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A Novel Extended Expansion Engine Mechanism

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Abstract

Due to the increasing population and intense energy consumption, energy efficiency and efficient use of energy in the power units in the vehicle are important concepts for our earth. Particularly in hybrid engine vehicles produced in recent times, Atkinson (or Miller) internal combustion engines are preferred over conventional internal combustion engines. Atkinson's work from 1882 to the present day in many extended expansion engine mechanism design is presented. This paper deals with the design of a novel extended expansion engine mechanism that is intended to achieve trefoil hypotrochoid curve motion in the crankshaft connection using the planetary gear system. The kinematic equations related to the presented new crank-conrod mechanism have been obtained and compared with the conventional crank-conrod mechanism. As a result of the paper, the equations required for kinematic analysis were derived, and the kinematic data of the novel designed mechanism were compared and evaluated according to the data of the conventional system. As a result of this evaluation, it was shown that the desired extended expansion stroke could be obtained by using this mechanism and the parameters affecting the kinematic analysis were determined. However, it has been seen that by changing the critical design parameters, different piston path can be achieved with the novel designed mechanism.

Keywords: Extended expansion; planetary gear mechanism; kinematics; crank-conrod mechanism; trefoil hypotrochoid curve

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1. Introduction

The development of extended expansion (also called over expanded) engines started with Atkinson cycle engines, invented in 1882 and patented in 1886 and 1887 [1, 2]. In Atkinson cycle engines, the expansion and exhaust strokes are longer than the suction and compression strokes. In this way, more work can be done in one cycle. Compared with a conventional Otto cycle engine, the extended expansion engine can provide a greater expansion rate and thus a higher thermal efficiency, while preventing knocking with a normal effective compression ratio [3]. When Atkinson cycle engines are examined, the expansion and exhaust strokes are made with the aid of complex mechanisms and connections, which are longer than suction and compression strokes.

The expansion in the Otto cycle is on the in-cylinder pressurized atmosphere after stroke. In the Atkinson cycle, expansion due to the full expansion occurs at the pressure inside the cylinder after stroke. It is noteworthy that an Atkinson cycle engine is more efficient than a conventional four stroke Otto cycle engine [4].

Another alternative to the Atkinson cycle is the Miller cycle (also called the modified Atkinson cycle). The most noticeable difference between the Miller cycle and the Atkinson cycle is that when the complete expansion stroke occurs in the Atkinson cycle, a controlled (or partial) extended expansion stroke occurs in the Miller cycle. Obviously, the Atkinson cycle is more efficient than the Miller cycle, since the work done in a cycle in the Atkinson cycle will be more than the Miller cycle and also Otto cycle [3-7].

Several mechanisms have been tried for extended expansion engines. The first attempt at these mechanisms was made by Atkinson in 1882. Atkinson introduced the opposite piston engine known as the "differential engine" in this paper and obtained a variable cylinder volume in operation by connecting the opposite pistons with the complicated mechanism [1, 8].

In another extended expansion engine version, designed by Atkinson in 1887, the piston is not directly connected to the crankshaft and a complicated mechanism consisting of many rods is also used [2, 8, 9].

Atkinson's expansion ratio of extend expansion engine is about 1.78 times the compression ratio, and with the long expansion stroke, the higher thermal efficiency is achieved, while the short compression ratio reduces the engine knock [2-3].

After Atkinson's work, multi-link mechanical systems have been researched and developed so that the Atkinson cycle can be obtained. The most noteworthy of these designs is the Honda EXLink (Extended Expansion Linkage), which is the world's first mass-produced multi-link Atkinson cycle engine (Fig 1). Expansion ratio of the Honda EXLink engine is about 1.5 times the compression ratio. [3, 10-12].

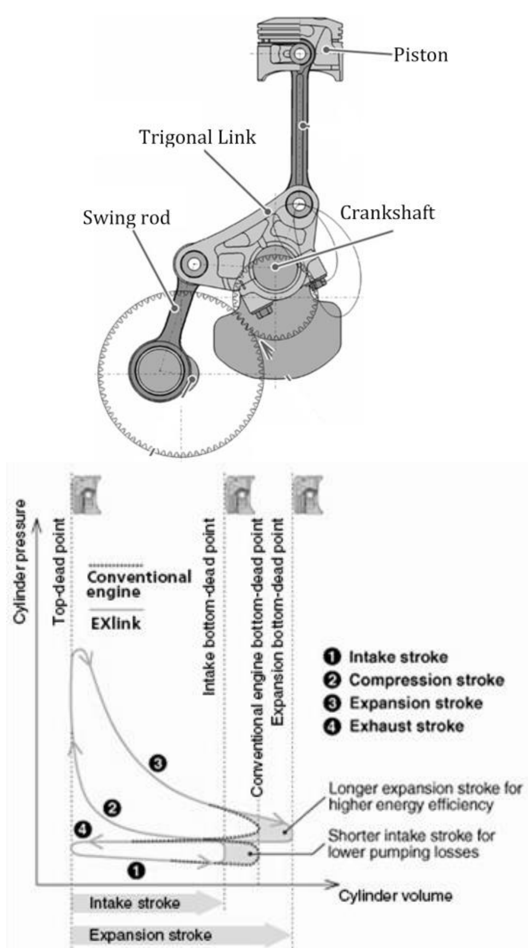

Fig. 1. Honda EXLink (Extended expansion linkage) engine [3]

With the modification made in the valve timing diagram, the partial Atkinson cycle effect (also called Miller Cycle) can be achieved in the cycles of conventional Otto or Diesel cycle engines. The methods applied in the valve timing modification are known as “late intake valve closure" (LIVC) and "early intake valve closure" (EIVC). These methods are often used with "Variable Valve Timing" technology. Today, especially in hybrid vehicles, such extended expansion internal combustion engines are used [13-16].

Another method which is a mechanical system and which is parallel to the design concept described in this paper is the use of planetary gear sets to create an extended expansion engine cycle. In conventional engines, since the movement of the crankshaft and conrod joint is circular, each stroke is the same. In Atkinson cycle engines where the planetary gear set is used, the formation of the strokes differs from conventional engines since the movement of the crank and the conrod joint is similar to a trefoil hypotorchoid or triangle curve (Fig. 2). In addition to reciprocating Atkinson cycle engines, rotary Atkinson cycle engine works are also carried out [3, 17-20].

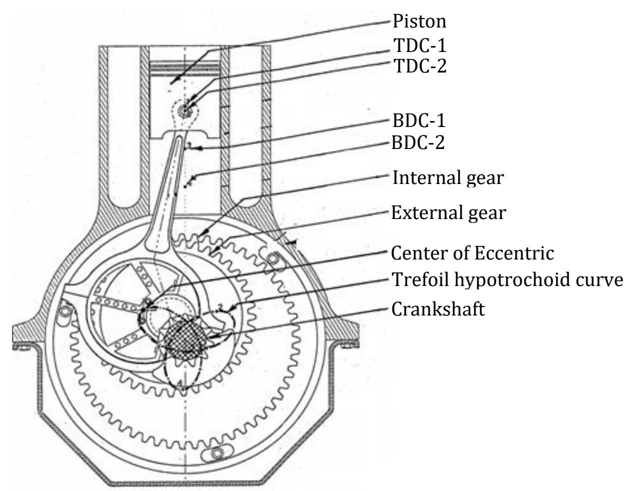

Fig. 2. Atkinson cycle engine using planetary gear system [18]

When the previous studies and papers are examined, many mechanisms have been designed on extended expansion engines. These mechanisms are usually rods or gear systems are used. Today, especially in hybrid vehicles, extended expansion engines with turbocharging are often preferred. In hybrid vehicles where energy efficiency is front-line, extended expansion engines will continue to be used instead of conventional engines. The extended expansion engines used in the literature in the planetary gear system have a semi-planetary structure. In the novel designed extended expansion engine mechanism, conventional planetary gear set structure is used and kinematic analysis was performed on this paper.

2. The Kinematics of the Novel Extended Expansion Engine Mechanism

The designed novel mechanism comes from a simple planetary gear mechanism. The crankshaft is connected to center of the sun gear. The motion is transmitted through the planet gears to the inner ring gear. Unlike conventional planetary gear sets, there are outer ring gears on the outside of the inner ring gear. As a result, the motion of the sun gear is transmitted to the outer ring gear (Fig. 3). There is a central offset (a) between the center of outer ring gear (A) and center of the inner ring gear (B). The movement in each course of the piston is also different because of the offset caused by the outer ring gear.

The ratio of sun gear to inner ring gear speeds refers to the planetary gear set gear ratio (r_{SIR}). At the same time, the r_{SIR} is equal to the number of teeth of the inner ring gear and the sun gear. In a four-stroke engine, r_{SIR} must be 2, since one cycle is completed in 720° crank angle.

$$r_{SIR} = \frac{n_S}{n_{IR}} = \frac{D_{IR}}{D_S} = 2 \qquad (1)$$

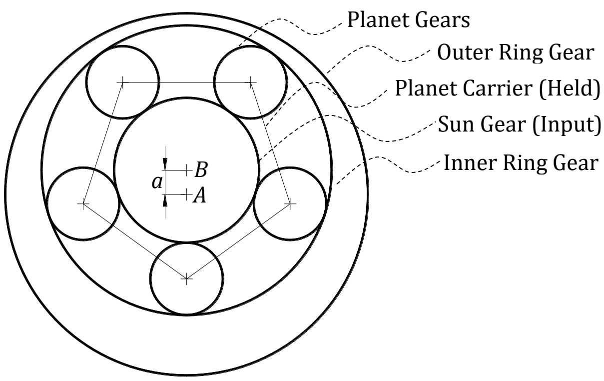

Fig. 3. Designed planetary gear mechanism

The angle at which the central offset (a) line is made by the x-axis when the piston is at the top dead center (TDC) is expressed as γ-angle. This angle directly affects the piston stroke characteristic. The position of the crank-conrod mechanism on the designed planetary gear mechanism is shown in fig. 4.

In planetary gear mechanisms, one element of the mechanism is the input and the other element is the output and the other must be constant (held position) so that different motion rates can be obtained. In this system, the ring gear, the sun gear and the planetary carrier are designated as output, input and held, respectively. The center of the sun gear, where the crankshaft is connected due to the central offset created by the ring gear (between outer and inner ring gear), also causes offset. Depending on the crankshaft motion, the piston movement is also very attractive from the conventional mechanism.

As shown in fig. 5, the crankshaft is connected to the sun gear center. Due to the offset between center of the inner ring gear and the outer ring gear, the center of the sun gear is moving in a specific orbit depending on the motion of the planetary gear set. Assuming that the crankshaft (sun gear) is rotated clockwise (CW), the planet gears move counterclockwise (CCW), causing the orbital gear to move CCW. However, the relationship between the angle of rotation (θ) of the sun gear and the angle of rotation (α) of the internal ring gear can be expressed as follows.

$$\alpha = \frac{\theta}{r_{SIR}} = \frac{\theta}{2} \qquad (2)$$

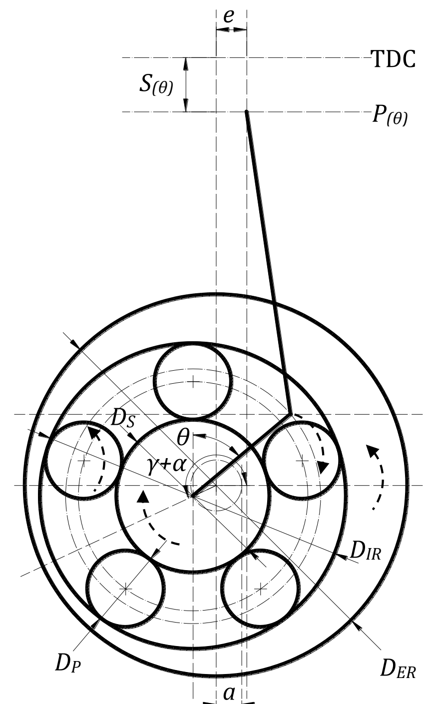

Fig. 4. Position of the crank-conrod mechanism

In order to obtain the motion equations, fig. 4 is simplified and the critical motion points are named as fig. 5.

In fig. 5, point A refers to the center of the outer ring gear and is considered as the reference point for the system. For this reason, the coordinates of point A on the x- and y-axes are assumed to be zero. The central offset (a) between the center of the sun gear (A) and the center of the outer ring gear (B) ensures that this system is unique. Because of the central offset (a), the y-axis position of the point D ($y_{3(\theta)}$) depends on the θ angle of the crankshaft rotation. This is different from the conventional crankshafts. Since there is no such deviation in conventional crankshaft assemblies, point A and point B are the same points. This situation can be expressed as in the following equations.

$$|AB| = a \qquad (3)$$

$$x_{1(\theta)} - x_0 = a.cos(\alpha + \gamma) \qquad (4)$$

$$x_{1(\theta)} = a.cos\left(\frac{\theta}{2} + \gamma\right) \qquad (5)$$

$$y_{1(\theta)} - y_0 = a.sin(\alpha + \gamma) \qquad (6)$$

$$y_{1(\theta)} = a.sin\left(\frac{\theta}{2} + \gamma\right) \qquad (7)$$

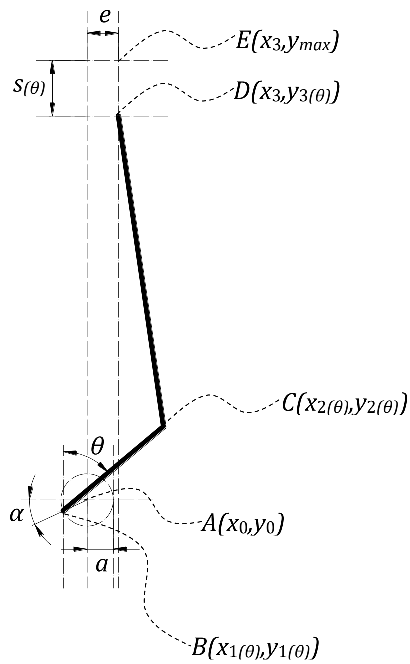

Fig. 5. Nomenclature of critical motion points in crank-conrod mechanism.

The distance between point B and point C is the crank radius (half stroke). The change in position of the point C due to the θ-angle in the conventional crank-pinion mechanism is a complete circular motion. However, in the presented novel mechanism, the position change of the point C is circular, while the change of the position of the point C is trefoil hypotrochoid curve because of the central offset (a). Depending on the angle θ for a system with the offset, the positional change of the point C on the x- and y-axes can be expressed as:

$$|BC| = b \tag{8}$$

$$x_{2(\theta)} - x_{1(\theta)} = b.cos(-\theta + \gamma) \tag{9}$$

$$x_{2(\theta)} = b.cos(-\theta + \gamma) + x_{1(\theta)} \tag{10}$$

$$x_{2(\theta)} = b.cos(-\theta + \gamma) + a.cos\left(\frac{\theta}{2} + \gamma\right) \tag{11}$$

$$y_{2(\theta)} - y_{1(\theta)} = b.sin(-\theta + \gamma) \tag{12}$$

$$y_{2(\theta)} = b.sin(-\theta + \gamma) + y_{1(\theta)} \tag{13}$$

$$y_{2(\theta)} = b.sin(-\theta + \gamma) + a.sin\left(\frac{\theta}{2} + \gamma\right) \tag{14}$$

When the designed system is r_{SIR}=2, when the crank is turned two turns in the CW direction, the outer ring gear rotates CCW one turn. Fig. 6 shows the change of points B and C over two complete revolutions of the crankshaft. In the designed system, the path of the point C at the first crankshaft and the path at the point C at the second crankshaft are different from each other. However, since the central offset a=0 in the conventional crank-conrod mechanism, the path taken by point C is circular.

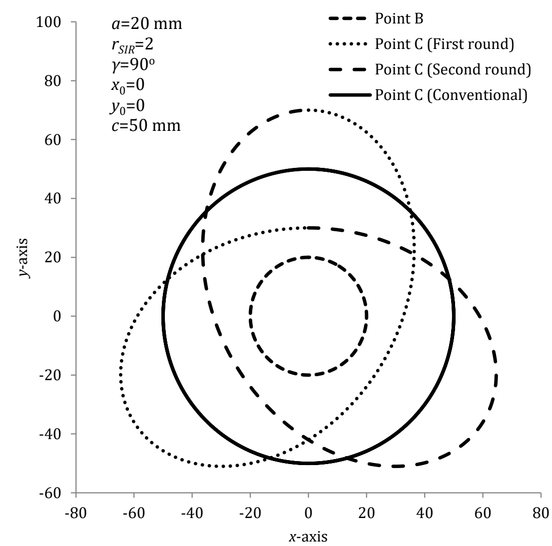

Fig. 6. The trefoil path of point C

The difference between point D and point A on the x-axis is called piston offset (e). The positional change in the x- and y-axes of the point D depending on the angle θ for the a and e for a system can be expressed as:

$$|CD| = c \tag{15}$$

$$e = x_3 - x_0 \tag{16}$$

$$x_3 = e \tag{17}$$

$$\left(x_3 - x_{2(\theta)}\right)^2 + \left(y_{3(\theta)} - y_{2(\theta)}\right)^2 = c^2 \tag{18}$$

$$y_{3(\theta)} = \left(c^2 - \left(x_3 - x_{2(\theta)}\right)^2\right)^{0,5} + y_{2(\theta)} \tag{19}$$

$$y_{3(\theta)} = \left(c^2 - \left(e + x_0 - b.cos(-\theta + \gamma) - a.cos\left(\frac{\theta}{r_{SIR}} + \gamma\right) - x_0\right)^2\right)^{0,5} + b.sin(-\theta + \gamma) + a.sin\left(\frac{\theta}{r_{SIR}} + \gamma\right) + y_0 \tag{20}$$

Here $y_{3(\theta)}$ can also be expressed as the piston path ($s_{(\theta)}$), which is defined as the instantaneous distance to the top dead center (TDC) of the piston.

$$s_{(\theta)} = y_{max} - y_{3(\theta)} \tag{21}$$

$$s_{(\theta)} = \Bigg| y_{max} - \Bigg(c^2 - \Big(e + x_0 - b.\cos(-\theta+\gamma) - a.\cos\left(\frac{\theta}{r_{SIR}}+\gamma\right) - x_0 \Big)^2 \Bigg)^{0,5} - b.\sin(-\theta+\gamma) - a.\sin\left(\frac{\theta}{r_{SIR}}+\gamma\right) - y_0 \Bigg| \tag{22}$$

The piston path characteristic is varied by the effects of a, b, c, e, γ and r_{SIR} constants. In the conventional system, a=0, r_{SIR} =1 and γ=π. In the novel mechanism, a>0, r_{SIR} = 2 and $\gamma \geq 0$. Here, the value of a affects the variable piston path, while the value of γ completely affects piston movement characteristic (See Fig 7 and Fig 8.).

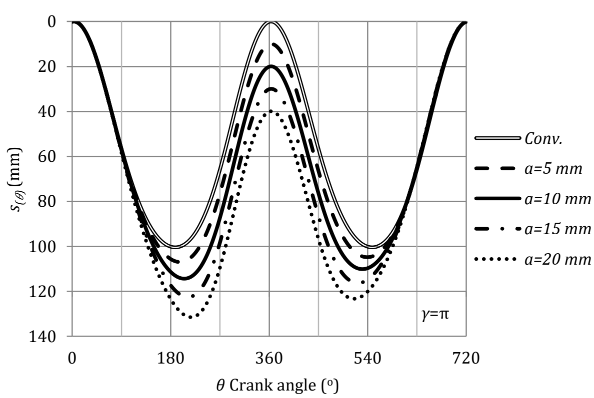

Fig. 7. Effect of central offset (a) on piston path

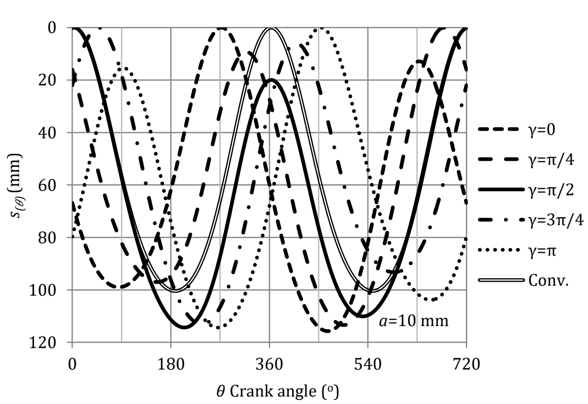

Fig. 8. Effect of γ angle on piston movement characteristics

The piston path ($s_{(\theta)}$) can be expressed more simply by using the following formulas and assumptions.

$$s_{(\theta)} = \xi_{(\theta)} + \psi_{(\theta)} + \varphi_{(\theta)} - y_0 + y_{max} \tag{23}$$

$$\xi_{(\theta)} = -a\left(\sin\left(\frac{\theta}{r_{YG}}+\gamma\right)\right) \tag{24}$$

$$\psi_{(\theta)} = -\left(c^2 - \left(-a\left(\cos\left(\frac{\theta}{r_{YG}}+\gamma\right)\right) - b(\cos(\theta-\gamma)) + e\right)^2\right)^{0.5} \tag{25}$$

$$\varphi_{(\theta)} = b(\sin(\theta-\gamma)) \tag{26}$$

The piston speed is expressed as a derivative of piston path with respect to the angle θ.

$$v_{(\theta)} = \frac{ds_{(\theta)}}{d\theta} = \frac{d\xi_{(\theta)}}{d\theta} + \frac{d\psi_{(\theta)}}{d\theta} + \frac{d\varphi_{(\theta)}}{d\theta} \tag{27}$$

The piston acceleration is expressed as the derivative of the piston speed or second derivative of piston path with respect to the angle θ.

$$a_{(\theta)} = \frac{dv_{(\theta)}}{d\theta} = \frac{d^2\xi_{(\theta)}}{d\theta^2} + \frac{d^2\psi_{(\theta)}}{d\theta^2} + \frac{d^2\varphi_{(\theta)}}{d\theta^2} \tag{28}$$

The piston acceleration is expressed as the derivative of the piston acceleration or second derivative of piston speed or third derivative of piston path with respect to the angle θ. Jerk, also known as jolt, surge, or lurch, is the rate of change of acceleration. jerk can also be expressed in standard gravity per second (g/s). Jerk is not widely used in conventional crank-conrod mechanisms. However, in cam profile designs of internal combustion engines, the jerk is often used. Since the movement characteristics in the novel mechanism is quite different from the conventional crank-conrod mechanism, the jerk is an important indicator for the analysis of the design.

$$j_{(\theta)} = \frac{da_{(\theta)}}{d\theta} = \frac{d^3\xi_{(\theta)}}{d\theta^3} + \frac{d^3\psi_{(\theta)}}{d\theta^3} + \frac{d^3\varphi_{(\theta)}}{d\theta^3} \tag{29}$$

The reason why the crank angle (θ) is used instead of the time expression in all the presented equations is that the angular velocity is assumed to be constant. As is known, at constant angular velocity, the crank angle is a function of time.

3. Numerical Simulation and Results

For numerical simulation, the dimensional values of a single-cylinder conventional internal combustion engine are used, and the kinematic characteristics of this conventional engine and the novel designed extended expansion mechanism are comparatively investigated.

For both cases the compression ratios (ε), in other words the compression stroke(h_c), were kept constant and the measured values of the novel designed mechanism were determined to be constant.

The stroke length and the connecting rod length of the reference engine with 8.2: 1 compression ratio are 58 mm and 105 mm, respectively. [21].

According to the literature, it is preferred that the expansion stroke for the novel designed mechanism is about 1.5 times the stroke compression stroke. [1, 2, 16, 18].

The constant variables of a, b, c, e, γ and r_{SIR} for the conventional and novel designed mechanism are given in table 1.

Table 1. Constant variables determined to be used in calculations

Constant variables	Conventional	Novel mechanism
a (mm)	0	15
b (mm)	29	37
c (mm)	105	105
e (mm)	0	0
γ (°)	270	270
r_{SIR} (--)	1	2

Using the values in Table1 and Equations 23, 27, 28 and 29 the variation of the kinematic curves of this engine with respect to θ is investigated as follows.

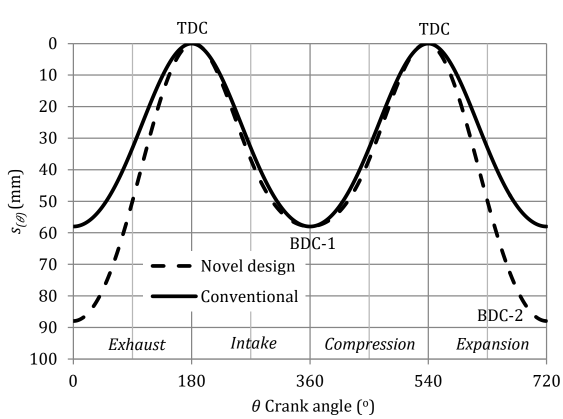

Fig. 9. Change of $s_{(\theta)}$according to θ

The change of $s_{(\theta)}$ with according to θ is obtained as fig. 9. In the conventional mechanism, all strokes are equal to each other. In the novel designed mechanism, the exhaust and expansion strokes are longer than the intake and compression strokes. In both mechanisms, the change of intake and compression strokes according to θ is almost the same as the values given in table 1. There are also two different BDC with the novel designed mechanism.

In the conventional mechanism, compression ratio and expansion ratio are equal to 8.2:1, while in the novel designed mechanism, the compression ratio is 8.2:1 and the expansion ratio is about 11.8:1. Due to the expansion ratio being larger than the compression ratio, this mechanism causes extended expansion.

The change of $v_{(\theta)}$ with according to θ is obtained as fig. 10. As seen in the fig. 10, the maximum piston speed in the novel designed mechanism is higher than the conventional mechanism. These high levels indicate the situation where extended expansion occurs.

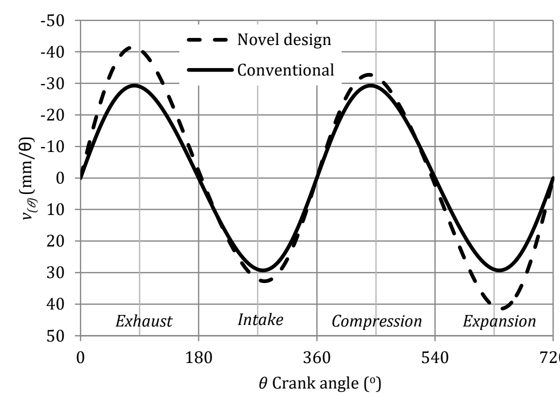

Fig. 10. Change of $v_{(\theta)}$according to θ

To minimize the maximum piston speed, in other words, to obtain values close to those of the conventional mechanism, the extended expansion process can be angularly extended by a further amount (beyond the TDC) by changing the γ value. In this case, of course, all kinematic situations will change.

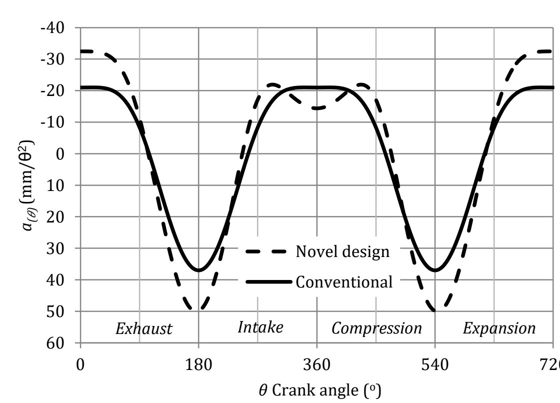

Fig. 11. Change of $a_{(\theta)}$according to θ

When fig. 11 is examined, the change of acceleration according to θ differs from the conventional mechanism because of the motion characteristic of the novel designed mechanism. Although the acceleration characteristic of the novel designed mechanism differs from the conventional mechanism, it is possible to obtain values close to the conventional mechanism by changing the parameters given in table 1.

The change of $j_{(\theta)}$ with according to θ is obtained as fig. 12. The concept of jerk is not a widely used concept in crank-conrod mechanisms. However, when kinematic analysis of a novel designed mechanism is made, the change of acceleration can be very important in determining all design parameters.

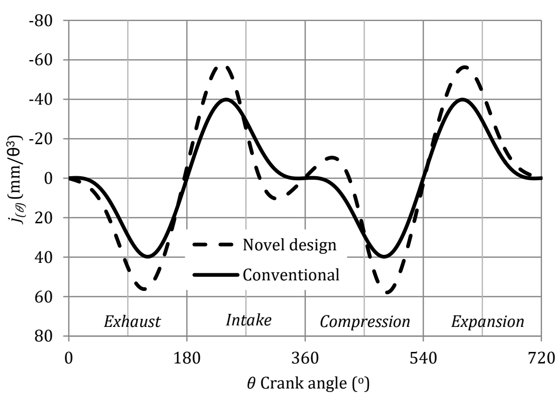

Fig. 12. Change of $j_{(\theta)}$according to θ

Jerk can be used as a marker of mechanical irregularity for such a mechanism. For this reason, this irregularity in the novel designed mechanism is important for the usability of the mechanism to be adequately corrected by changing the values in Table 1.

Figure 13 shows the theoretical PV diagrams for the otto (conventional), EXlink and novel engine mechanisim The volume change is equal for compression and expansion strokes for the conventional mechanism. In the EXlink mechanism, suction stroke is shorter than in the conventional mechanism and the pump loss is reduced in this case. On the contrary, the expansion stroke is more convenient than the conventional mechanism. Advantages and disadvantages of the EXlink mechanism compared to the conventional and Atkinson mechanism have been described previously. The novel engine mechanism is very similar to the PV diagram of the modified Atkinson cycle known in the literature in figure 13. However, this information is not exactly accurate. By varying the design parameters of the novel mechanism as described above (see Figure 7 and Figure 8), a PV diagram can be obtained between the Atkinson mechanism and the EXlink mechanism to achieve different results.

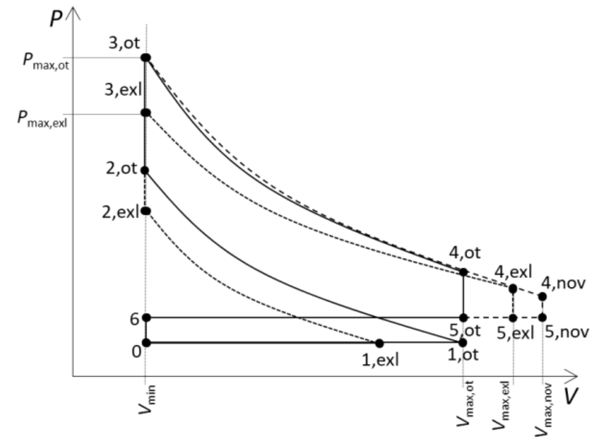

Fig. 13. Comparative PV diagrams

3. Conclusions

There are many extended expansion engine mechanisms in the literature. The use of the planetary gear system and the fact that the extended expansion stroke is provided by the central offset (a) and the piston path characteristic varies by γ makes this system originally designed. By means of the obtained equations, a comparative kinematic analysis was carried out by adopting a sample system as a reference.

For the novel designed mechanism, firstly, a generalized piston path equation is obtained and then this equation is derivatized to obtain velocity, acceleration and jerk equations respectively. Using the obtained equations, a kinematic analysis is made for a system with an extended expansion ratio of 1.5 according to the literature. The absolute maximum speed in the system according to the calculations is higher than in the conventional system. Although this is an undesirable situation, the system can be further improved by changing the design parameters. In this paper, the feasibility and kinematic propensity of the system has been examined.

Based on these results, it is possible to make the optimization of the newly designed system more realistic by making kinematic analysis, air standard cycle or real cycle modeling.

Nomenclature

A	:	Center of outer ring gear
a	:	Central offset
$a_{(\theta)}$	:	Accelaration change depending on θ
B	:	Center of inner ring gear
b	:	Crankshaft half stroke
BDC	:	Botton Dead Center
C	:	joint point of crank-conrod
c	:	lenght of conrod
CCW	:	Counterclockwise
CW	:	Clockwise
EIV	:	Early Intake valve Closure
$j_{(\theta)}$	:	jerk change depending on θ
LIVC	:	Late Intake Valve Closure

r_{SIR} : ratio of inner ring gear to sun gear
$s_{(\theta)}$: Piston path change depending on θ
TDC : Top Dead Center

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