A Study of a Variable Compression Ratio and Displacement Mechanism Using Design of Experiments Methodology

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Abstract

Due to the ever increasing requirements for engine performance, variable compression ratio and displacement are drawing great interest, since these features provide further degrees of freedom to optimize engine performance for various operating conditions. Different types of mechanisms are used to realize variable compression ratio and displacement. These mechanisms usually involve relatively complicated mechanical design compared with conventional engines. While the flexibility in these mechanisms introduces additional engine design and control possibilities, it also increases the challenges for the development and optimization, since how the geometry of the mechanisms affects the engine performance is not straightforward, and building prototype engines of various mechanical design parameters takes a long time with high cost.

This paper presents a study of a multiple-link mechanism that realizes variable compression ratio and displacement by varying the piston motion, specifically the Top Dead Center (TDC) and Bottom Dead Center (BDC) positions relative to the crankshaft. The study focuses on obtaining desirable behavior for both compression ratio and displacement. It is determined a major requirement for the design of the mechanism is that when the control action changes monotonically over its whole range, the compression ratio and displacement should change in opposite directions monotonically. This paper gives an approach on how to achieve multiple-link mechanism designs that fulfill this requirement. First, a condition is obtained on how the TDC and BDC positions should change with respect to the control action to fulfill the design requirement. Then Design of Experiments (DoE) methodology is used for creating sets of geometric designs of the mechanism. Kinematics of the mechanism are calculated for each design and checked against the condition. Based on the outcome, a feasible design is selected and detailed study on such characteristics as piston motion, stroke length, combustion chamber volume, displacement, crank angle range of the expansion stroke, and compression ratio is performed. Directions for future work in simulation and testing of the mechanism are given.

1. Introduction

Ever increasing requirements for engine performance and stricter exhaust emissions and fuel economy standards have been driving engine technology advancement. Variable compression ratio technology has long been recognized as a method to improve engine fuel economy [1]-[3]. As the compression ratio increases, the engine thermal efficiency increases. However, too high compression ratio will lead to engine knock. With variable compression ratio, at low power levels the engine operates at a higher compression ratio to capture the benefits of high fuel efficiency; at high power levels the engine operates at a lower compression ratio to prevent knock. By continuously changing the compression ratio, an engine enables optimum combustion efficiency at all engine speed and load conditions, realizing better engine performance, lower fuel consumption, and lower exhaust emissions.

There are various approaches to realize variable compression ratio, such as moving the cylinder head, moving the crankshaft axis, and using multiple links for connection between piston and crankshaft. While the main purpose of these mechanisms is to realize variable compression ratio, they might also lead to changes in many other aspects of the engine, such as piston motion, stroke length, charging and discharging efficiency, and combustion processes. These characteristics need to be considered when designing the variable compression ratio mechanism and determining its detailed dimension values. Optimization of related engine control parameters need be performed accordingly as well.

This paper focuses on studying a multiple-link connecting rod mechanism [4]-[6]. This mechanism realizes variable compression ratio by varying the piston motion, specifically the TDC and BDC positions of the engine. So far major studies of the multiple-link mechanism have focused on the variable compression ratio characteristics. However, a good design of the geometry of the mechanism should realize not only the desired compression ratio, but also advantageous displacement behavior of the engine. Since higher compression ratio is used for low and medium power levels, where decreased displacement is advantageous, the desired design of the multiple-link mechanism is that when the control action changes monotonically, the compression ratio and the displacement change monotonically in opposite directions.

This paper presents an approach to obtain geometric designs of the mechanism that fulfill the requirement mentioned above. First a condition is obtained in the format of how the TDC and BDC positions should change with respect to the control action in order that the mechanism will give the desired behavior. Equations describing the kinematics of the components in the mechanism are derived. The piston position can be calculated during the whole revolution of the crankshaft for specific mechanism dimensions and control action. To obtain geometric designs of the mechanism that satisfy the condition, Design of Experiments (DoE) methodology is used for creating sets of geometric parameters, including crank radius, crank offset, upper link length, lower link dimensions, control link length, control link position, and control shaft length. Piston motion corresponding to each parameter set is calculated and the behavior is checked against the condition. A feasible design that satisfies the condition is selected for further study. Such characteristics as detailed piston motion profile, piston velocity and acceleration, crank angle range of the expansion stroke, combustion chamber volume, displacement, and compression ratio are investigated for the design. These results serve as a basis for further analysis and optimization of the mechanism geometric design.

The paper is organized as follows: the multiple-link mechanism is described and its equations of motion are derived in Section 2. Section 3 presents the condition on how the TDC and BDC positions should change with respect to the control action to fulfill the design requirement for achieving desired compression ratio and displacement behavior. In Section 4, a Design of Experiments (DoE) plan is established for the mechanism dimensions, and the kinematics of the mechanism are calculated for each design and checked against the condition. Further detailed study of a design that satisfies the condition is presented in Section 5. Section 6 provides directions for future studies of the mechanism using simulation software and testing. The last section summarizes the paper.

2. Multiple-link Mechanism

Configuration of the multiple-link mechanism is shown in Figure 1. Origin (O) is the crankshaft axis. The crank pin (A) and piston pin (E) are connected by means of an upper link (DE) and a lower link (ABD). The lower link is also connected, via a control link (BC), to the eccentric journal of a control shaft (SC). When the crankshaft rotates, the lower link revolves around the crankshaft axis and pivots around the crank pin. Meanwhile, motion of the lower link is constrained by the control link. Note the crankshaft (OA), the lower link (AB), the control link (BC), and segment (OC) which can be considered a ground link, form a four-bar linkage. As a result, when the crankshaft rotates, the control link acts as a rocker and rotates through a limited range of angles around the eccentric journal of the control shaft. The connection point (D) of the lower and upper links traces roughly an oval shape and the piston pin (E) undergoes reciprocal motion.

The control shaft (SC) is supported by the engine block. The distance between the center of the eccentric journal and the control shaft axis is denoted as l_s, and will be called control shaft length in this study. When the angle of the control shaft relative to the engine block (θ_s) is changed, the position of the control link will change, which subsequently tilts the lower link clockwise or counterclockwise and thereby affects the motion of the piston and shifts the TDC and BDC positions. This leads to change in both the combustion chamber volume and the swept volume, and consequently changes the compression ratio. By manipulating the control shaft angle θ_s the engine can realize different compression ratios for various operating conditions.

Note that in order for the multiple-link mechanism to operate as desired, certain constraints need to be satisfied for its geometric dimensions over the whole range of the control action θ_s. A major requirement is that the lengths of links of the four bar linkage (OABC) need to satisfy the Grashof Condition, which states that the sum of the length of the shortest and longest links need be less than that of the remaining links. This constraint needs to be verified during design of the mechanism dimensions.

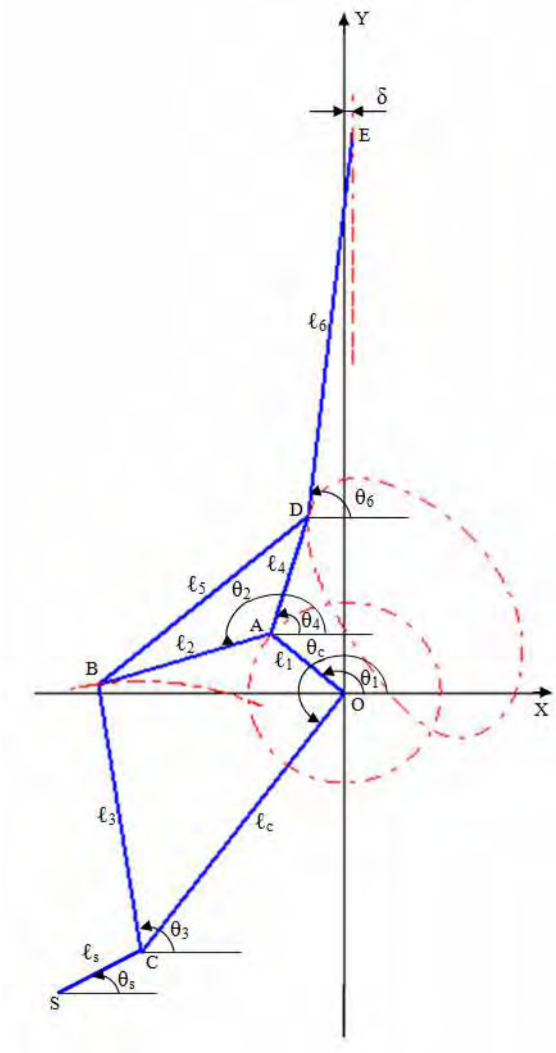


Figure 1   Multiple-link Mechanism

The equations governing the motion of the multiple-link mechanism are derived as follows. In Figure 1, denote the position of the control shaft axis (S) as (X_S, Y_S). The position of control link point (C) is then:

$$X_C = X_S + l_s \cos\theta_s \tag{1}$$

$$Y_C = Y_S + l_s \sin\theta_s \tag{2}$$

Note for the four bar linkage (OABC), we have:

$$\overrightarrow{OA} + \overrightarrow{AB} = \overrightarrow{OC} + \overrightarrow{CB} \tag{3}$$

Rewrite equation (3) and define a new vector $\vec{Z}$ as:

$$\vec{Z} = (X_Z, Y_Z) = \overrightarrow{OA} - \overrightarrow{OC} = \overrightarrow{CB} - \overrightarrow{AB} \tag{4}$$

Since $\overrightarrow{OA}$ and $\overrightarrow{OC}$ are known at each crank angle θ_1 and control shaft angle θ_s, $\vec{Z}$ can be calculated and we have

$$X_Z = l_1 \cos\theta_1 - X_c \tag{5}$$

$$Y_Z = l_1 \sin\theta_1 - Y_c \tag{6}$$

From equation (4) we also have

$$X_Z = l_3 \cos\theta_3 - l_2 \cos\theta_2 \tag{7}$$

$$Y_Z = l_3 \sin\theta_3 - l_2 \sin\theta_2 \tag{8}$$

By eliminating θ_2 from equations (7) and (8), we obtain

$$X_Z \cos\theta_3 + Y_Z \sin\theta_3 = \frac{l_3^2 - l_2^2 + X_Z^2 + Y_Z^2}{2l_3} \tag{9}$$

Define the right hand side of equation (9) as m, we have the solution as

$$\theta_3 = \tan^{-1}\left(\frac{Y_Z}{X_Z}\right) \pm \cos^{-1}\left(\frac{m}{\sqrt{X_Z^2 + Y_Z^2}}\right) \tag{10}$$

The two solutions of θ_3 correspond to the two modes of assembly for the four bar linkage (OABC), which are symmetric about line OC. The solution for Figure 1 is:

$$\theta_3 = \tan^{-1}\left(\frac{Y_Z}{X_Z}\right) + \cos^{-1}\left(\frac{m}{\sqrt{X_Z^2 + Y_Z^2}}\right) \tag{11}$$

Subsequently, θ_2 can be calculated from equation (8) as:

$$\theta_2 = \pi - \sin^{-1}\left(\frac{l_3 \sin\theta_3 - Y_Z}{l_2}\right) \tag{12}$$

Since $\overrightarrow{OB} = \overrightarrow{OC} + \overrightarrow{CB}$, the position of B can be calculated as:

$$X_B = X_C + l_3 \cos\theta_3 \tag{13}$$

$$Y_B = Y_C + l_3 \sin\theta_3 \tag{14}$$

Thus, the motion of the four bar linkage (OABC) is determined. The motion of point (D) can be calculated by making use of the lower link dimensions. Note:

$$\theta_4 = \theta_2 - \cos^{-1}\left(\frac{l_2^2 + l_4^2 - l_5^2}{2l_2 l_4}\right) \tag{15}$$

the Position of (D) can be obtained from $\overrightarrow{OD} = \overrightarrow{OA} + \overrightarrow{AD}$ as:

$$X_D = l_1 \cos\theta_1 + l_4 \cos\theta_4 \tag{16}$$

$$Y_D = l_1 \sin\theta_1 + l_4 \sin\theta_4 \tag{17}$$

Lastly, the position of the piston pin (E) can be obtained. Denote the crank offset as δ, then

$$\theta_6 = \cos^{-1}\left(\frac{\delta - X_D}{l_6}\right) \tag{18}$$

and subsequently from $\overrightarrow{OE} = \overrightarrow{OD} + \overrightarrow{DE}$, the piston pin position is obtained as:

$$Y_E = Y_D + l_6 \sin\theta_6 \tag{19}$$

Thus, with the geometric dimensions of the multiple-link mechanism known, we can calculate the motion of each component for the whole crank shaft revolution, given the control shaft angle. In this study the piston motion is the focus. With the piston motion profile calculated for the crank angle (θ_1) range from 0° to 360°, we can obtain the TDC and BDC positions and their corresponding crank angles. We can also calculate the velocity and acceleration of the piston, the stroke length, the crank angle range of the expansion stroke, the displacement, the combustion chamber volume, and the compression ratio.

3. Multiple-link Mechanism Design Criterion

The main purpose of the multiple-link mechanism is to realize variable compression ratio. However, because the mechanism realizes this feature by changing the piston motion, it may lead to changes in the stroke length, and consequently the displacement, as well. When compression ratio and displacement change, various characteristics of the engine, such as the charging and discharging processes, volumetric efficiency, combustion process and efficiency, and power output all can change. When designing a multiple-link mechanism, all these influences need to be considered. Since compression ratio and stroke length (and displacement) are ultimately determined by the geometry of the mechanism, this paper focuses on finding an approach to obtain the geometric dimensions that give advantageous compression ratio and displacement behavior. This is the basis for further design and optimization of the mechanism.

As discussed before, when engine load decreases, compression ratio should increase to capture the benefits of higher fuel efficiency; while for a decreased engine load demand, a decreased displacement is advantageous, since for decreased displacement and stroke length, the pumping loss and frictional loss are reduced. With variable displacement, less throttling of the intake air will be needed to take care of the load variation. This is a desired property for the engine.

From the above discussion, we can see that a desired design of the multiple-link mechanism is that when the control action changes monotonically, the compression ratio and the displacement should change monotonically in opposite directions. Denote compression ratio as ε and displacement as V_d — note they are both functions of the control shaft angle θ_s — then for the whole range of θ_s, we want to have either Scenario I:

$$\frac{d\varepsilon(\theta_s)}{d\theta_s} < 0 \tag{20}$$

$$\frac{dV_d(\theta_s)}{d\theta_s} > 0 \tag{21}$$

or Scenario II:

$$\frac{d\varepsilon(\theta_s)}{d\theta_s} > 0 \tag{22}$$

$$\frac{dV_d(\theta_s)}{d\theta_s} < 0 \tag{23}$$

Let us investigate Scenario I. Denote the combustion chamber volume as V_c, which is also a function of θ_s. Because

$$\varepsilon(\theta_s) = 1 + \frac{V_d(\theta_s)}{V_c(\theta_s)} \tag{24}$$

we have

$$\frac{d\varepsilon}{d\theta_s} = \frac{V_c \frac{dV_d}{d\theta_s} - V_d \frac{dV_c}{d\theta_s}}{V_c^2} \tag{25}$$

To satisfy (20), we need

$$V_c \frac{dV_d}{d\theta_s} < V_d \frac{dV_c}{d\theta_s} \tag{26}$$

From (21) and (26), it is obvious that we must have

$$\frac{dV_c(\theta_s)}{d\theta_s} > 0 \tag{27}$$

That is, to achieve lower compression ratio when θ_s increases, both V_d and V_c need to increase. Regarding changes of TDC and BDC positions, denote the position of TDC as Y_{TDC} and that of BDC as Y_{BDC}, which are functions of θ_s. Since

$$V_d = \frac{1}{4}\pi d^2 (Y_{TDC} - Y_{BDC}) \tag{28}$$

where d is the cylinder bore diameter, we have

$$\frac{dV_d}{d\theta_s} = \frac{1}{4}\pi d^2 \left(\frac{dY_{TDC}}{d\theta_s} - \frac{dY_{BDC}}{d\theta_s} \right) \tag{29}$$

Also note the change of the combustion chamber volume is caused by the change of the position of TDC, we have

$$\frac{dV_c}{d\theta_s} = -\frac{1}{4}\pi d^2 \frac{dY_{TDC}}{d\theta_s} \tag{30}$$

From (21) and (29) we know

$$\frac{dY_{TDC}}{d\theta_s} > \frac{dY_{BDC}}{d\theta_s} \tag{31}$$

From (27) and (30) we know

$$\frac{dY_{TDC}}{d\theta_s} < 0 \tag{32}$$

Thus we have

$$\frac{dY_{BDC}}{d\theta_s} < \frac{dY_{TDC}}{d\theta_s} < 0 \tag{33}$$

This indicates that when the control shaft angle increases, both TDC and BDC positions will move lower, and BDC will move lower at a bigger rate than that of TDC. Substitute (29) and (30) into (26), and also note (33), it is derived

$$1 < \frac{dY_{BDC}}{d\theta_s} \bigg/ \frac{dY_{TDC}}{d\theta_s} < \varepsilon \tag{34}$$

With similar analysis, it is found that Scenario II gives the same result as in (34). This is expected because Scenario I and II are in essence the same and they only differ by how we define the positive direction of the control action θ_s. The condition shown in (34) indicates that in order the multiple-link mechanism can realize the desired compression ratio and displacement behavior, when the control shaft angle changes, both TDC and BDC positions need to move in the same direction; the change rate with respect to the control shaft angle for the BDC position should be bigger than that of the TDC, but smaller than ε times that of the TDC.

To simplify the calculation, instead of finding the compression ratio corresponding to each control shaft angle, for the condition in (34) we can choose to use the minimum compression ratio ε_0 the mechanism needs to realize, and thus get:

$$1 < \frac{dY_{BDC}}{d\theta_s} \bigg/ \frac{dY_{TDC}}{d\theta_s} < \varepsilon_0 \tag{35}$$

This is a stronger condition for a design to fulfill the requirement. In the following section we will use this condition as a criterion for selection of the mechanism's geometric dimension.

4. Design of Experiments

The previous section presents the condition for a mechanism design to give desirable compression ratio and displacement characteristics. However, it can be seen from Section 2 that the individual and combined influence of the geometric parameters and the control parameter on the motion of the piston, specifically TDC and BDC positions, is not straightforward. How they affect the compression ratio and displacement could be complicated and may not be easily determined. This places a challenge on the design and optimization of the multiple-link mechanism dimensions. In this section, a DoE method is used for creating a series of multiple-link mechanism parameter combinations, which are investigated to obtain the mechanism designs that fulfill the design requirement.

Refering to Figure 1, the multiple-link mechanism geometric design parameters include the following:

$$l_1, l_2, l_3, l_4, l_5, l_6, \delta, X_s, Y_s, l_s$$

First, a specific range for each parameter is set up based on experience, including reasonable physical costraints or limits. Table 1 shows the geometric parameter ranges adopted for this study. Then, in the space spanned by all the parameters, DoE is used to generate a series of parameter combinations, and the properties of the mechanism for each combination are calculated. For this study, we set the range of the control parameter θ_s to be from -45° to 45°, and choose the mechanism properties when θ_s is 0° as the basis for calculation. The targeted compression ratio range is from about 8 to about 14. The combustion chamber volume is set in such a way that when θ_s is 0°, the compression ratio realized is the designed value, which is chosen to be 11 for this study. For each parameter set, the piston motion is calculated for the whole range of θ_s. At each control shaft angle, TDC and BDC positions are determied, and the combustion chamber volume is calculated based on how the TDC position changes from when θ_s is 0°. The displacement is calculated using the TDC and BDC position information. The compression ratio and other characteristics are calculated subsequently.

Table 1 Geometric Parameters and Their Ranges

Parameter	Minimum (mm)	Maximum (mm)
l_1	42	52

l_2	115	135
l_3	120	140
l_4	58	72
l_5	135	155
l_6	150	170
δ	-4	8
X_s	-126	-110
Y_s	-136	-120
l_s	20	40

EasyDoE ToolSuite® from IAV Automotive Engineering, Inc. is used to establish a test plan, which is actually a design plan for the geometric parameters in this study. Using this utility program, ranges and steps are set for each parameter, and a model structure is selected. Creation of the test plan is also dependent on the design method selected. D-optimality minimizes the generalized variance of the parameter estimates. V-optimality minimizes the variance of the predicted response. In this study, the test plan is created such that 60% are D-optimal points, 30% are V-optimal points and 10% are space-filling points. With the design method selected, 158 parameter sets are created for investigation. For each of the created parameter set, it is checked whether the geometry gives a truly operational multiple-link mechanism for the whole range of the control action. i.e., the design needs to satisfy the Grashof Condition. Figure 2 shows the DoE test plan results and the distribution of l_1, l_2, l_3 in three dimensions.

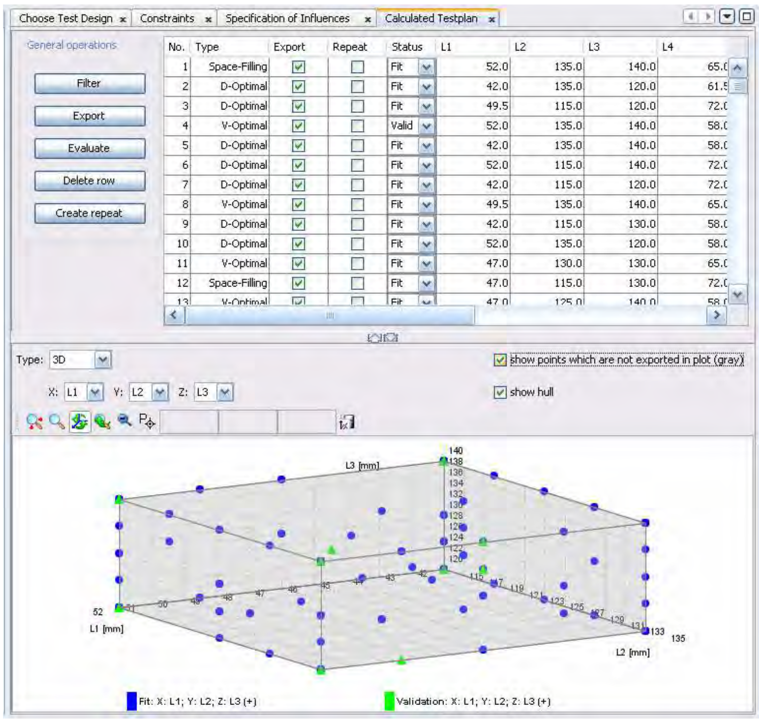


Figure 2   DoE Test Plan Results

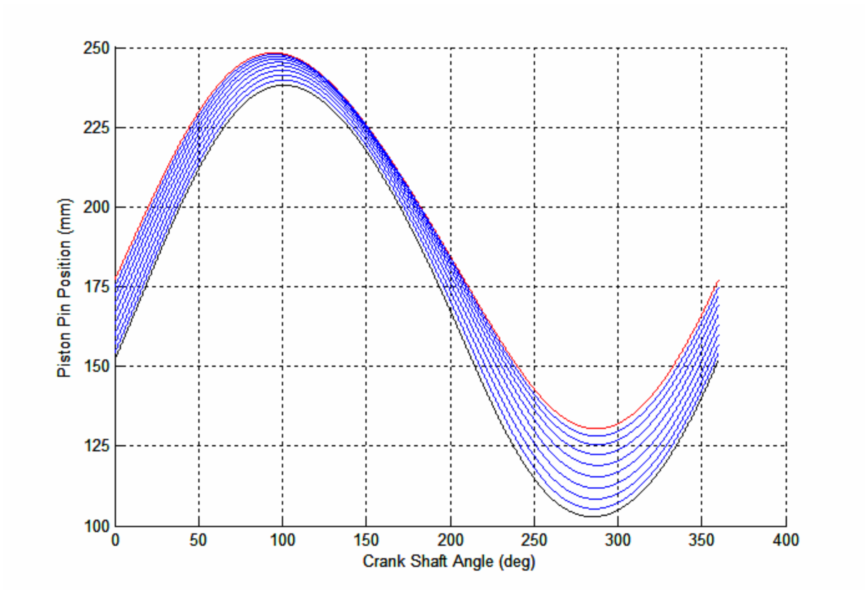
5. Test Plan Results Analysis

Behavior of the multiple-link mechanism is calculated for each parameter set of the test plan over the whole range of the control action. Criterion obtained in Section 3 is checked against each parameter set. For the DoE plan created in Section 4, only a small percentage of designs satisfy the condition. A feasible solution is selected from those designs for further study. Table 2 lists the parameter values of this solution.

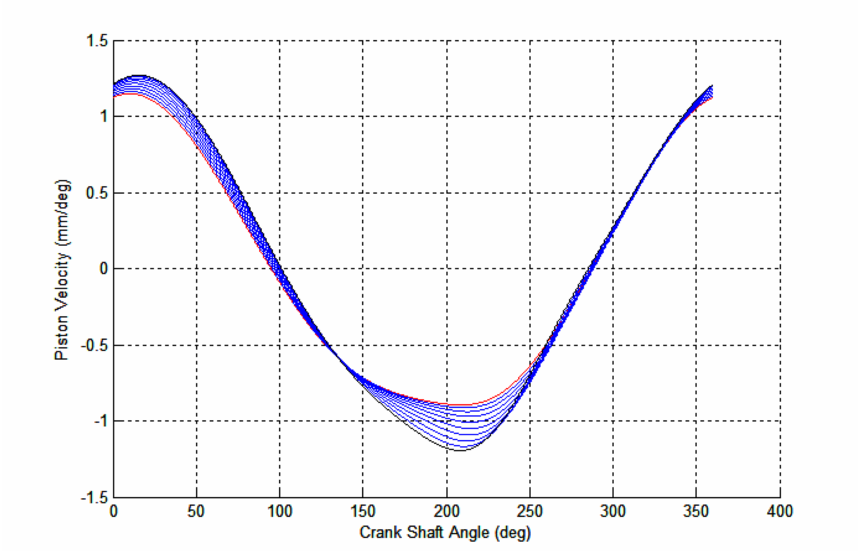
Table 2 Selected Mechanism Parameters

Parameter	Value (mm)
l_1	42
l_2	115
l_3	140
l_4	58
l_5	155
l_6	150
δ	-4
X_s	-110
Y_s	-120
l_s	30

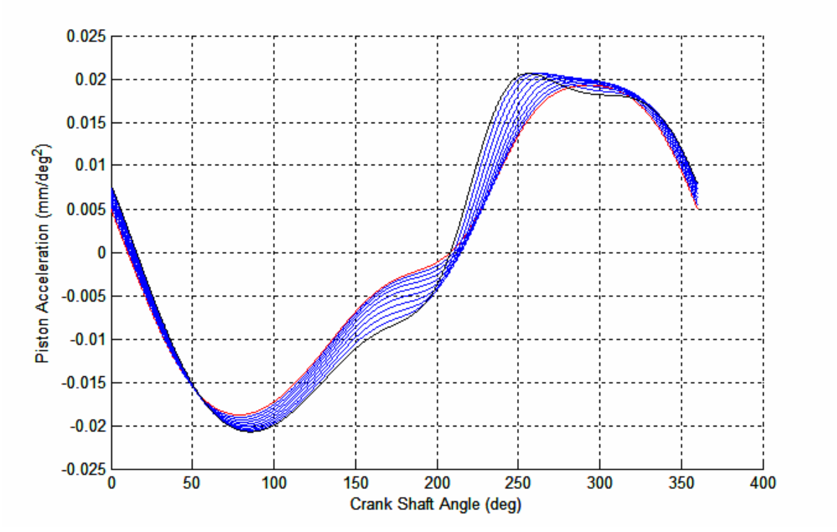
Properties of the selected mechanism design are presented as follows. Figure 3 (a), (b), and (c) show the piston position, velocity and acceleration during a crank revolution respectively, while the control shaft angle changes from -45° to 45° in 10° increments. The red lines correspond to -45° control shaft angle, while the black lines correspond to 45°. These Figures are typical for mechanism geometric dimensions that satisfy the design condition. It can be seen that when the control shaft angle changes, the TDC and BDC move in the same direction but at different rates. When the control shaft rotates counterclockwise, i.e., when θ_s increases, both TDC and BDC move lower; the maximum velocity and acceleration rates increase. Also deserves note is that the crank angle locations corresponding to the TDC and BDC also change.



(a)



(b)



(c)

Figure 3 Piston Position, Velocity and Acceleration

Figure 4 shows the TDC and BDC positions with respect to the control shaft angle. It indicates clearly that when the control shaft turns counterclockwise, both TDC and BDC positions are lowered, and the rate of change of the BDC position is greater than that of the TDC position. Figure 5 shows that the design criterion, i.e., the ratio of the rate of change of the BDC position to that of the TDC position, is between the compression ratio and 1, which indicates that the design satisfies the design condition.

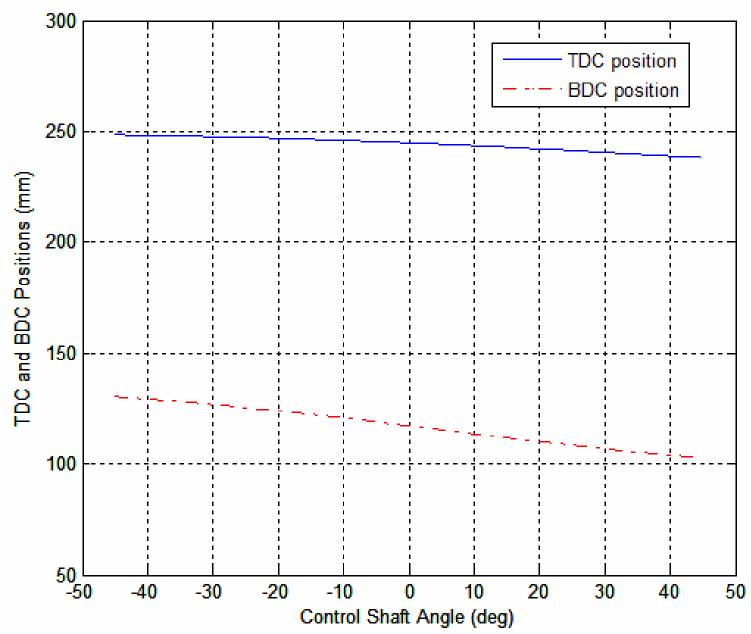


Figure 4 TDC and BDC Positions

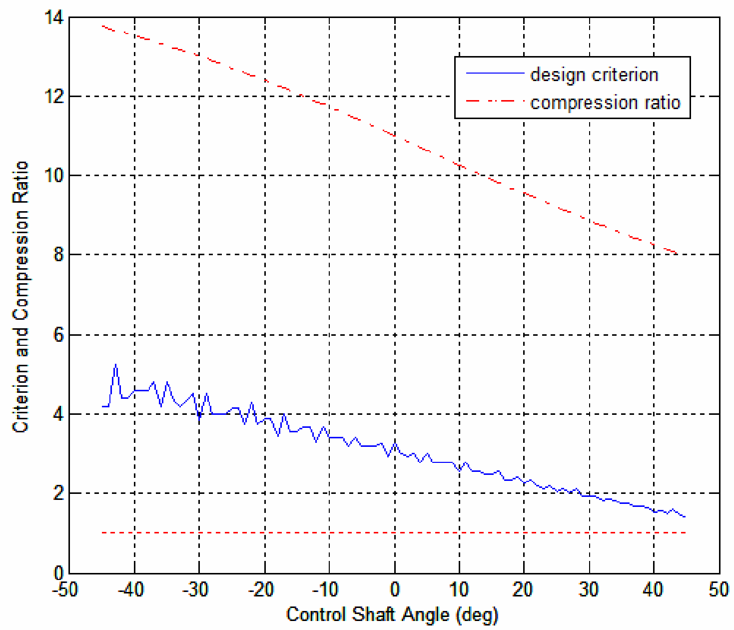


Figure 5 Design Criterion

Figure 6 and 7 show how the compression ratio and stroke length change with respect to the control shaft angle, respectively. It is clear that this case belongs to Scenario I described in Section 3. When the control shaft rotates counterclockwise from -45° to 45°, the compression ratio decreases from 13.74 to 7.98, and the stroke length increases from 117.9 mm to 135.3 mm. Correspondingly, when the bore diameter of the cylinder is 89 mm, the displacement increases from 0.7335 liter to 0.8420 liter, and the combustion chamber volume increases from 0.0576 liter to 0.1207 liter. Figure 8 shows how the displacement and combustion chamber volume change with the control shaft angle.

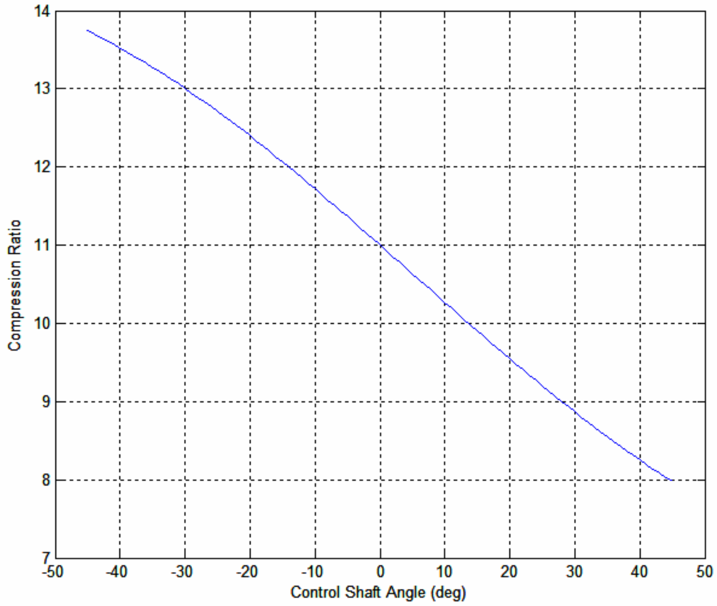


Figure 6   Compression Ratio

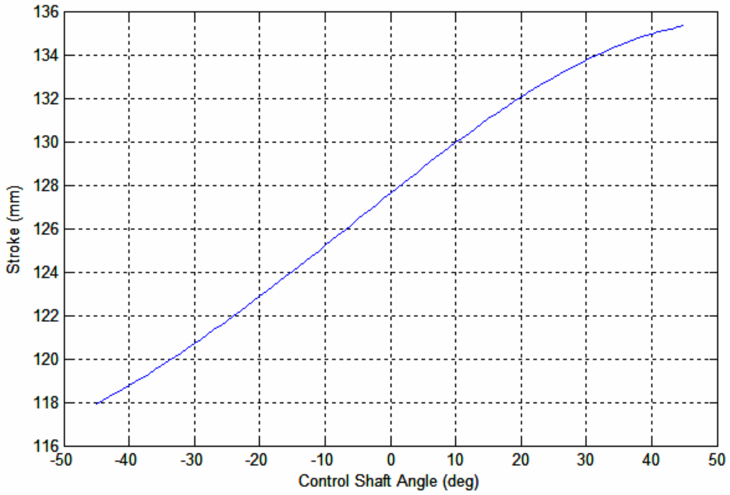


Figure 7   Stroke Length

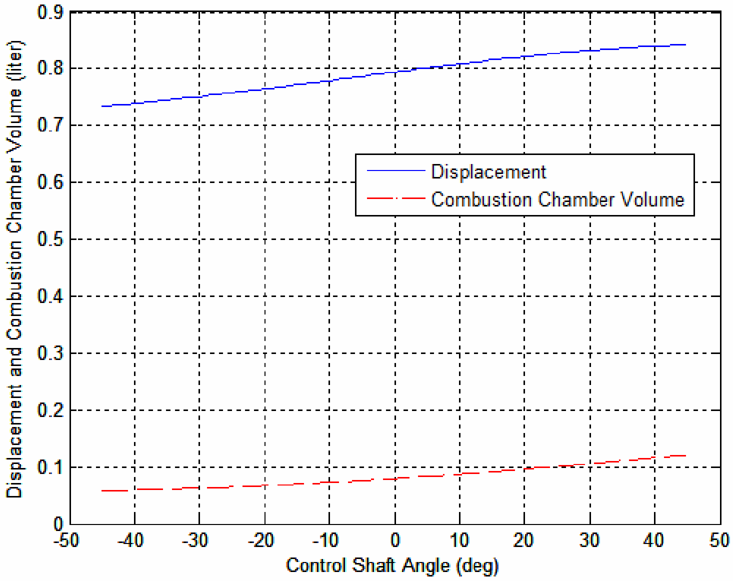


Figure 8   Displacement and Combustion Chamber Volume

Figure 9 shows how the crank angle range of the expansion stroke changes with respect to the control shaft angle. When the control shaft rotates counterclockwise, the curve basically follows a decreasing trend. The Figure shows that when the mechanism is at the highest compression ratio, the expansion stroke covers 192.4 crank angle degree, while when the mechanism is at the lowest compression ratio, the number reduces to 184.0 crank angle degree.

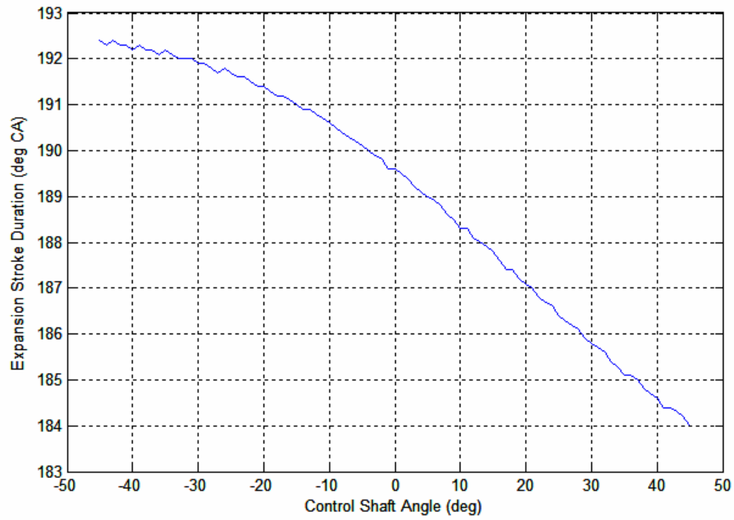


Figure 9   Crank Angle Range of the Expansion Stroke

Figure 10 and Figure 11 show the maximum piston speed and acceleration (absolute values) respectively. When the control shaft rotates counterclockwise, they both basically follow the increasing trend in the range, with a small deviation for the acceleration near the end of the range.

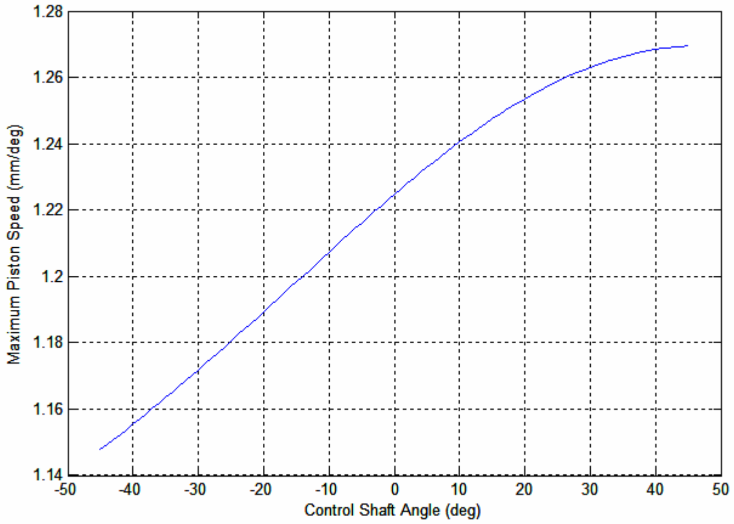


Figure 10   Maximum Piston Speed (Absolute Value)

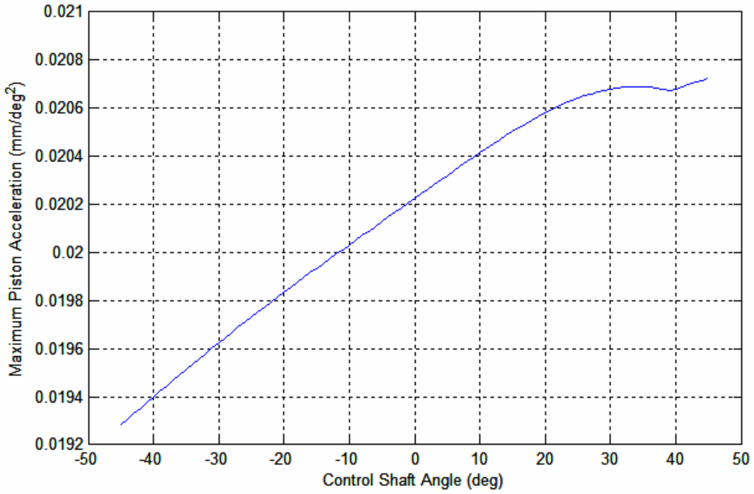


Figure 11   Maximum Piston Acceleration (Absolute Value)

6. Future Work

The approach presented in this paper gives multiple-link mechanism designs that fulfill the desired compression ratio and displacement behavior. As shown in Section 5, major kinematics characteristics of the mechanism can be calculated. To further investigate the engine performance when adopting specific mechanism design, various simulation and testing studies need to be performed. Detailed simulation of the engine performance involves various subsystems, processes and controls, such as charging and discharging processes, fuel delivery, combustible mixture formation, ignition timing, valve timing, combustion properties etc., which are related to the mechanism design and piston kinematics directly or indirectly. Commercially available engine simulation software, such as GT-Power, might be used for this purpose, provided it could take into account the specific piston motion and the subsequent influences on various aspects of the engine performance. Related study in this area will be performed and presented in the future.

Besides performing necessary simulations, it is desirable to build prototype engines of various multiple-link gemmetric design parameters and carry out tests. However, this approach would take a long time with high cost. To deal with this challenge, A&D Co. Ltd. developed a highly flexible testing engine, whose piston is electro-hydraulically controlled and can realize arbitrary motion profiles within the physical limits of the system. Figure 12 shows the configuration of the testing engine system. The system consists of two ADX controllers, which are high performance real time hardware platforms that realize specific control functions. One ADX serves as a universal engine controller that implements such functions as ignition, injection etc. The other ADX mainly implements hydraulic control of the piston and intake and exhaust valves. Based on the targeted engine testing condition, the hydraulic control ADX will create necessary encoder signals that correspond to the rotational behavior of the engine emulated. These signals are sent to the ignition/injection control ADX for engine operation control. The hydraulic control ADX uses the piston position sensor measurement as feedback, and control the piston motion to desired trajectories by manipulating the hydraulic actuator that is attached to it. During the testing engine operation, the hydraulic system will actuate or load the piston in order that it will follow the target curve of motion. Although greatly different from the mechanical design of a real engine, by realizing the same piston motion, the testing engine in a certain sense duplicates the operation and performance of the real engine. The testing results will provide further insights for improving the mechanism design. By changing the target piston motion curve, the system can be used for studying the characteristics of multiple-link mechanism engine of various geometric designs. Preliminary testing shows that the system gives excellent performance in realizing desired piston motion trajectories. Detailed work in this area will be implemented and presented in the future.

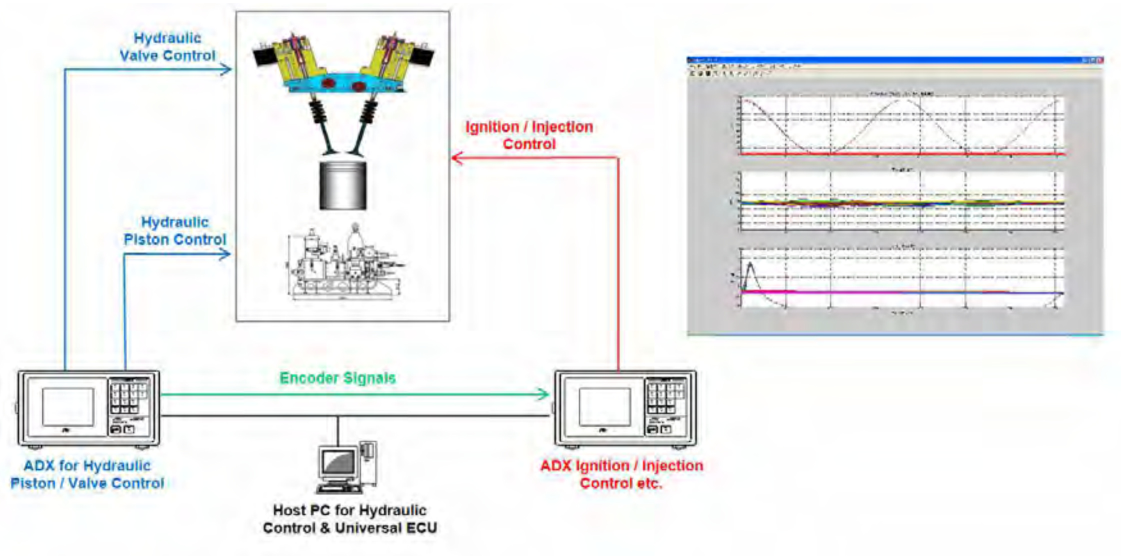


Figure 12   Highly Flexible Testing Engine System

7. Conclusion

This paper presents an approach on how to achieve multiple-link mechanism designs that realize desirable behavior for both compression ratio and displacement. A condition is obtained on how the TDC and BDC positions should change with respect to the control action to fulfill the design requirement. DoE methodology is used for creating sets of geometric designs of the mechanism, which are checked against the condition. Selected design that satisfies the condition is further studied in details. The design approach and obtained results serve as a basis and provide insights for further analysis and optimization of the multiple-link mechanism.

References

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