reddy et al. [12] did an experimental investigation of the influence of compression ratios 14, 15, 16 and 18 and engine loads of 3kg to 12 kg, with increments of 3kg, utilized for diesel on the brake power, brake thermal efficiency, brake mean effective pressure and specific fuel consumption for the Kirloskar variable compression ratio dual fuel engine. Ebrahimi, Rahim [13] in his experiment showed that output power increases with increase in stroke length if compression ratio is less than certain value, on other hand the power output first increases and then start decreases with increase in stroke length if compression ratio exceeds certain value. The output power decreases with increase in stroke length for further increase in compression ratio. Asthana, S. et al. [14] reviewed the work of technological advancements in the design of a VCR engine and describes the various techniques by which VCR is being implemented and provides a qualified study for original and modified technology on the basis of engine rigidity and piston kinematics, engine friction, specific fuel consumption, and efficiency. Moteki, K. et al. [15] presented a variable compression ratio (VCR) system that has a new multiple link in piston-crankshaft mechanism. The multi-link mechanism varies the piston position at top dead center (TDC), making it possible to vary the compression ratio of the engine continuously. Jiang, S. et al. [16] focused on a multiple-link mechanism that realizes variable compression ratio and displacement with varying piston motion, specifically the Top Dead Center (TDC) and Bottom Dead Center (BDC) positions relative to the crankshaft. They used Design of Experiments (DoE) methodology for creating sets of geometric designs of the mechanism, in which kinematics are calculated and checked against the conditions.

1. METHODOLOGY

The joint between the piston rod and the connecting rod has the control lever attached. The other end of the lever is connected to the follower through a hinge. The lever gets its motion from the follower. And the motion of the follower in a contour is controlled by a rotating cam with a spring arrangement. The speed of camshaft is half the speed of the crankshaft. The camshaft gets the power from the crankshaft itself by means of a belt drive (for the wooden model). Mechanism for the variable stroke length engine, which was drafted in CREO, is shown in figure 1. The various parameters of the model were designed and calculated as below:

1.1 Piston and Cylinder [17]

For the model, the bore of the engine cylinder is assumed to be equal to D = 100mm and the Length of the cylinder, L = 150mm. The (L/D) ratio of the cylinder is usually in the range 1.15 to 1.5. A larger range is taken considering variable stroke conditions. Assuming clearance, C = 2.5mm on each side we calculate,

Piston diameter, $$d = D - 2C \tag{1}$$
$$= 95\text{mm}.$$

Piston length, $$l = 0.5d \text{ to } 1.2d \tag{2}$$
$$= 50\text{mm}.$$

The crankshaft diameter is chosen as, D_c = 50 mm, crank radius, r = 50mm. The distance between the main bearings,

$$B = 2d \text{ to } 3d \tag{3}$$
$$= 285 \text{ mm}$$

The connecting rod length is an important parametric consideration. In our design the length of the connecting rod is kept short because we also use a “piston rod” and a “control lever”. This arrangement is to control the position of the small end of the connecting rod required for the variation of stroke. The connecting rod has greater angular swing when it is short as compared to the crank radius. The angular swing is suitably controlled by the control lever itself.

Connecting rod length,

$$L_1 = 1.6\, r \tag{4}$$
$$= 80 \text{ mm}.$$

The design of the piston rod is made with the care to minimize the side thrust on the piston. This is done by having a suitable $\left(\frac{L_2}{r}\right)$ ratio. For moderate speed engines the ratio varies from 4 to 5 (V. B. Bhandari, 2017).

Piston rod length,

$$L_2 = 4r \text{ to } 5r \tag{5}$$

Or $$L_2 = 4.4\, r \tag{6}$$

$= 220$mm.

Angle of ascent, $\alpha = 80°$

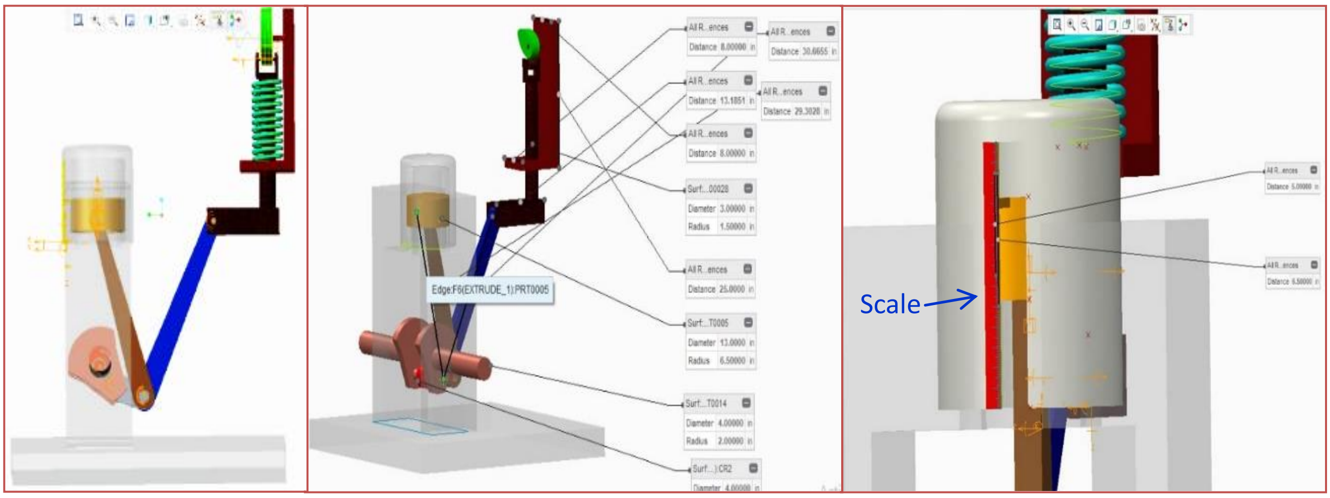

Fig. 1: Model of Variable Stroke Length Engine Mechanism

Length of Control Lever:
The control lever is basically a rocker arm which applies a moment to control the position of the small end of the connecting rod about a hinge.

The ratio of the lengths of the lever on either side of the hinge is taken as

$$\frac{L_3}{L_4} = 3. \qquad (7)$$

Assuming, $L_4 = 50$ mm,

$L_3 = 3 \times L_4$

$= 150$mm.

1.2 Cam profile [18]

The geometry is shown in Figure 2.

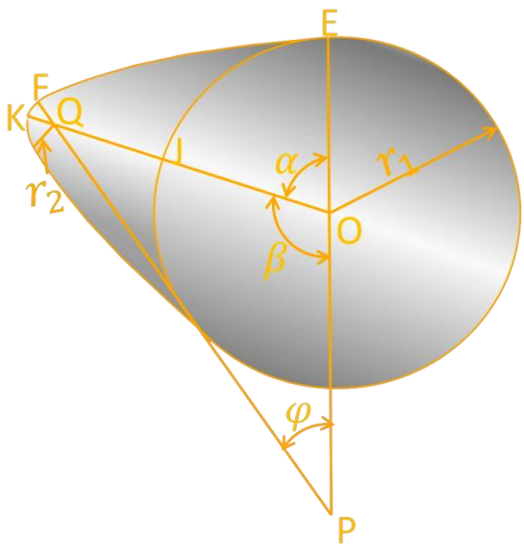

Fig. 2: Cam profile geometry

We assume the following:

Maximum lift of the follower is taken as 40mm.
Base circle radius, r_1 = 50mm;
Nose circle radius, r_2 = 20mm;

The distance between the cam center and nose center is calculated as,

$$OQ = \text{Follower lift} - (r_2 - r_1) \qquad (8)$$

$= 70$ mm

We know, $PQ = PF - FQ = PE - FQ = OP + OE - FQ$

$= OP + 50 - 20 = OP + 30$

From triangle OPQ,

$$PQ2 = OP2 + OQ2 - 2 \times OP \times OQ \cos\beta \qquad (9)$$

$$(OP + 30)2 = OP2 + 702 - 2 \times OP \times 70 \times \cos(180° – 80°) \qquad (10)$$

Solving we get, $OP = 112$mm.
Therefore, Radius of circular flanks is given by

$R = PE = OP + OE$

$= 162$ mm.

And, $PQ = OP + 30$

$= 142$mm.

From triangle OPQ,
The angle of contact on the circular flank, $\emptyset$ is given by the equation,

$$\frac{OQ}{\sin\emptyset} = \frac{PQ}{\sin\beta} \qquad (11)$$

Or, $\emptyset = 29°$

1.3 Belt and Pulley (Wooden Model) [18]

The power is transmitted from the crankshaft to the camshaft by means of a belt drive. Pulleys are mounted on the shafts. One pulley is mounted on the camshaft of the model.
Speed of the crankshaft = N_1
Speed of the camshaft = N_2
Camshaft pulley diameter = D_2
A groove on the crankshaft serves as a pulley itself. So, Crankshaft pulley diameter, D_1=50mm.
The speed of the camshaft is half of the speed of the crankshaft. Therefore,

$$\frac{N_1}{N_2} = \frac{2}{1}$$

Now, Velocity ratio is given as,

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$$\frac{N_1}{N_2} = \frac{D_2}{D_1} \quad (12)$$

Or, $D_2 = 100$ mm.

The center distance between the shafts is found to be 43cm. Since the center distance is small and is less than, so we choose a V-Belt Drive.

Now, the Length of the belt is calculated as,

$$L_6 = \frac{\pi}{2} \times (D_1 + D_2) + 2x + \frac{(D_1 - D_2)^2}{4x} \quad (13)$$

$$= \frac{\pi}{2} \times (50 + 100) + 2 \times 430 + \frac{(100-50)^2}{4 \times 430}$$

$= 110$ cm

Therefore, considering the calculations, a standard V-belt is chosen which has length 114cm.

1.4 Spring (For Animated model) [19]

The follower lift is 40 mm. So accordingly, the required deflection of the spring should be 40mm; i.e., δ = 40 mm. The spring material is patented and cold drawn steel wire of Grade -1. For the spring material, G = 43350 N/mm^2, spring index: C = 10, maximum force: P = 50 N, ultimate stress: S_{ut} = 324 N/mm^2 , shear stress: τ = 0.5 × 36.5 = 162 N/ mm^2.

Wahl's factor is calculated as,

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} \quad (14)$$

$$= \frac{4 \times 6 - 1}{4 \times 6 - 4} + \frac{0.615}{6}$$

$= 1.145$

Shear stress is given by,

$$\tau = K\left(\frac{8PC}{\pi d^2}\right) \quad (15)$$

Or, $162 = 1.145\left(\frac{8 \times 50 \times 10}{\pi d^2}\right)$

Or, $\cong 3$ mm

Mean coil diameter is calculated as,

$$D = C \times d \quad (16)$$

$= 30$ mm

Number of Active coils is calculated as,

$$\delta = \left(\frac{8PND^3}{GD^4}\right) \quad (17)$$

Or, $40 = \left(\frac{8 \times 50 \times N \times 30^3}{43350 \times 3^4}\right)$

Or, $N = 13$.

Assuming, the spring has square and ground ends. Number of inactive coils is 2. Therefore, Total number of coils is given by,

$$N_t = N + 2 \quad (18)$$

$= 15$ coils.

Now, the actual deflection of the spring is given by,

$$\delta = \left(\frac{8PND^3}{GD^4}\right)$$

Or, $\delta = 39.98$ mm.

Solid length of the spring is calculated as,

$$l_s = N_t \times d \quad (19)$$

$= 45$ mm.

Axial gap between adjacent coils is found as 0.2mm and Solid length as 47mm. Therefore,

Total Axial gap $= (N_t - 1) \times$ axial gap between adjacent coils (20)

$= 2.8$ mm.

Free length of the spring is calculated as,

$$L_5 = \text{solid length} + \text{total axial gap} + \delta \quad (21)$$

$\cong 90$ mm.

Pitch of the spring coil is calculated as,

$$p = \frac{Free\ Length}{N_t - 1} \quad (22)$$

$= 6.4$ mm.

The spring stiffness is calculated as,

$$k = \left(\frac{Gd^4}{8ND^3}\right) \quad (23)$$

$= 1250.5$ N/m.

Table 1 shown below has listed various parts of the model, their parameters with symbol and dimensions.

Table 1 Dimensions of various parameters of the parts of model.

Sl. No.	Parts	Parameters	Symbols	Dimensions
1	Cylinder	Bore	D	100mm
		Length	L	150mm
2	Piston	Diameter	d	95mm
		Length	l	50mm
3	Crankshaft	Diameter	D_c	50mm
		Crank radius	r	50mm
		Distance between main radius	B	285mm
4	Connecting Rod	Length	L_1	80mm
5	Piston Rod	Length	L_2	220mm
6	Control Lever	Lengths	L_3	150mm
			L_4	50mm
7	Cam	Base radius	r_1	50mm
		Nose radius	r_2	20mm
		Angle of ascent	α	80^0
		Angle of contact on circular cam	Φ	29^0
		Follower lift	δ	39.98mm
8	Spring	Wire diameter	d_s	03mm
		Mean coil diameter	D_s	30mm
		No. of active coils	N	13nos.
		Total no. of coils	N_t	15nos.
		Free length of spring	L_5	90mm
		Pitch	p	6.4mm
		Stiffness	k	1250.5N/mm
9	Pulley	Cam shaft pulley diameter	D_2	100mm
		Crank shaft pulley diameter	D_1	50mm
10	Belt	Length	L_6	110cm
		Centre distance	x	43cm

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The wooden working model mechanism which adds a 2cm length in suction and compression stroke is shown in figure 3.

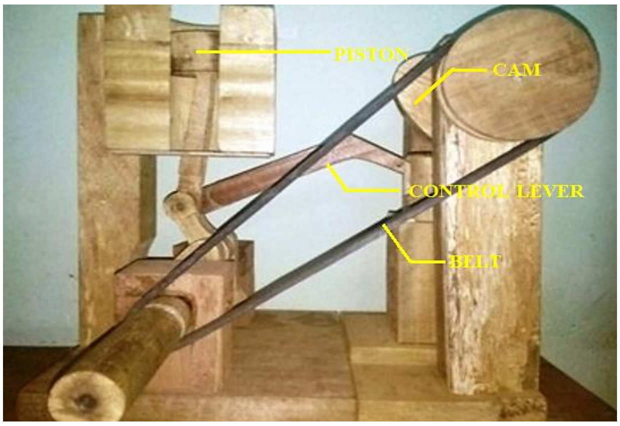
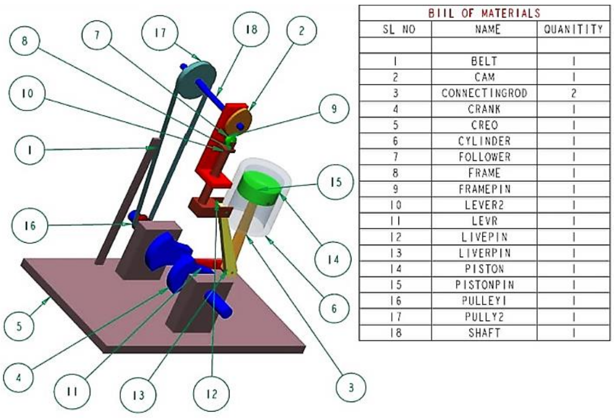


Fig. 3: Wooden model of the mechanism

CREO prepared model with different parts name has been shown in Figure 4.



BIIL OF MATERIALS		
SL NO	NAME	QUANITITY
1	BELT	1
2	CAM	1
3	CONNECTINGROD	2
4	CRANK	1
5	CREO	1
6	CYLINDER	1
7	FOLLOWER	1
8	FRAME	1
9	FRAMEPIN	1
10	LEVER2	1
11	LEVR	1
12	LIVEPIN	1
13	LIVERPIN	1
14	PISTON	1
15	PISTONPIN	1
16	PULLEY1	1
17	PULLY2	1
18	SHAFT	1

Fig. 4: Prototype model with Bill of Materials

Two animated views showing two different (opposite) positions of cam are shown in figure 5.

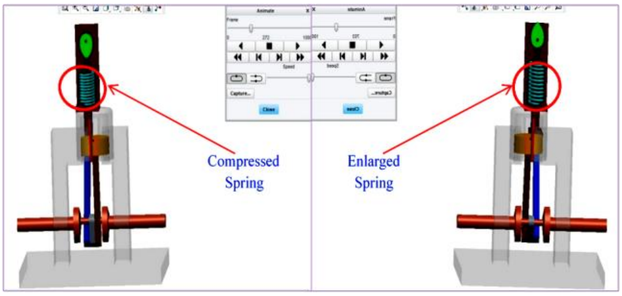


Fig. 5: Two different positions of cam in animated model.

2. TEST AND RESULTS

After fabrication of the model, a scale is fitted to the cylinder. The cylinder is made to have a slit which allows the visual of the piston's movement inside the cylinder. The scale is then used to measure the variation of the strokes during the 4 strokes of the engine cycle. The stroke length during suction, compression, power and exhaust strokes is measured by means of the scale. The difference in stroke length during the first two strokes and the last two strokes of the engine cycle gives the stroke variation of the model as listed in Table 2.

Table 2: Difference in stroke lengths obtained during the test

Sl. No.	*Stroke*	*Stroke Length (cm)*	*Character*
1	Suction	18	L_1
2	Compression	18	L_2
3	Power	16	L_3
4	Exhaust	16	L_4

Average stroke lengths during suction and compression,

$$L_a = \frac{L_1 + L_2}{2} \tag{24}$$
$$= 18 \text{ cm}$$

Average stroke lengths during power and exhaust,

$$L_b = \frac{L_3 + L_4}{2} \tag{25}$$
$$= 16 \text{ cm}$$

Therefore, variation in stroke length during suction and compression, $L_b - L_a$ = 18cm - 16cm = 2cm. The percentage increase in volumetric efficiency depends on the varying suction stroke length, which is given by Percentage increase [20],

$$\eta = \frac{L_v - L}{L} \times 100\%. \tag{26}$$

Here, L_v is the value of suction stroke length after variation and L is the length of the conventional stroke. So, Increase in Volume = $\frac{18-16}{16}$ = 0.125 = 12.5 % and hence increase in volumetric efficiency accordingly.

A dynamic analysis is done on the movement of the piston inside the cylinder. A graph is plotted between the stroke length and time period during different strokes for a number of cycles. The graph shows that the stroke length during suction and compression is larger than the power and exhaust strokes. Then the cycle repeats. Hence the results fulfill our objective. Figure 6 shows the analysis graph of the simulating model prepared in Creo [21].

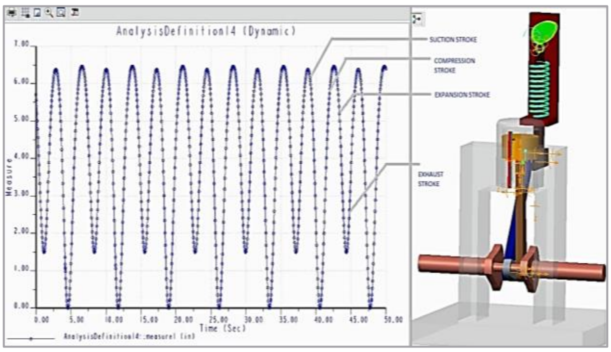


Fig. 6: Dynamic analysis of the simulating model

The Table 3 shown below reflects the value of percentage increase in efficiency for the different value of suction stroke (the values are taken from the Creo model analysis graph).

Normal length of suction stroke (L) in cm	Variable values of length of suction stroke (L_V) in cm	% increase in volumetric efficiency ($\%\eta_v$)
5	6.5	30 %
5	6.4	28%
5	6.7	34%
5	6.3	26%
5	6.6	32%

Table 3: Variation of suction length and percentage increase in volumetric efficiency against normal suction length

Figure 7 shows a graph indicating percentage increase in volumetric efficiency at variable stroke length

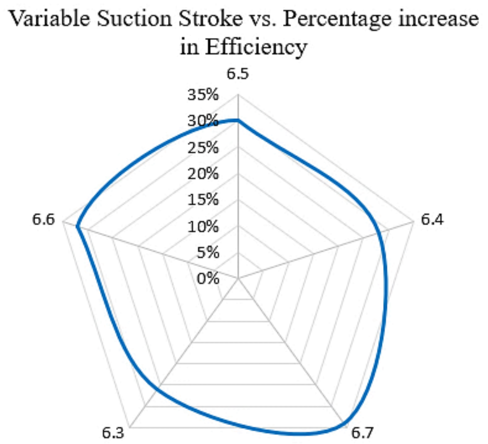


Fig. 7: Plot of percentage increase in volumetric efficiency with respect to variable suction stroke

The time domain and frequency domain analysis of the results listed in Table 3 is shown in Figure 8 plotted in MATLAB. (Version 11)

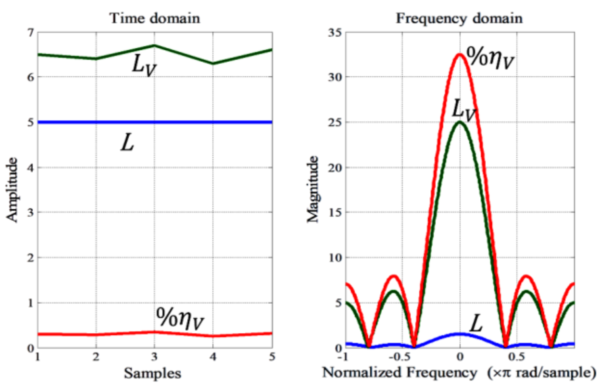


Fig. 8: Time domain and Frequency domain of results listed in Table 3.

3. CONCLUSION

This model, which is based on a simple cam operated lever mechanism, using a spring will replace turbocharger and supercharger, and will carry out the similar task as they do. The model is characterized by first, volumetric efficiency is increased by 12.5% and may be more for a better replica. Secondly, of low cost as compared to a turbocharger / supercharger installed engine. Thirdly, Intake air pressure is not increased as in a turbocharger or supercharger. So, the issue of detonation in petrol engine is avoided.

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