Geometric Parameter Design of a Multiple-Link Mechanism for Advantageous Compression Ratio and Displacement Characteristics

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Abstract

Variable compression ratio and variable displacement technologies are adopted in internal combustion engines because these features provide further degrees of freedom to optimize engine performance for various operating conditions. This paper focuses on 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. It is determined that a major requirement for the design of this mechanism is when the control action changes monotonically over its whole range, the compression ratio and the displacement should change in opposite directions monotonically. This paper presents an approach on how to achieve multiple-link mechanism geometric designs that fulfill this requirement. First, a necessary and sufficient condition, and a stronger sufficient condition are 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, for which kinematics are calculated and checked against the conditions. A feasible design that satisfies the conditions is selected and detailed study on such characteristics as piston motion, stroke length, displacement, combustion chamber volume, and compression ratio etc. is performed. The design approach and obtained results serve as a basis for further analysis and optimization of the multiple-link mechanism.

Introduction

Variable compression ratio technology has long been recognized as a method to improve engine fuel economy [1, 2, 3, 4, 5, 6, 7]. At low power levels the engine operates at higher compression ratio to capture the benefits of higher thermal efficiency, while at high power levels the engine operates at 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. Variable stroke and consequently variable displacement is also a desired feature [8, 9, 10, 11, 12, 13]. For fixed displacement engines, the load variation is usually handled by throttling the intake air. For a variable displacement engine, when engine load demand decreases, the displacement is decreased to at least partially take care of the need. With less throttling and decreased stroke, the pumping loss and frictional loss are reduced, which is advantageous for fuel economy. Variable displacement also provides the potential for improved exhaust emissions.

There are various approaches to realize variable compression ratio, such as moving the cylinder head, changing the combustion chamber volume through a secondary piston, moving the crankshaft axis, and using multiple links for connection between piston and crankshaft etc. There are different types of mechanical realization of variable displacement as well. For the mechanisms that are capable of realizing both variable compression ratio and variable displacement, it makes sense to study both features in a comprehensive way to improve the engine performance. This paper focuses on studying a multiple-link connecting rod mechanism [5, 6, 7]. This mechanism was initially designed to realize variable compression ratio by varying the piston motion, specifically the TDC and BDC positions of the engine. So far the studies of the mechanism have focused on the variable compression ratio characteristics. However, a good design of the geometry of this mechanism should also provide advantageous displacement behavior of the engine. This paper presents an approach to obtain geometric designs of the mechanism that realize both advantageous compression ratio and displacement characteristics.

The paper is organized as follows. First a description of the multiple-link mechanism is given. Equations describing the kinematics of the mechanism are derived. Then a necessary and sufficient condition, and a stronger sufficient condition are 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. Next a Design of Experiments (DoE) methodology is used for creating sets of geometric parameters of the mechanism, 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 conditions.

A feasible design that satisfies the conditions is selected for further study. Such characteristics as detailed piston motion profile, piston velocity and acceleration, displacement, combustion chamber volume, compression ratio and crank angle range of the expansion stroke etc. are investigated for the design.

Multiple-Link Mechanism

A schematic 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 *C* of a control shaft *SC*. When the crankshaft *OA* 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 *C* and the control shaft axis *S* is called control shaft length in this study, and denoted by l_s. 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. By manipulating the control shaft angle θ_s the mechanism can realize different compression ratios and displacements for various engine 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 lengths 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.

The equations governing the kinematics of the multiple-link mechanism are derived as follows. First the motion of the four bar linkage *OABC* need be obtained. From Figure 1, it is evident the position of crank pin *A* is:

$$X_A = l_1 \cos\theta_1 \tag{1}$$

$$Y_A = l_1 \sin\theta_1 \tag{2}$$

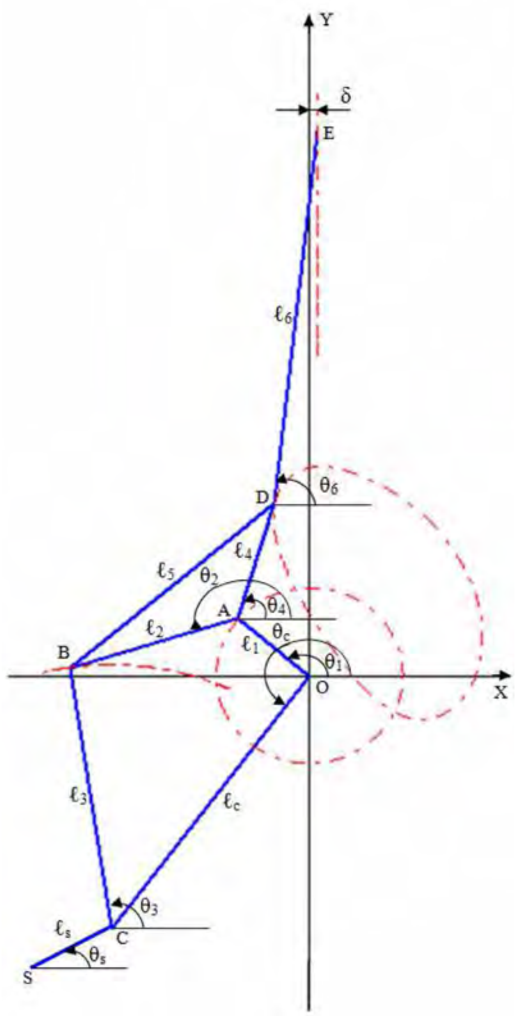

Figure 1. Multiple-link Mechanism

Denote the position of the control shaft axis *S* (X_S, Y_S), then the position of the control link point *C* is also easily obtained:

$$X_C = X_S + l_s \cos\theta_s \tag{3}$$

$$Y_C = Y_S + l_s \sin\theta_s \tag{4}$$

Note:

$$\overrightarrow{OA} + \overrightarrow{AB} = \overrightarrow{OC} + \overrightarrow{CB} \tag{5}$$

Rewrite equation (5) and define a new vector $\overrightarrow{OZ}$ as:

$$\overrightarrow{OZ} = \overrightarrow{OA} - \overrightarrow{OC} = \overrightarrow{CB} - \overrightarrow{AB} \tag{6}$$

Since $\overrightarrow{OA}$ and $\overrightarrow{OC}$ are already obtained by equations (1)(2)(3)(4) for each crank angle θ_1 and control shaft angle θ_s, $\overrightarrow{OZ}$ is known. Denote the position of Z (X_Z, Y_Z), then from the second part of equation (6) we have:

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

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

From equations (7) and (8), and also considering the feasible ranges of the angles in Figure 1, we can obtain:

$$\theta_3 = \tan^{-1}\left(\frac{Y_Z}{X_Z}\right) + \cos^{-1}\left(\frac{l_3^2 - l_2^2 + X_Z^2 + Y_Z^2}{2l_3\sqrt{X_Z^2 + Y_Z^2}}\right) \tag{9}$$

$$\theta_2 = \pi - \sin^{-1}\left(\frac{l_3 \sin\theta_3 - Y_Z}{l_2}\right) \tag{10}$$

Note $\overrightarrow{OB} = \overrightarrow{OC} + \overrightarrow{CB}$, the position of control link point B can be obtained as:

$$X_B = X_C + l_3 \cos\theta_3 \tag{11}$$

$$Y_B = Y_C + l_3 \sin\theta_3 \tag{12}$$

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

$$\theta_4 = \theta_2 - \cos^{-1}\left(\frac{l_2^2 + l_4^2 - l_5^2}{2l_2 l_4}\right) \tag{13}$$

From $\overrightarrow{OD} = \overrightarrow{OA} + \overrightarrow{AD}$ it is obtained:

$$X_D = X_A + l_4 \cos\theta_4 \tag{14}$$

$$Y_D = Y_A + l_4 \sin\theta_4 \tag{15}$$

Lastly, the position of the piston pin E can be obtained. Denote the crank offset δ, it is evident:

$$\theta_6 = \cos^{-1}\left(\frac{\delta - X_D}{l_6}\right) \tag{16}$$

From $\overrightarrow{OE} = \overrightarrow{OD} + \overrightarrow{DE}$, the position of piston pin E is obtained as:

$$Y_E = Y_D + l_6 \sin\theta_6 \tag{17}$$

Thus with the geometric dimensions of the multiple-link mechanism known, and given the control shaft angle, we can calculate the motion of each component for the whole crank shaft revolution. In this study the piston motion is the focus. With the piston motion profile calculated for the whole 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 displacement, the combustion chamber volume, the compression ratio and other characteristics of interest.

Design Conditions

While the multiple-link mechanism is initially designed to realize variable compression ratio, due to its realizing the feature by changing the piston motion, it may lead to changes in the stroke length, and consequently the displacement as well. Since the piston motion is 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.

Let us first investigate the desired behavior for compression ratio and displacement. On one hand, when engine load decreases, obviously it is advantageous that the displacement also decreases. When the demand in load change is handled by varied displacement, less throttling is needed, which reduces the pumping loss. A decreased stroke length also reduces the frictional loss. On the other hand, compression ratio should increase when engine load decreases, which helps capture the benefits of higher thermal efficiency, and consequently provides better fuel economy. Thus we see that the desired design of the multiple-link mechanism is when the control action changes monotonically, the compression ratio and the displacement should change monotonically in opposite directions.

Denote compression ratio ε and displacement 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:

$$\frac{d\varepsilon(\theta_s)}{d\theta_s} < 0 \tag{18}$$

$$\frac{dV_d(\theta_s)}{d\theta_s} > 0 \tag{19}$$

or Scenario II:

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

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

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

$$\varepsilon(\theta_s) = 1 + \frac{V_d(\theta_s)}{V_c(\theta_s)} \tag{22}$$

it is obtained:

$$\frac{d\varepsilon}{d\theta_s} = \frac{V_c \frac{dV_d}{d\theta_s} - V_d \frac{dV_c}{d\theta_s}}{V_c^2} \tag{23}$$

To satisfy (18), we need

$$V_c \frac{dV_d}{d\theta_s} < V_d \frac{dV_c}{d\theta_s} \tag{24}$$

From (19) and (24), it is obvious that we must have:

$$\frac{dV_c(\theta_s)}{d\theta_s} > 0 \tag{25}$$

(19) and (25) indicate that to achieve a 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 Y_{TDC} and that of BDC Y_{BDC}, which are functions of θ_s. Since

$$V_d = \frac{1}{4}\pi d^2 \left(Y_{TDC} - Y_{BDC}\right) \tag{26}$$

where *d* is the cylinder bore diameter, we have:

$$\frac{dV_d}{d\theta_s} = \frac{1}{4}\pi d^2 \left(\frac{dY_{TDC}}{d\theta_s} - \frac{dY_{BDC}}{d\theta_s}\right) \tag{27}$$

Also note the change of the combustion chamber volume is caused by the change of the position of TDC, it is obtained:

$$\frac{dV_c}{d\theta_s} = -\frac{1}{4}\pi d^2 \frac{dY_{TDC}}{d\theta_s} \tag{28}$$

From (19) and (27) we know

$$\frac{dY_{TDC}}{d\theta_s} > \frac{dY_{BDC}}{d\theta_s} \tag{29}$$

From (25) and (28) we know

$$\frac{dY_{TDC}}{d\theta_s} < 0 \tag{30}$$

Thus it is obtained:

$$\frac{dY_{BDC}}{d\theta_s} < \frac{dY_{TDC}}{d\theta_s} < 0 \tag{31}$$

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 (27) and (28) into (24), and also note (31) and (22), it is derived:

$$1 < \frac{dY_{BDC}}{d\theta_s} \bigg/ \frac{dY_{TDC}}{d\theta_s} < \varepsilon \tag{32}$$

With similar analysis, it is found that Scenario II gives the same result as in (32). 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 (32) indicates that in order for the multiple-link mechanism to 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.

The above process of deriving (32) indicates it is a necessary condition for the desired mechanism behavior. However, we can start from (32), reverse the process and end up with Scenario I and II as well. Thus it is proved that (32) is a necessary and sufficient condition.

Using condition (32) to determine if a design will give the desired mechanism behavior still involves finding out the compression ratio over the whole range of the control shaft angle, which needs the displacement and combustion chamber volume information. To simplify, we can choose to use the minimum compression ratio ε_0 the mechanism needs to realize, and turn the condition into:

$$1 < \frac{dY_{BDC}}{d\theta_s} \bigg/ \frac{dY_{TDC}}{d\theta_s} < \varepsilon_0 \tag{33}$$

This is a stronger condition for a design to fulfill the requirement. Since the minimum compression ratio a design needs to realize is set in advance and known, condition (33) can be checked by using just the information of how the TDC and BDC positions change with the control shaft angle, without the need to calculate the displacement, the combustion chamber volume, and the compression ratio over the whole range of the control shaft angle. It is evident (33) is a sufficient but not necessary condition.

One thing that deserves mentioning is that the conditions derived above apply not only to the multiple-link mechanism this paper studies. So long as a mechanism realizes the features by changing the TDC and BDC positions of the engine, the conditions apply.

While the conditions provide insight for the design of the geometric dimensions of the mechanism, it can be seen from the equations describing the kinematics of the multiple-link mechanism that the individual and combined influence of the geometric parameters on how the TDC and BDC positions will change is not straightforward. An analytical solution of what geometric parameter set of the mechanism will satisfy the conditions could not be easily determined. This places a challenge on the design and optimization of the multiple-link mechanism geometric dimensions. In this paper the approach of using DoE and numerical calculation is adopted.

Design of Experiments

DoE method is used for creating a series of multiple-link mechanism geometric parameter combinations, which are investigated to obtain the mechanism designs that fulfill the conditions.

Referring 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 constraints or limits. Then, in the space spanned by all the parameters, DoE is used to generate a series of parameter sets, and the properties of the mechanism for each parameter set are calculated. In this study, an experiment plan is created which consists of D-optimal points, V-optimal points and space-filling points. With the design method selected, several hundreds parameter sets are created for investigation. For each of the created parameter sets, the piston motion is calculated for the whole range of the control shaft angle. Grashof Condition is checked to make sure the geometry gives a truly operational multiple-link mechanism. After the TDC and BDC positions are determined, the design conditions are checked to find out if the geometry leads to a mechanism that gives both advantageous compression ratio and displacement characteristics.

For this study, the range of the control parameter θ_s is set to be from −45° to 45°, and the mechanism properties when θ_s is 0° are chosen 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 the DoE plan created above, it is found out that only a very small percentage of designs satisfy the conditions. A feasible solution is selected for more detailed study. Piston motion characteristics, including position, velocity and acceleration are calculated. The displacement is calculated using the TDC and BDC position information. The combustion chamber volume is calculated based on how the TDC position changes from when θ_s is 0°. The compression ratio is calculated subsequently. Other characteristics such as maximum piston speed and acceleration rate, crank angle range of the expansion stroke etc. are also studied.

Properties of the Selected Design

Table 1 lists the parameter values of a geometric design of the mechanism that satisfies the conditions. Properties of this design are presented below.

Table 1. Geometric Parameters of the Selected Design

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

Figures 2(a), (b), and (c) show the piston position, velocity, and acceleration respectively during a crank revolution when the control shaft angle changes from −45° to 45° in 5° increments. The red lines correspond to −45° control shaft angle, the green lines correspond to 0°, and the black lines correspond to 45°. These figures are typical for mechanism geometric dimensions that satisfy the design condition. It can be seen from Figure 2(a) that when the control shaft angle θ_s increases, i.e., when the control shaft rotates counterclockwise, both the TDC and BDC move lower; the TDC and BDC move in the same direction with the BDC moving at a higher rate than the TDC. These observations are in consistence with (31). Also of note from Figure 2(a) is that when the control shaft angle changes, the crank angle locations corresponding to the TDC and BDC also change. This indicates that the crank angle ranges of the engine working cycle strokes might change as well. Figure 2(b) and 2(c) show that the piston velocity and acceleration change with the control shaft angle in a more complex way; when θ_s increases, the maximum piston speed and acceleration rate (absolute values) both basically increase.

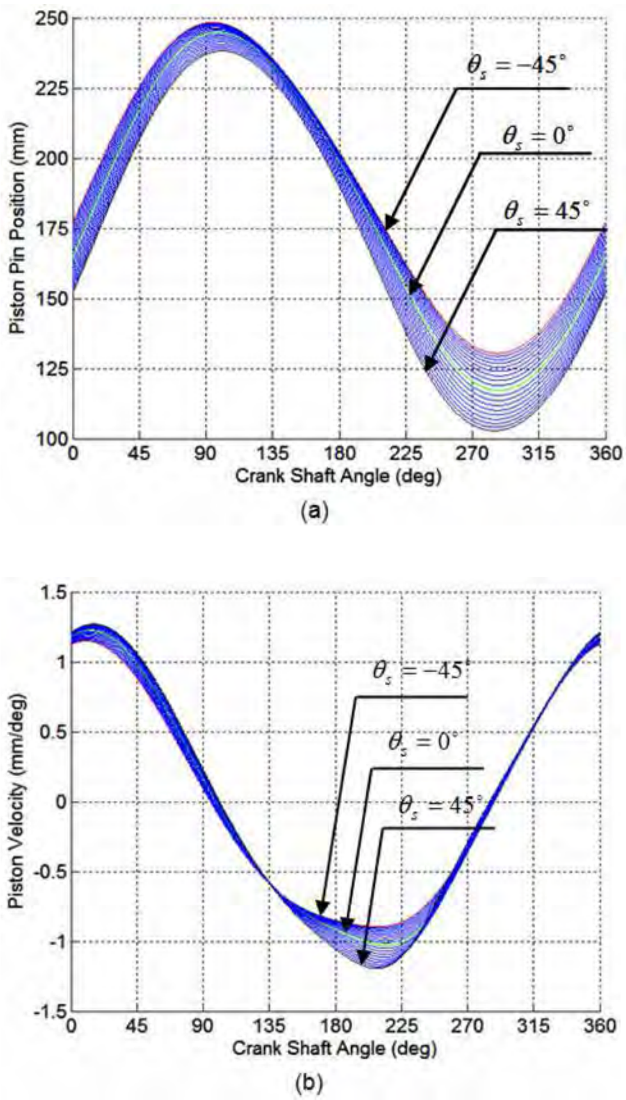

Figure 2.

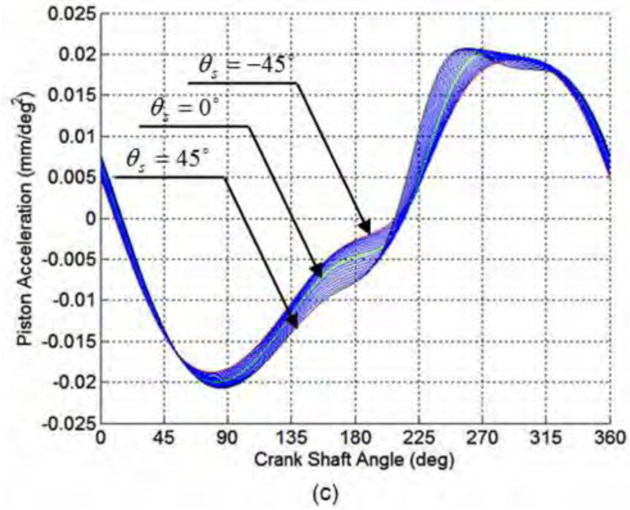

Figure 2. (cont.) Piston Position, Velocity and Acceleration

Figure 3 specifically shows the TDC and BDC positions with respect to the control shaft angle. It is evident that when the control shaft turns counterclockwise, both the TDC and BDC positions are lowered, and the rate of change of the BDC position is greater than that of the TDC position. This further confirms the observations from Figure 2(a) and indicates the design conforms to (31).

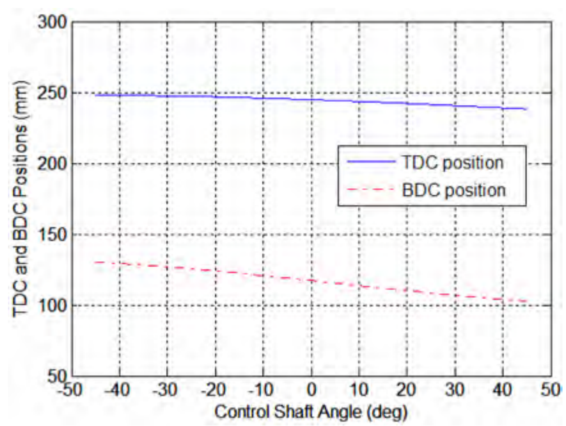

Figure 3. TDC and BDC Positions

With the TDC and BDC positions obtained, it is trivial to calculate the stroke length. Figure 4 shows that the stroke length increases from 117.9 mm to 135.3 mm when the control shaft rotates counterclockwise from −45° to 45°.

To investigate how the conditions (32) and (33) apply to this design, let us first find out how the displacement, the combustion chamber volume, and the compression ratio change with the control shaft angle, although checking condition (33) does not need these information. Since the TDC and BDC positions are already obtained for each control shaft angle, the displacement can be easily calculated using (26), given the cylinder bore diameter, which is 89.0 mm for this study. To calculate the combustion chamber volume for the whole range of the control shaft angle, we need make use of the targeted compression ratio when θ_s is 0°. At this control shaft angle, the stroke length is 127.6 mm and the displacement is 0.794 liter. To realize the targeted compression ratio of 11, the combustion chamber volume is set to 0.0794 liter. Since the change in the combustion chamber volume is caused by the change of the TDC position, when the control shaft angle changes from 0°, based on how much change is incurred in the TDC position, the corresponding combustion chamber volume can be calculated. After the combustion chamber volume is obtained, the compression ratio is subsequently obtained using (22).

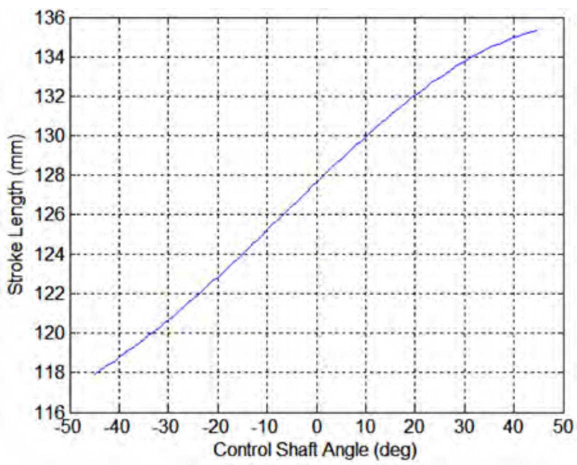

Figure 4. Stroke Length

Figure 5 shows how the displacement and the combustion chamber volume change with the control shaft angle. When the control shaft rotates counterclockwise from −45° to 45°, the displacement increases from 0.733 liter to 0.842 liter, which is an increase of 14.8%; and the combustion chamber volume increases from 0.058 liter to 0.121 liter, which is an increase of 109.5%. Figure 6 shows how the compression ratio changes with the control shaft angle. In the same control shaft angle range, the compression ratio decreases from 13.74 to 7.98. The behavior of the displacement and the compression ratio clearly indicates that this design belongs to Scenario I described earlier and conforms to (18) and (19).

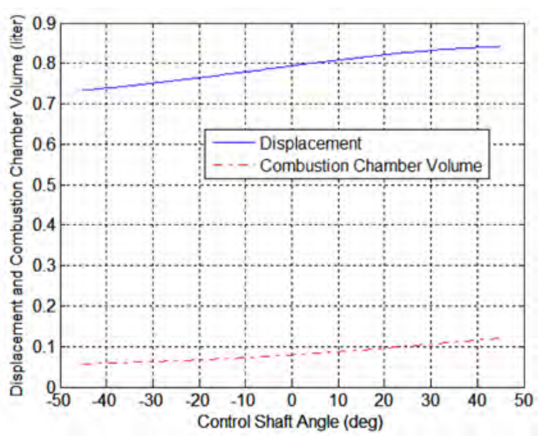

Figure 5. Displacement and Combustion Chamber Volume

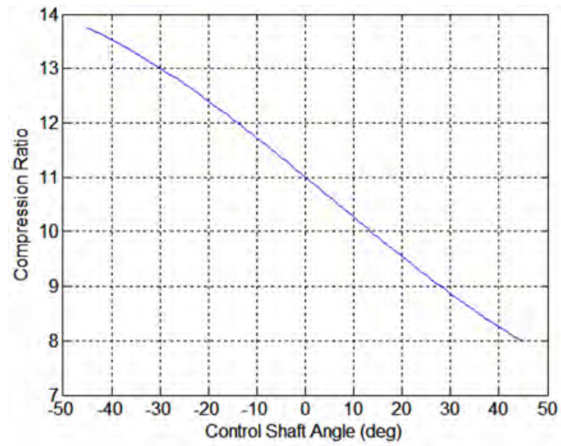

Figure 6. Compression Ratio

Based on how the TDC and BDC positions change with the control shaft angle, the design criterion, i.e., the ratio of the rate of change of the BDC position to that of the TDC position, can be easily calculated. Figure 7 shows that the design criterion basically follows a decreasing trend when the control shaft angle changes from −45° to 45°, and it is between 1 and the compression ratio. This indicates that the design satisfies condition (32). Note that the criterion is actually less than the minimum compression ratio, which is about 8, for the whole range of the control shaft angle, which indicates that the stronger condition (33) is also satisfied in this case.

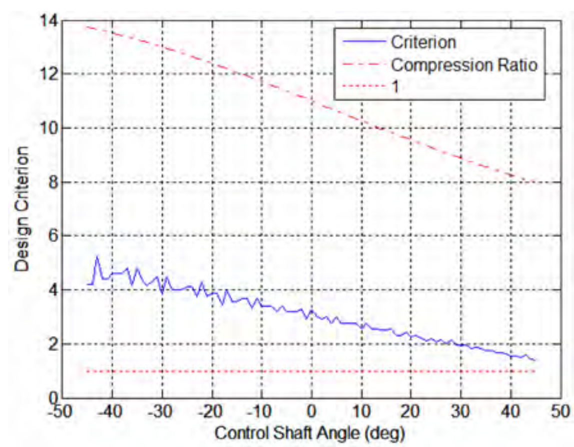

Figure 7. Design Criterion

The above analysis shows that the design satisfies the conditions and gives compression ratio and displacement behavior as desired. Some other characteristics indicated by the data may also be of interest for further studies of the mechanism performance. Maximum piston speed and acceleration rate (absolute values) are shown in Figure 8 and 9 respectively. When the control shaft rotates counterclockwise, they both basically follow the increasing trend in the range studied, with a small deviation for the acceleration rate near the end of the range.

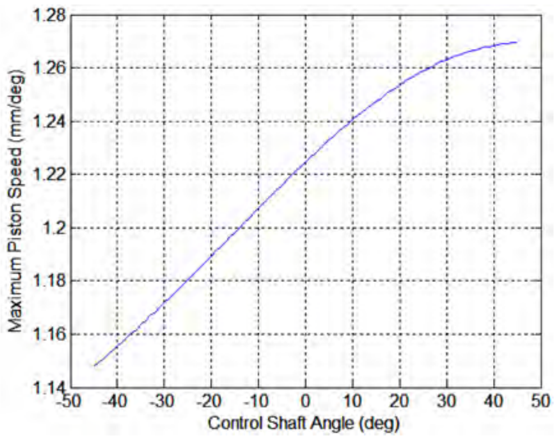

Figure 8. Maximum Piston Speed (Absolute Value)

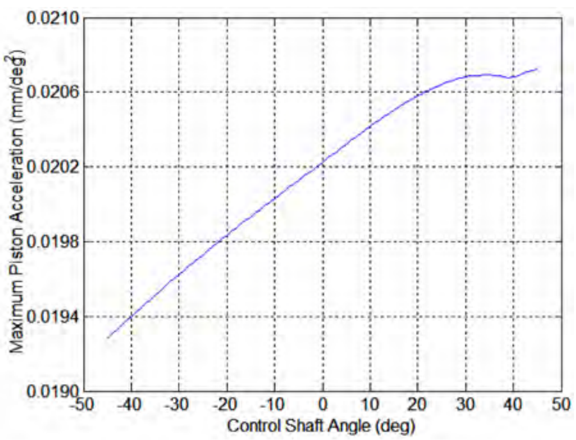

Figure 9. Maximum Piston Acceleration (Absolute Value)

Also of interest is how the crank angle range of the expansion stroke changes with respect to the control shaft angle. Figure 10 shows that when the control shaft rotates counterclockwise, the crank angle range of the expansion stroke basically follows a decreasing trend. When the mechanism is at the highest compression ratio, the expansion stroke covers 192.4 crank angle degrees, while when the mechanism is at the lowest compression ratio, the number reduces to 184.0 crank angle degrees.

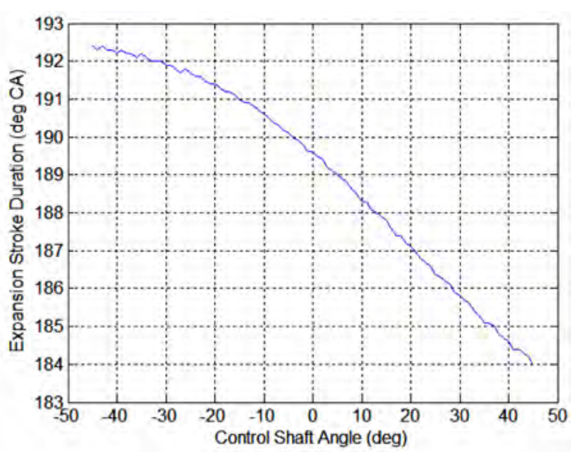

Figure 10. Crank Angle Range of the Expansion Stroke

The study so far basically focuses on only the kinematics of the mechanism. To further investigate the engine performance when adopting a specific mechanism design, detailed study of dynamics involving various subsystems, processes and controls, such as charging and discharging processes, volumetric efficiency, fuel delivery, combustible mixture formation, ignition timing, valve timing, combustion process and efficiency, and power output etc., which are related to the mechanism design and piston motion directly or indirectly, need be performed. Related study in these areas is not covered in this paper.

Summary/Conclusions

This paper presents an approach on how to achieve multiple-link mechanism designs that realize advantageous behavior for both compression ratio and displacement. Conditions are 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 conditions. Selected design that satisfies the conditions is further studied in detail. The design approach and obtained results serve as a basis and provide insights for further analysis and optimization of the multiple-link mechanism.

References

1. Schwaderlapp, M., Habermann, K., and Yapici, K., "Variable Compression Ratio - A Design Solution for Fuel Economy Concepts," SAE Technical Paper 2002-01-1103, 2002, doi:10.4271/2002-01-1103.
2. Mendler, C. and Gravel, R., "Variable Compression Ratio Engine," SAE Technical Paper 2002-01-1940, 2002, doi:10.4271/2002-01-1940.
3. Roberts, M., "Benefits and Challenges of Variable Compression Ratio (VCR)," SAE Technical Paper 2003-01-0398, 2003, doi:10.4271/2003-01-0398.
4. Rabhi, V., Beroff, J., and Dionnet, F., "Study of a Gear-Based Variable Compression Ratio Engine," SAE Technical Paper 2004-01-2931, 2004, doi:10.4271/2004-01-2931.
5. Moteki, K., Aoyama, S., Ushijima, K., Hiyoshi, R. et al., "A Study of a Variable Compression Ratio System with a Multi-Link Mechanism," SAE Technical Paper 2003-01-0921, 2003, doi:10.4271/2003-01-0921.
6. Takahashi, N., Aoyama, S., Moteki, K., and Hiyoshi, R., "A Study Concerning the Noise and Vibration Characteristics of an Engine with Multiple-Link Variable Compression Ratio Mechanism," SAE Technical Paper 2005-01-1134, 2005, doi:10.4271/2005-01-1134.
7. Hiyoshi, R., Aoyama, S., Takemura, S., Ushijima, K. et al., "A Study of a Multiple-link Variable Compression Ratio System for Improving Engine Performance," SAE Technical Paper 2006-01-0616, 2006, doi:10.4271/2006-01-0616.
8. Freudenstein F. and Maki E. R., "Development of an Optimum Variable-Stroke Internal-Combustion Engine Mechanism from the Viewpoint of Kinematics Structure",

Journal of Mechanism, Transmission, and Automation in Design, June 1983, Vol. 105

9. Alsterfalk, M., Filipi, Z., and Assanis, D., "The Potential of the Variable Stroke Spark-Ignition Engine," SAE Technical Paper 970067, 1997, doi:10.4271/970067.
10. Wong, V., Stewart, M., Lundholm, G., and Hoglund, A., "Increased Power Density via Variable Compression/ Displacement And Turbocharging Using The Alvar-Cycle Engine," SAE Technical Paper 981027, 1998, doi:10.4271/981027.
11. Rosso, P., Beard, J., and Blough, J., "A Variable Displacement Engine with Independently Controllable Stroke Length and Compression Ratio," SAE Technical Paper 2006-01-0741, 2006, doi:10.4271/2006-01-0741.
12. Yamin Jehad A. A. and Dado Mohammad H., "Performance Simulation of a Four-Stroke Engine with Variable Stroke-Length and Compression Ratio", Applied Energy 77 (2004), Page 447-463
13. Tavares Fernando, Johri Rajit and Filipi Zoran, "Simulation Study of Advanced Variable Displacement Engine Coupled to Power-Split Hydraulic Hybrid Powertrain", Journal of Engineering for Gas Turbines and Power, December 2011, Vol. 133

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