UNCONVENTIONAL MILLER ENGINE

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Abstract: *This paper presents kinematic, dynamic and efficiency characteristics of a new concept of IC engine. The main purpose of this new IC engine concept is to increase engines' thermal efficiency, increased efficiency is achieved through more complete expansion cycle also known as Miller cycle. In this engine design Miller cycle is achieved in a different way than in conventional Miller engine, the movement of the piston is different than in conventional piston mechanisms. Also, this article describes basic parts and shape of this new engine. The results show that with this unconventional thermodynamic cycle engine has higher efficiency than with the standard Otto cycle*

Keywords: Miller cycle, dynamic, kinematic, efficiency, IC engine

INTRODUCTION

Few inventions have had as great an impact on society, the economy, and the environment as the reciprocating internal combustion (IC) engine and the personal transportation culture that it has spawned. Yet for decades, or a century, basic IC-engine design has not changed much. Even modern internal combustion engines convert only one third of the energy of fuel into useful work. The rest of the energy is lost on waste heat, the friction of moving engine parts or to pumping air into and out of the engine. Such low efficiency of today`s internal combustion engine is the consequence of several factors. One of the major reasons for low efficiency of engine is contributed to expansion ratio of engine. Many scientific publication in the field of finite time thermodynamic demonstrate how important is to make longer expansion stroke than compression [1, 2]. Biggest problem here is how to make different lengths of piston stroke in engine mechanism. At the start of the 20th century, engineers tried to achieve strokes of different lengths with complex linkages that were hopelessly impractical. By the use of complex mechanical linkages where the expansion ratio is greater than the compression ratio, Atkinson cycle engine resulting in greater efficiency than with engines using the alternative Otto cycle. Heat gained from burning fuel increases the pressure, thereby forcing the piston to move, expanding the air volume beyond the volume when compression began. Atkinson engine were used in practice for a while and were more efficient than the Otto cycle, but the complicated linkages that enabled this efficiency suffered excessive wear, and reliability problems led to its demise [1].A further step made in this paper is to make analysis of a new engine the heat engine that will be able to provide adequate kinematics to achieve the realization of more complete expansion cycle. In this paper was used thermodynamic model previously presented in [2] to obtaining the improvement between this new cycle and the standard Otto cycle. The results obtained herein include the dynamic, kinematic and efficiency characteristics of irreversible reciprocating new engine cycle which is very similar to Miller cycle. Also in this paper was presented basic description of the new engine that will be able to realize thermodynamic cycle with increased efficiency described in detail in [3].

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CONVENTIONAL AND NEW MILLER CYCLE ENGINE CONCEPT

Miller Cycle is an interesting concept, invented by American Ralph Miller, it changed the long-standing basic principle, Otto cycle. As is well known conventional Otto cycle engines have 4 stages in each cycle - intake, compression, expansion and exhaust. Each of them takes roughly equal time, i.e. stroke lengths are equal. Miller Cycle engine (Fig.1) differs from it by delaying the inlet valves closing well into the compression stroke. In Mazda's Miller Cycle V6 engine, inlet valves close at 47 degrees after BDC (bottom dead center, i.e, the lowest position of piston during a cycle). This equals to 20% of the height of stroke. In other words, during the first 20% of the compression stroke, the intake valves remain opening, thus air flows out without compression, this delaying of inlet valves is presented in Fig. 2. The Miller cycle , has an expansion ratio exceeding its compression ratio. In effect, the compression stroke is two discrete cycles: the initial portion when the intake valve is open and final portion when the intake valve is closed. In new valveless IC engine with more complete expansion of working fluid Miller cycle is achieved in a different way, through kinematics described in Fig. 3 and explained in detail in [3].

Figure 1. Miller engine [4]

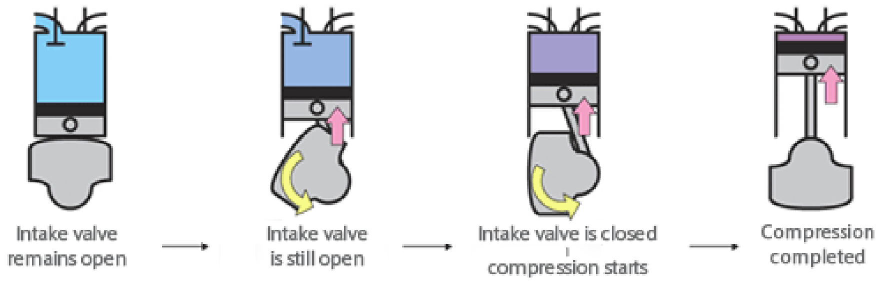


Figure 2. Delayed closure of intake valves during compression stroke [4]

As seen in Fig. 3, in this case to the conventional piston mechanism scheme is added one more movement (rotation) of cylinders around axis at exactly defined position. Precisely defined position of the crankshaft and movable cylinders is very important, because these values (E1 and E2) define more complete expansion of working fluid and full discharge combustion chamber of the residual products of combustion, impact of these values on more complete expansion and compression ratio are described in [5]. Gear ratio between crankshaft and movable cylinders is (-1), in this way piston for one revolution make all four strokes. Described mechanism allows the realization of strokes with different lengths. Basic parts of engine are shown in Fig 3. The engine consist of the lower part of engine block (1) and the upper part of the engine block (2), movable cylinders - rotors (3), in which are placed pistons (4), connected to the crankshaft (5), through the piston rod (6) and piston pin (7). Pistons are placed radially to the crankshaft, whereby the rotation axes of crankshaft located at precisely defined position. Connection between crankshaft and rotors was established through gears (8).

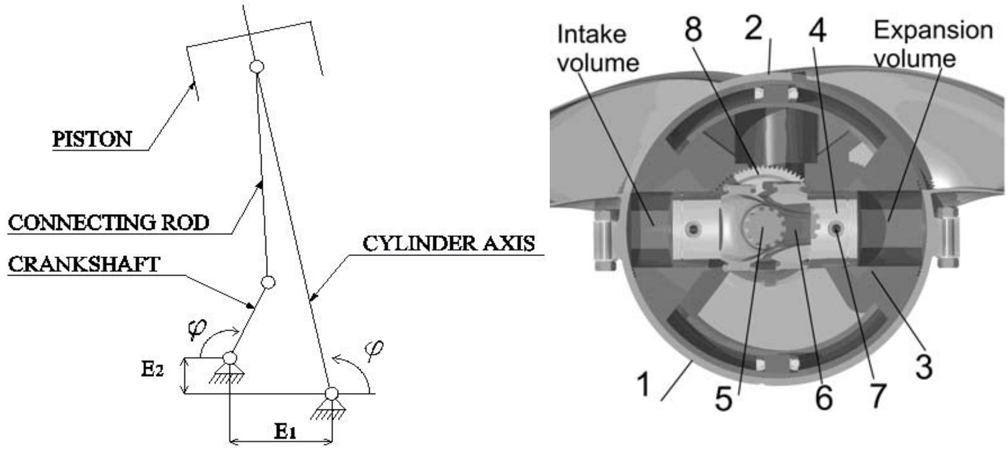


Figure 3. Kinematics scheme and CAD model of the new piston mechanism [5]

Described mechanism allows the realization of strokes with different lengths. Such kinematics would allow different pressure variation as a function of volume. One such example of pressure-volume diagram (PV) is shown in Fig. 4, where it can be seen how with this motion piston can reach much lower value of pressure at the end of expansion. It is also noted that the longest stroke is actually exhaust stroke and the shortest one is compression stroke.

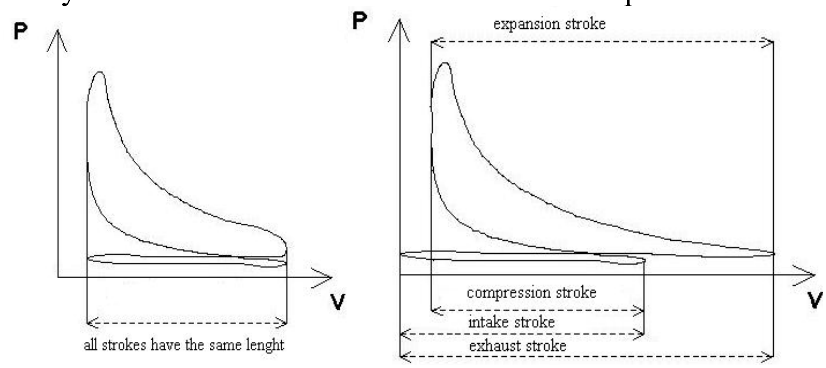


Figure 4. Comparisons of PV diagram in convetional SI engine and new Miller engine [6]

The main difference between Miller`s cycle and the new cycle is closely related to point 6 from the PV diagram in Fig. 5. As can be seen from the PV diagram, volume at the end of exhaust is equal to zero. Certainly, volume with this value is impossible to achieve in real engine, but very close to zero is possible with this internal combustion engine.

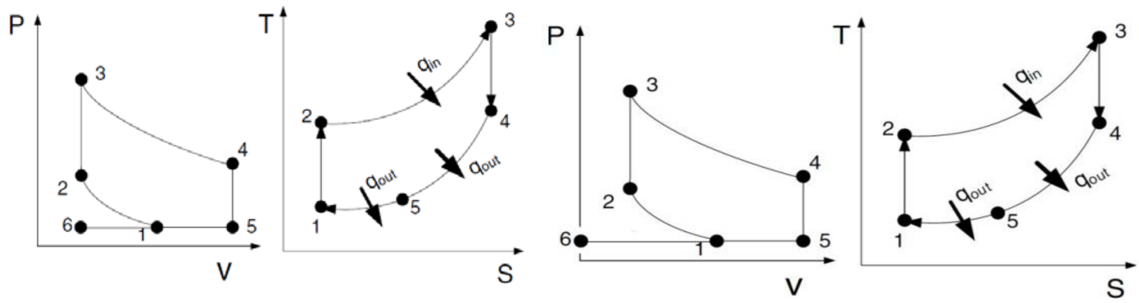


Figure 5. Comparisons of PV and TS diagram in ordinary Miller cycle and new cycle [5]

With this engine design can be easily achieved such motion where piston at the end of exhaust stroke comes very close to engine housing and remove all products of combustion. As it was mentioned, today's internal combustion engine design is not able to provide more complete expansion of working fluid in cylinder. In this new mechanism movement of piston is different than in conventional IC engine. Such unconventional motion has significant impact on pressure volume changes in relation to conventional engine. The described kinematics scheme can be implemented in several ways, in this case was chosen a way which was described in detail in [3]. This paper only briefly present the basic design of radial–rotary valveless IC engine with more complete expansion, cross section and kinematics scheme of this engine can be seen from Fig. 3.
Besides conventional kinematic of piston motion which provides all four strokes with equal lengths, there is one more characteristic of standard IC engine design that is in close relation to earlier disruption of expansion. Exhaust valves open timing is also one of the reasons for lower efficiency of IC engine [7, 8]. As far as the valve lift diagram is concerned, it should be observed that the opening and the closing events cannot be instantaneous, and the gas flow accelerates and decelerates with a certain delay, due to its inertia; therefore, some advances in opening and some delays in closing are required for advantageous operations. In Fig. 6, where the engine working cycle is depicted, in terms of pressure versus displacement, an example is shown of a valve timing solution that features advance and delay times (with respect to the dead centres) for both the intake and the exhaust valves. The process of filling the cylinder with fresh air-fuel mixture can be analysed with the help of Fig. 8, this figure show the charge air speed through the intake valve versus crank angle for different engine speeds with constant valve events [9]. At first, the analysis is limited to the last part of the intake phase, when the movement of the piston, after BDC, is inverted and the air continues to flow into the cylinder because of the inertia phenomenon

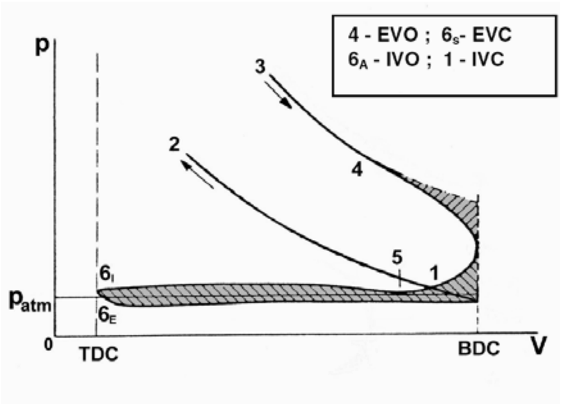


Figure 6. Four-stroke exhaust and intake process. The shaded area indicates work losses related to the Exhaust Valve Opening Advance and to the pumping effect

The air flow towards the cylinder therefore vanishes with a certain crank angle delay with respect to BDC. Moreover, this delay differs, according to the different engine speeds. If the intake valve closing delay exceeds the points where the air speed decreases to zero, a reject phenomenon occurs and the piston expels the air through the intake valve, that is, the air speed increases in the opposite direction. The burnt gas back flow phenomenon, which occurs at the beginning of the intake phase, and the air mass flow rejection phenomenon, which occurs at the end of the intake phase. The main effects of the intake valve closing delay concern: the engine volumetric efficiency, the torque characteristic and the load control of SI engines. Figure 4 also shows the intake valve opening advance and the exhaust valve closing delay (with respect to the TDC), which are responsible for the "valve overlap", that is, the phase when both valves are open at the same time. At the beginning of the intake phase, the valve opening advance pre-arranges the intake valve in an open position when the fresh air (or mixture) begins to flow into the cylinder after top dead centre and reduces the throttling between the valve and its seat. In this first part of the intake stroke, the air is unable to enter the cylinder because of the presence of a residual gas pressure that produces a back-flow into the intake manifold when the exhaust valve is still open. This internal exhaust gas recirculation, during the valve overlap, affects the combustion process and, as a consequence, the pollutant emissions.

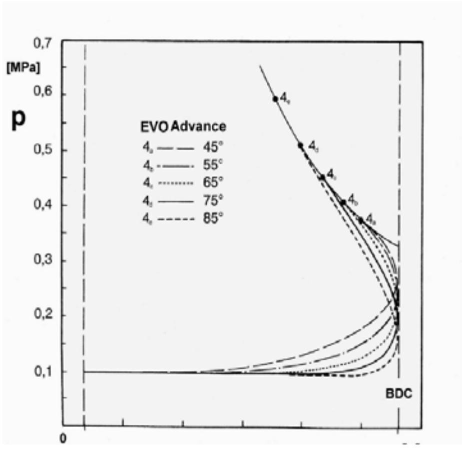


Figure 7.Diagram of the effect of exhaust valve opening advance on a four-stroke engine cycle [9]

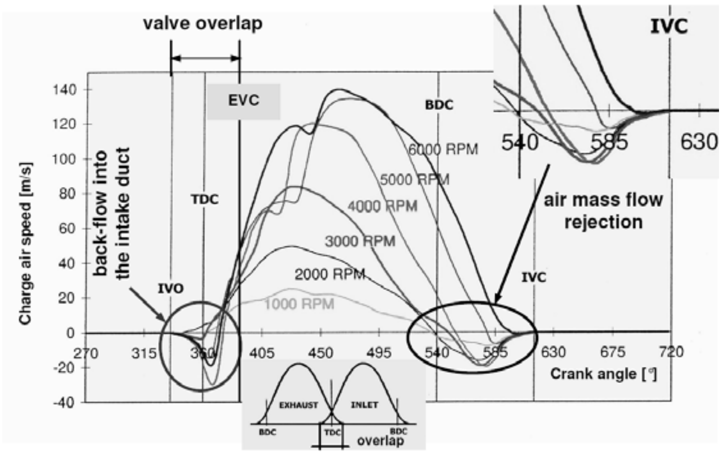


Figure 8. The burnt gas back flow phenomenon [9]

As far as the discharge of burnt gas is concerned, Fig.7 shows the effect of different exhaust valve opening advances on the indicated cycle work and the related pumping losses [9]. An optimum trade-off can be searched for, between the lost and recovered cycle work, due to the employed opening advance. Therefore in ordinary spark ignition engines there is inevitable loss of pressure at the end of expansion. This pressure lost lead to decreasing efficiency. In the described concept flow of fresh mixture and products of combustion was performed in a different way. During working process in this engine there is no possibility for making back-flow of fluid, because this engine is valveless. Flow of working fluid was achieved through intake and exhaust channel which are connected to intake and exhaust manifold (Fig.9). This design of engine housing will provide better flow of working fluids into the cylinders which are placed in engine rotor, besides this feature with this unconventional piston motion and construction of exhaust channel it is easy to avoid earlier leaving of gas under the pressure, thus will enable increased efficiency.

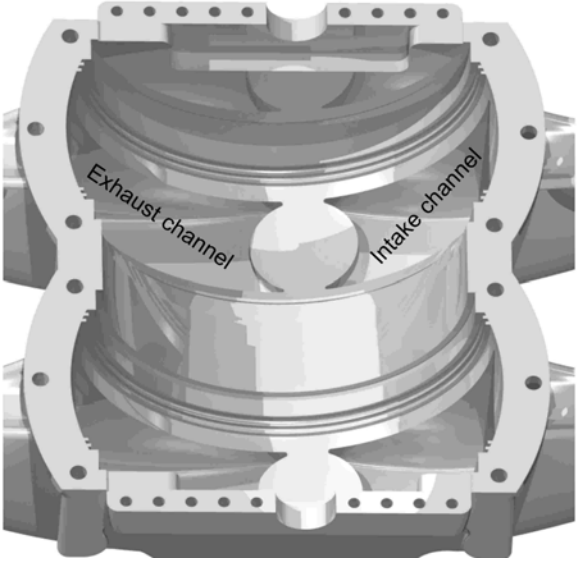


Figure 9. Upper part of engine housing with intake and exhaust channel

KINEMATIC, DYNAMIC AND EFFICIENCY ANALYSIS

Analysis of thermal efficiency

Thermodynamic analysis was calculated with the model previously presented in [2]. Efficiency was examined through eq. (1-3) where eq. (4) is used for solving efficiency values of this new thermodynamic cycle.

$$Q_{in} = M\int_{T_2}^{T_3} C_v dT = M\int_{T_2}^{T_3}(b + k_1 T)dT = M\left[b(T_3 - T_2) + 0{,}5k_1(T_3^2 - T_2^2)\right] \tag{1}$$

The heat rejected by the working fluid, during the process $4 \rightarrow 5$, is

$$Q_{out1} = M\int_{T_5}^{T_4} C_v dT = M\int_{T_5}^{T_4}(b + k_1 T)dT = M\left[b(T_4 - T_5) + 0{,}5k_1(T_4^2 - T_5^2)\right] \tag{2}$$

The heat rejected by the working fluid, during the process $5 \rightarrow 1$, is

$$Q_{out2} = M\int_{T_5}^{T_1} C_p dT = M\int_{T_5}^{T_1}(a + k_1 T)dT = M\left[a(T_5 - T_1) + 0{,}5k_1(T_5^2 - T_1^2)\right] \tag{3}$$

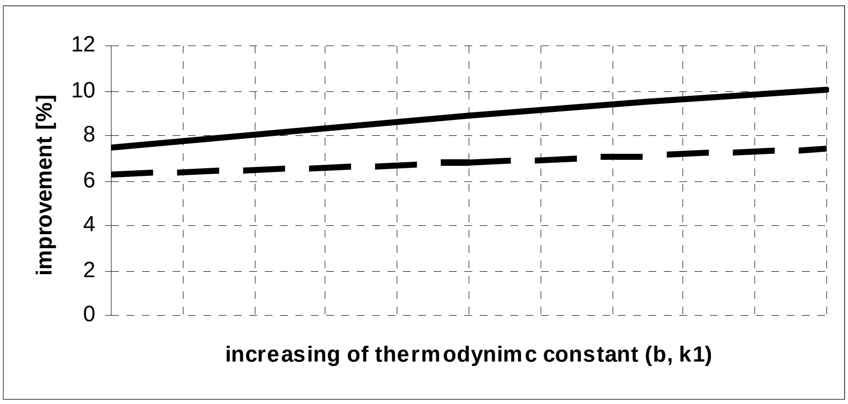


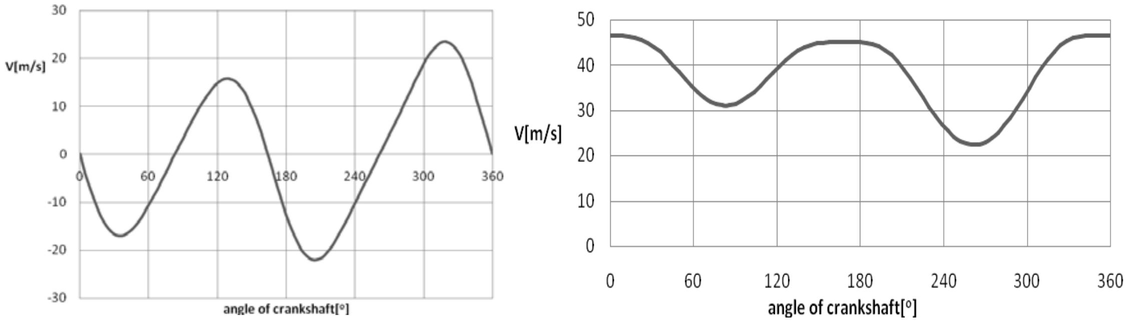
Figure 10. Improvement of efficiency in relation to Otto cycle

$$\eta_t = 1 - \frac{M\int_{T_5}^{T_4} C_V dT + M\int_{T_5}^{T1} C_P dT}{M\int_{T_2}^{T_3} C_V dT} \tag{4}$$

From the numerical results, it can be concluded that it is of great importance to realize complete expansion of working fluid in engine cylinder. In Fig. 10 are described how values of thermodynamic constants have impact on efficiency in conventional Otto cycle and in proposed new cycle. It is obvious that in any case there is an improvement of efficiency. The thermal efficiency decreases with increases values of k1 and b, but with the increase of these values improving efficiency in relation to the standard Otto cycle also increases, as well as can be seen from Fig. 14. In this case was performed calculation for the expansion ratio of 15, and compression ratio of 10.

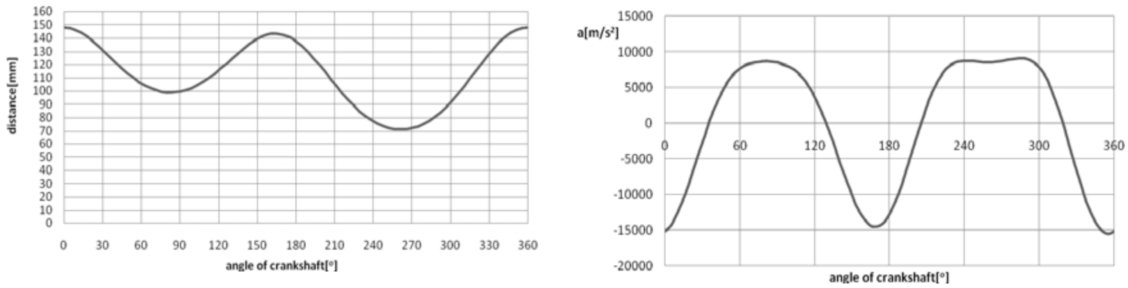
Kinematic analysis

This design of IC engine allows development of all four cycle for one revolution of crankshaft. For these reasons, this engine can achieve the same number of working cycles as conventional IC engine with the half angular velocity of crankshaft. Therefore in the further analysis was used angular velocity of crankshaft of 3000 rpm, because this values of angular speed corresponding to 6000 rpm in conventional IC engine, which is near the maximum for standard engines. The graphic in Fig. 11a) represents the change in velocity of piston through the cylinder. As can be see from the same Fig. 11a) piston velocity law is very similar to piston velocity in standard engine. However, noted is that the amplitude for some strokes is not the same, which is direct consequence of non-conventional kinematics, in other words the tendency to make a longer expansion than compression stroke. In the next diagram on the Fig.11b) change of absolute velocity of piston is shown. Here can be observed the greatest differences between conventional and non-conventional kinematics. From the Fig. 11 b) it can be noticed that the velocity of piston is never equal to zero, in other words, piston in this engine never stops, as is well known in standard IC engine piston need four times per whole cycle to stop. Also velocity is nearly constant in a large part of movement. Finally, Fig. 12b) represents the relative acceleration of the piston. This result is especially interesting, because he shows how the engine construction achieves more complete expansion. As is well know shape of the acceleration curve depends on the kinematics factor, i.e. the number of maximum values of acceleration function depends of ratio R/L (crankshaft radius/connecting rod length). In this mechanism for the first part of movement acceleration curve have one maximum values and for the second part have two maximum, this can be concluded by observing Fig. 12b), in fact this feature achieve more complete expansion, which affect on the piston motion law presented in detail in Fig. 12a). After solving all kinematics equations it is easy to find values of piston movement, speed and acceleration, one such example for angular velocity of 3000 rpm is represented on the previously described Fig. 11 and Fig. 12.



a) b)

Figure 11. Relative (a) and apsolute piston speed (b) for the selected case of 3000 [rpm]



a) b)

Figure 12. Piston path (a) and relative acceleration of the piston (b)

Dynamic analysis

In this section is presented dynamic analysis of valveless IC engine with more complete expansion. Input parameters for solving forces on characteristics joints and engine torque are pressure in cylinder and mass of all movable engine elements. In relation to standard engine in this concept P-a diagram is different for the observed angle of crankshaft. Reason for this is contributed to the fact that all four cycle is done by only one revolution.With the information of calculated pressure changes during rotating of crankshaft it is easy to perform all necessary force data on different joints of engine elements. First it will be done for piston-connecting rod joint. Analysis of forces on the piston-connecting rod joint observed in a coordinate system that is stationary in relation to the piston is given in Fig. 13. On the left side of the same Fig. 13 is sketched how is chosen coordinate system. The forces are given on the right side of the Fig.13. Presented results are relating for the case when the crankshaft rotates with 3000 r/min, and when mass of the piston group and connecting rod are 0.889 and 0.5 kg respectively.

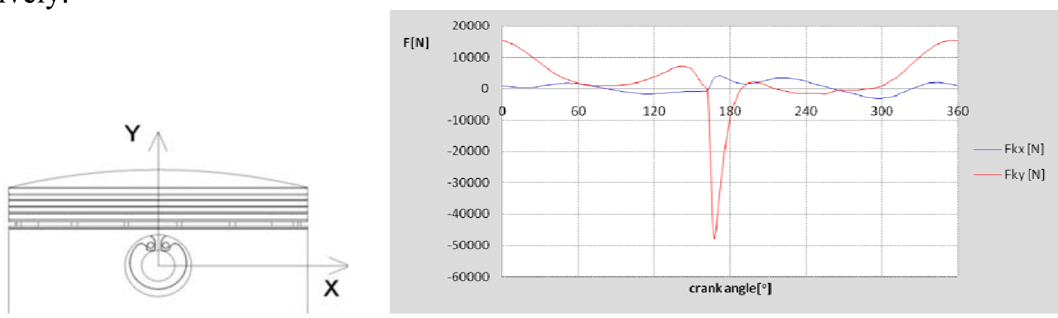


Figure 13. Resulting force in relation to crank angle on piston pin during engine operation at 3000 rpm

As can be seen form the same picture forces on the piston pin are very similar to the forces in conventional piston mechanism, in this case maximum force on piston pin is about 47079 [N]. Similar analysis can be performed for the next assembly. Analysis of forces on the connecting rod observed in coordinate system which is shown on the Fig. 14 (left) is given on the right side of the Fig 14.

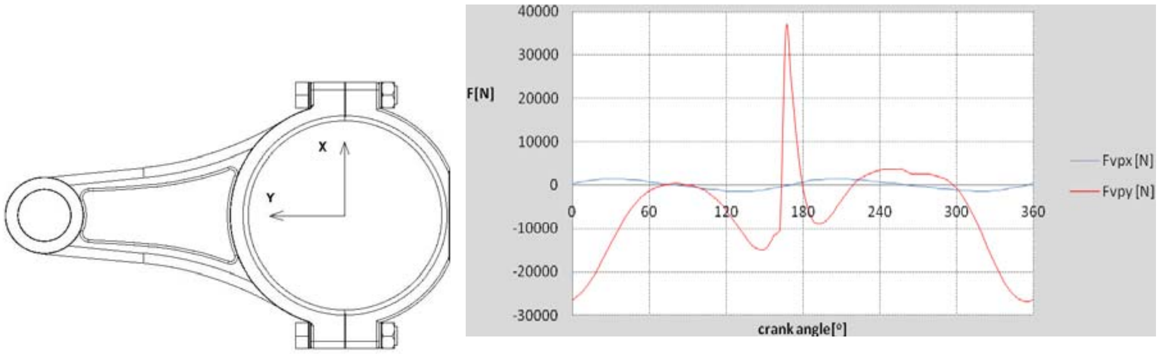


Figure 14. Resulting force in relation to crank angle on connecting rod during engine operation at 3000 rpm

Besides this coordinate system, same analysis can be presented in a different coordinate system, where in this case analysis will be give a more favorable picture of dynamic characteristics for further analysis of torque. As discussed above the same forces can be represented in a coordinate system of crank pin which is presented in Fig. 15. This coordinate system is of great importance because in this way it is easy to separate total force on crank pin into two different force components. This two force components are well known, and it is called radial and tangential force. Where the tangential component is used to determine the torque value on the crankshaft. In Fig. 15 are described values of these forces in relation to crank angle.

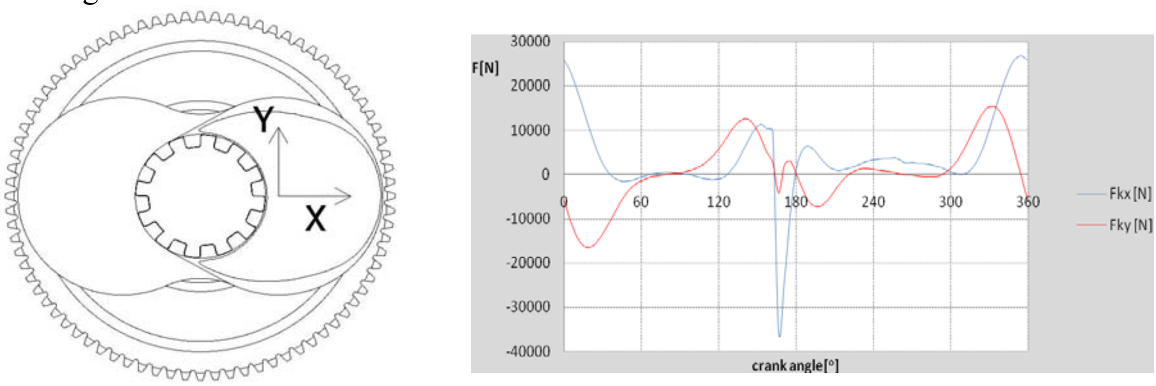
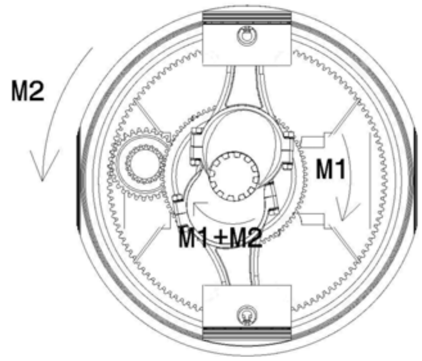


Figure 15.Resulting forces in relation to crank angle during engine operation at 3000 rpm

In this case was achieved maximum force on crank pin of 36202 [N]. Total engine torque in this concept will depend on the sum of two torque components. The first component of torque will be generated on the crankshaft due the effect of tangential force, exactly as it goes in conventional engines. But, the other torque component is consequence of existence of normal force on the cylinder walls. The total torque is obtained by adding torque on the crankshaft due the tangential force (M1) and torque due the normal force on cylinder wall of the rotor (M2), this addition is shown in a Fig 16. The resultant torque on the same graph is the value generated by a single piston-cylinder assembly. To gain insight into the changes of the torque values of the whole engine it is necessary to gather momentum as many times as there are number of piston, with respect to firing order.



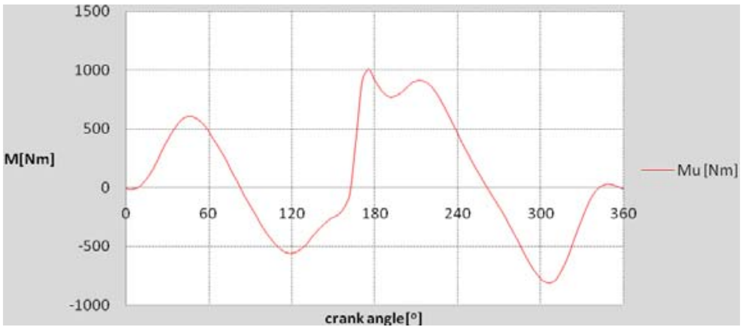


Figure 16. Engine torque due working process for one piston cylinder assembly

Average torque of a single cylinder is equal to the sum of M1 and M2, presented with eq. (8):

$$M_A^C = M_1 + M_2 = 15,95 + 59,68 = 75,53\left[Nm\right] \quad (5)$$

Finally it is easy to calculate total engine torque with two rotors and four cylinders described in detail on Fig. 17, and the changes of engine torque in relation to crank angle is presented through Fig. 18.

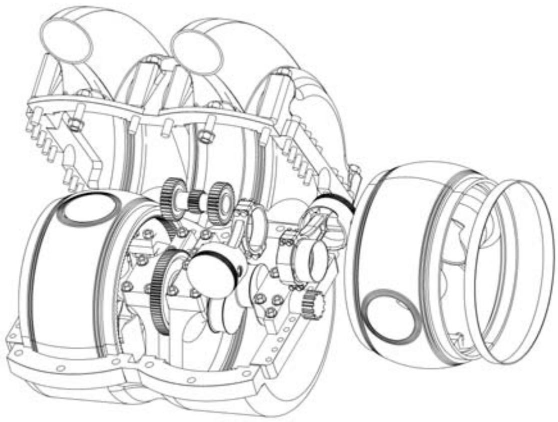


Figure 17. Basic parts of valveless IC engine with more complete expansion

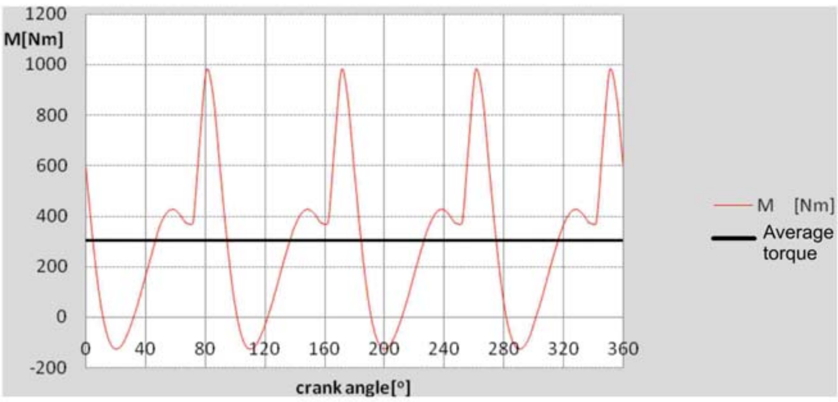


Figure 18. Total torque of the engine with two rotors and four cylinders

Average value of engine torque is evaluated due equation (6):

$$M_A^C = M_1 + M_2 = 15,95 + 59,68 = 75,53\left[Nm\right] \quad (6)$$

CONCLUSION

In this paper was presented one approach for modeling new concept of internal combustion engine. Main aim of this engine is realisation of Miller cycle, in other words realisation of more complete expansion of working fluid. Adaptation of this engine in the conventional vehicles are simple, described engine have dimensions and weight that are even smaller than the ordnary engines of the same power. Only difference is contributed to the engine speed. Presented engine mechanism have half of the engine speed of conventional IC engine for the same number of cycles, this practical means that reduction in gearbox would be smaller, and torque on the input shaft of the gearbox would be higher. In relation to conventional four-stroke IC engine where compression stroke and expansion stroke are symmetrical, in this unconventional concept of IC engine lengths of every stroke are different. Results of thermodynamic analysis show that this new cycle is more efficient than Otto cycle. The results obtained in the present study are of importance to provide a good guidance for the performance evaluation and improvement of practical Miller engines. Future studies should discuss the possible effects of mechanical losses of this concept, in order to achieve values not only thermodynamic efficiency but also mechanical efficiency of IC engines with more complete expansion accurately represented in [3,5,6,10], besides this for future studies is important to give a solution for solving problem of cooling, lubricating and sealing in this new concept.

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