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PROBLEMS WITH REPRESENTATION OF THE OIL FILM GENERATING CONDITIONS ON THE WANKEL ENGINE CYLINDER SLIDING SURFACE

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Abstract

The fundamental problem with wide application of the Wankel engine is to assure oil film continuity on trochoidal cylinder bearing surface. For the sake of considerable difference between curvature radius of the apex seal sliding surface and the trochoidal cylinder the oil film can be generated on the short fragment of the apex seal. However there is certain field of maneuver in the apex seal shape determination which allows to approach close radiuses of curvature of both elements in the areas where the highest gas forces load is occurred. In the paper simulation of the revised apex seal oil film parameters are presented. Author described also the test stand which is going to be used for experimental verification of the simulation results. Conclusions refer to possibility of replacement the constant radius shape of the apex seal sliding surface with shape that consist of two different curvatures.

Keywords: *the Wankel engine, oil film parameters*

1. Introduction

Problems with the apex seal of the Wankel engine has been known since the first engine of that type was designed. The first apex seal designs where just to make start of the engine possible, but in the later engine development apex seal was limiting the proper working time of the engine. In early prototypes usually after just few hour of working the blow-by between adjacent working chambers were enough to stop the engine. Later rotary engine constructions which were applied in serial produced NSU Ro80 cars or even the Mazda Renesis engine apex seal was not the main disadvantage but still it is less effective, less durable and generates more friction losses than modern piston rings in conventional reciprocating engines. Previously described difficulties can be easily explained if the apex seal angle of attack is taken into consideration. The angle between the apex seal axis and the normal to the trochoidal cylinder liner varies within limit of one radian while in conventional piston engine analogous angle do not exceed 0,001 rad. As a consequence in conventional engine the maximum gap thickness between piston ring and cylinder liner do not cut across few micrometers whereas in the Wankel engine gap reach size of several hundred micrometers. There is a question if the apex seal sections with circular or parabolic profile, that are being used nowadays, provide the best results both of the oil film thickness and friction losses caused by apex seal sliding on the trochoidal cylinder surface. As it is pointed out in present elaboration it is possible to propose the apex seal sliding surface shapes which guarantee acceptable oil film parameters when the apex seal is situated perpendicularly to the trochoid surface. What is more the oil film parameters can not get extremely worsen during inclination of the apex seal, mainly in areas where the long and short axis of the trochoid are evenly distant. When looking for analogy in functions of the apex seal and the piston ring it can be assumed that in the reciprocating engine the piston ring could cooperate with cylinder liner in similarly hard conditions as apex seal of the Wankel engine if the cylinder liner would change its diameter by few millimeter. The generating line of this cylinder liner would be sinusoidal shaped with amplitude of few millimeters and period comparable to piston stroke. This kind of model was engaged in computer simulations during the new profile of the apex seal examinations.

2. Parameters of traditional cylinder liner that creates conditions of the apex seal [1]

On the figure 1 changes course of the angle between tangent to the piston ring and tangent to the cylinder liner which is deformed in such manner that generating line of the cylinder liner has sinusoidal shape with amplitude of 1,6mm and period of 160mm.

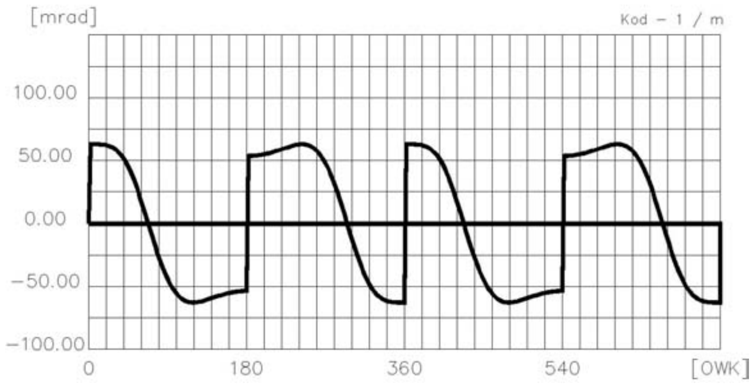

Fig. 1. The angle course changes of tangent to the cylinder liner generating line, the amplitude is 1,6mm and the period is 160mm

The tangent line to the sliding surface of the classical piston ring forms with piston axis lesser angle than angles shown on fig. 1. As a consequence the piston ring contacts with cylinder liner in predominant part of the working cycle alternately with its upper and lower edge and there is no possibility to generate continuous oil film between the piston ring and the cylinder liner. This situation is very similar to that one which occurs in the Wankel engine, where the apex seal also at considerable part of its path works with its edge. In apex seal that have been used so far the sliding surface was a parabolic shape. With the purpose of improving the oil film parameters the apex seal shape that consist of two parabolas was considered. 50% of the profile keep normal shape and the other part, closer to the edges, is multiplied. As a result the apex seal shape looks as it is shown on upper right corner of fig. 2.

When the corrected sliding surface was used oil film parameters presented on fig. 2 it the part signed as “Pier. Zg.” were obtained. If the next apex seal would not be corrected, so its profile is formed by one parabola, the oil film on the most part of apex seal path does not exist. It can be observed on fig. 2 part that is signed as “ Piersc. 2”. The not corrected profile shape is also presented in the middle part of fig. 2 above the diagrams.

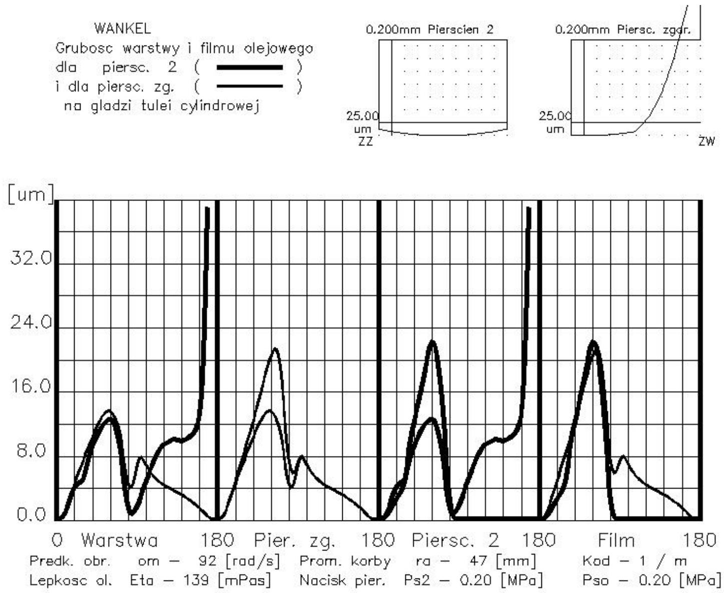

Fig. 2. Comparison of the oil layer thickness course that is left on the cylinder liner surface by two profiles of the apex seal – first part of the diagram; oil film thickness and layer thickness for double parabolic profile – second part of the diagram; oil film thickness course and layer thickness course for the parabolic profile with very high curvature radius – third part of the diagram; comparison of the oil film thickness that was generated by the one and double parabolic profiles – fourth part of the diagram

The sliding surface correction of the apex seal has to be done very carefully because improving oil film parameters in the key areas leads to deterioration of this parameters in other areas. On fig. 3 synthetic course of the oil film parameters generated by two following apex seals of the Wankel engine.

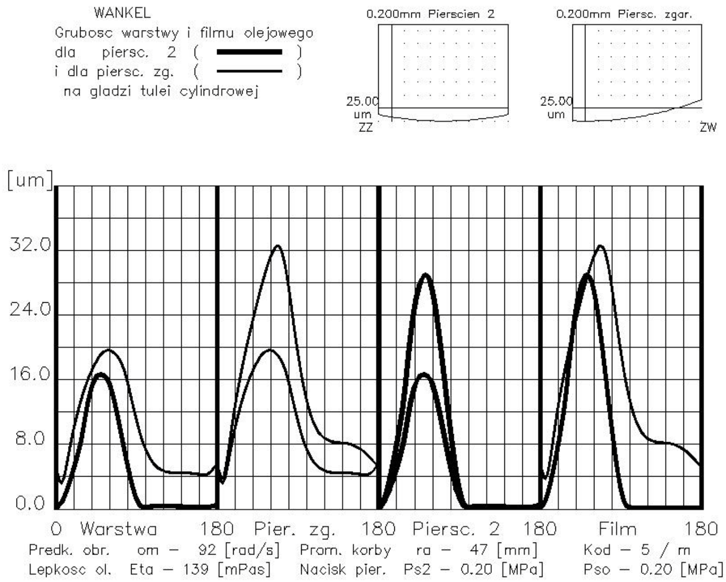

Fig. 3. Comparison of oil film thickness course which is lest on the cylinder surface by two apex seal profiles, for conventional apex seals of the Wankel engine – first part of the diagram; oil film and layer thicknesses for one parabolic profile – second part of the diagram; oil film and layer thicknesses for parabolic with very high curvature

radius – third part of the diagram; comparison of oil film thickness generated by two one parabolic apex seals – fourth part of the diagram

The results obtained from the simulations are classical course of the oil film parameters for types of apex seal that have been used so far. It can be noticed on the first part of the fig. 3 diagram that in the 90 degrees to 180 degrees area the classical one parabolic apex seal leaves no oil layer on the cylinder surface, which means that there is no oil film and intense wear of the trochoidal cylinder surface occurs. However comparing the maximum oil film thicknesses it can be observed that for classical apex seal this parameter is better because it reaches value of 33 μm fig. 3, while for the corrected two parabolic profiles only 25 μm – fig. 2.

The maximum oil film thickness areas for the classical apex seal provide to low friction forces values. It is possible that mechanical efficiency of the Wankel engine with regular apex seals is higher than with corrected two parabolic solutions. Simulations that have been used do not allow to unequivocally confirm this assumption. The partially lack of oil film on the trochoid surface needs other ways of friction losses estimation. The best way to estimate friction losses is to use one of the experimental method what is in realization phase with authors cooperation. In continuous oil film case the computer simulations allow to specify friction forces and example of its results is presented on fig. 4.

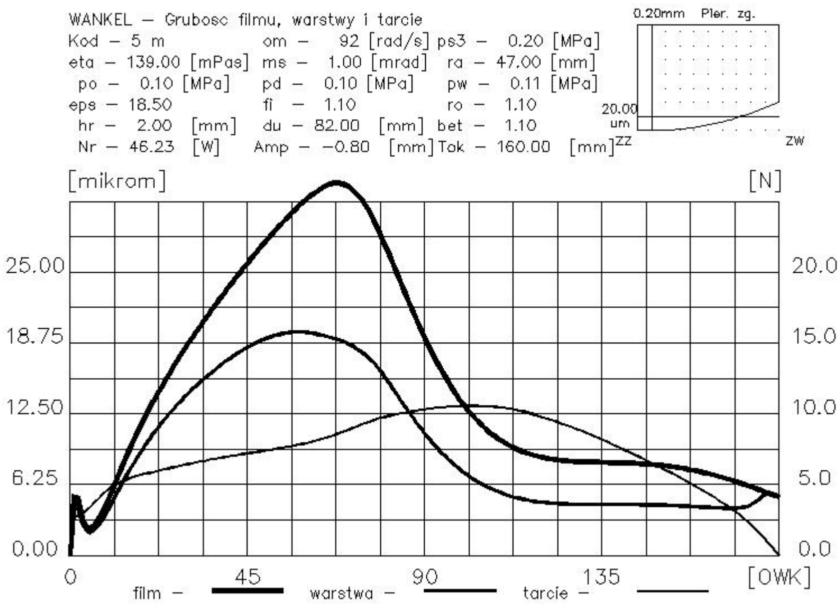

Fig. 4. The oil film and layer thicknesses for one parabolic profile and friction force course for the same apex seal profile

Very often next to basic oil film parameters which are the film thickness, layer thickness and friction force also piston profile filling by the oil film is determined. For analyzed cases profile filling by the oil film is illustrated on fig. 5 and fig. 6. In two parabolic corrected profile case the area where the oil film is generating by the parabola near the edge of slat is evident. It is area of 90 to 180 degrees range – fig. 5. For the typical apex seal with one parabolic profile filling by the oil film in 100 to 150 degrees range is lower so it is probable that the friction force is also lower – fig. 6.

In so far considerations because of its comparative nature the course of angle between tangent to the apex seal and tangent to the trochoid, which is presented on fig. 1, was taken into account. In actual conceptions of the Wankel engine the course of attack angle can be different and it depends on the Z parameter. On figure 7 the angle of attack course for greater value of the Z parameter.

In order to determine the initial oil film parameters which are obtained for matching the trochoid with two parabolic profile and for inclination angles shown on fig. 7 the suitable computer simulations were realized and it results are put together on fig. 8 and fig. 9.

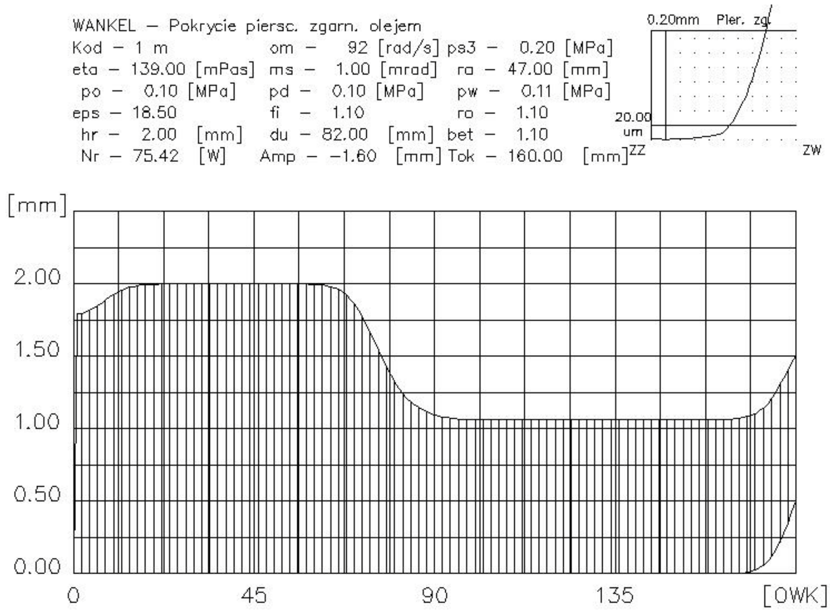

Fig. 5. Course of two parabolic profile filling by the oil film

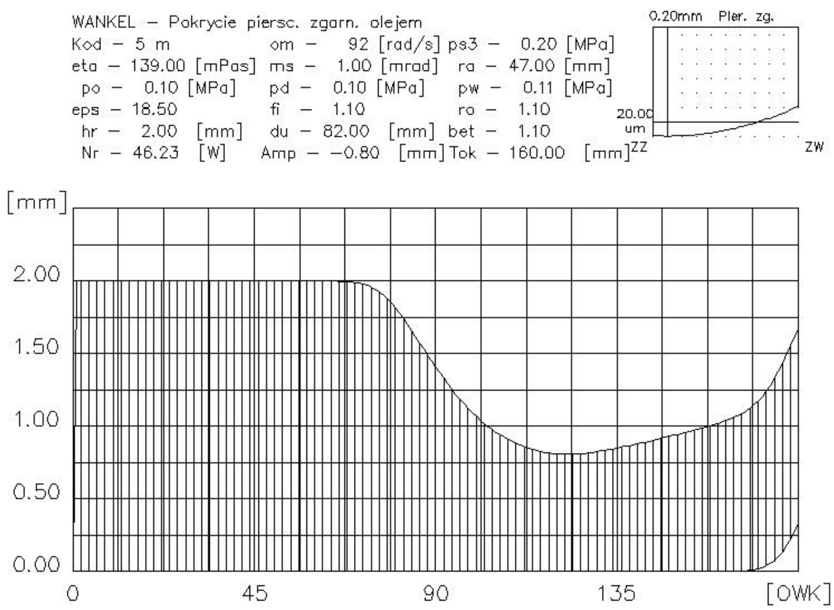

Fig. 6. Course of parabolic profile filling by the oil film, a high curvature radius profile

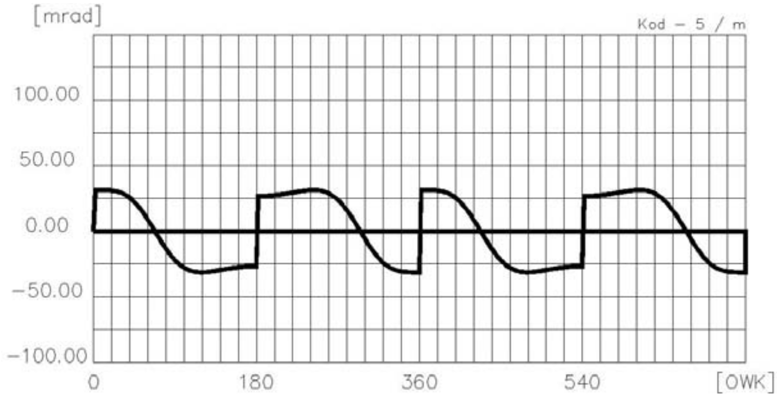

Fig. 7. The course of angle changes of tangent to the cylinder liner generating line, the cylinder liner deformation is sinusoidal with the amplitude of 0,8mm and the period of 160mm

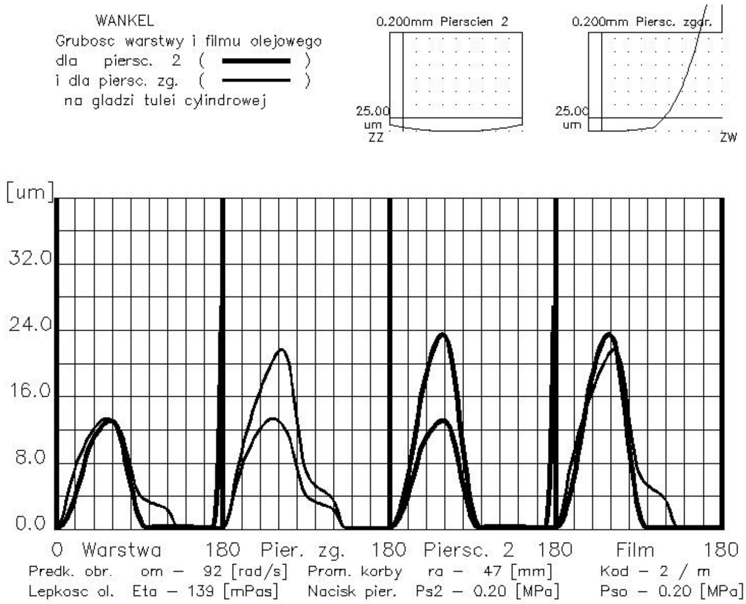

Fig. 8. Comparison of the oil layer thickness course that is left on the cylinder liner surface by two profiles of the Wankel engine apex seal with high value of the Z parameter – first part of the diagram; oil film thickness and layer thickness courses for two parabolic profile – second part of the diagram; oil film thickness course and layer thickness course for the parabolic profile with very high curvature radius – third part of the diagram; comparison of the oil film thickness that was generated by the one and two parabolic profiles – fourth part of the diagram

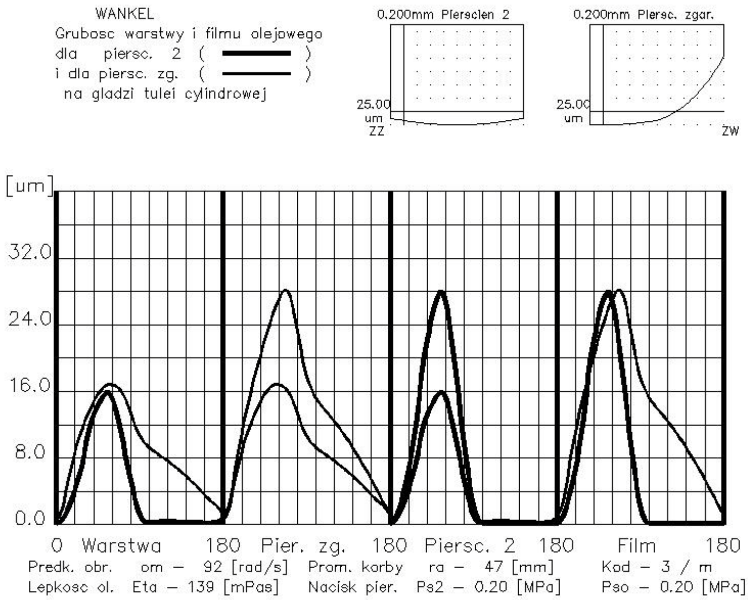

Fig. 9. Comparison of the oil layer thickness course that is left on the cylinder liner surface by two profiles of the Wankel engine apex seal with high value of the Z parameter – first part of the diagram; oil film thickness and layer thickness courses for two parabolic profile – second part of the diagram; oil film thickness course and layer thickness course for the parabolic profile with high curvature radius – third part of the diagram; comparison of the oil film thickness that was generated by the one parabolic profile and two parabolic profile which curvature radius is four times higher than on fig. 8 – fourth part of the diagram

The two parabolic profile was accepted for calculations, smaller curvature radius – fig. 8 and four times higher radius of curvature – fig. 9. It is easily to notice that in second case the results are better. The oil film in second part of the diagram demonstrated on fig. 9 is thicker and what is more important there is no break of the oil film what can be noticed on fig. 8 between 150 and 180 degrees.

3. The test stand used to verification of the computer simulations results

To the investigations authors have adopted the test stand which was formerly used for oil film investigations of piston rings of the marine two stroke engines. Scheme of the test stand is presented on fig. 10 [2].

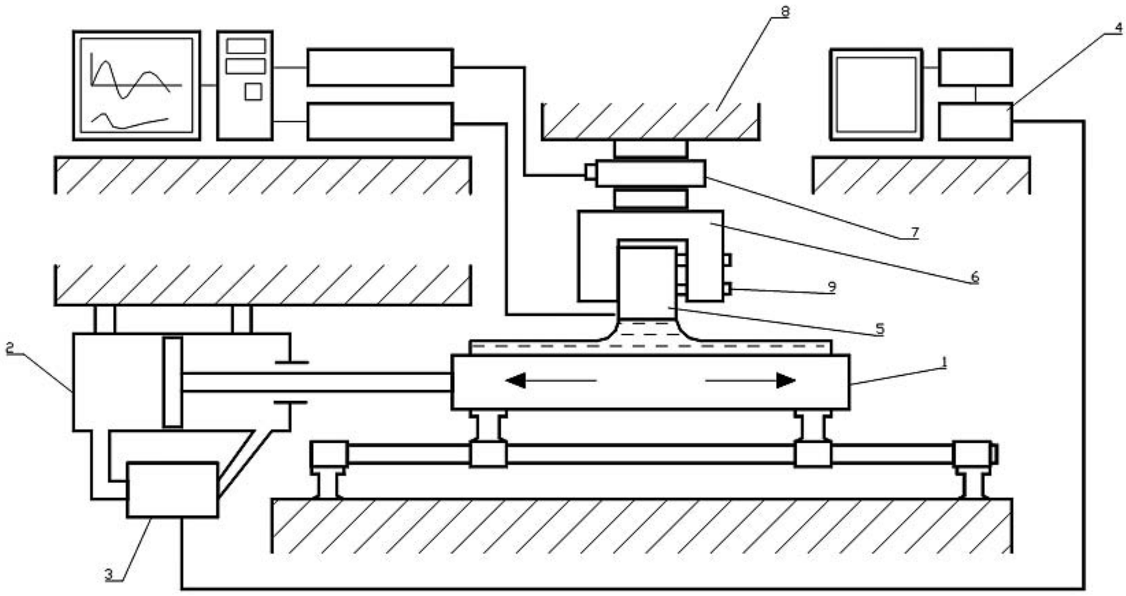

Fig. 10. Schema of construction of the research stand (description in text) [2]

On presented stand reciprocating movement is executed by cart with settled plate simulating cylinder sliding surface 1. This cart is driven by hydraulic servo-motor 2. Pressures in working chambers of servo-motor are regulated by electromagnetic valve 3 which uses the difference of voltage between signals from steering apparatus 4 and sliding resistor which is connected with piston rod. The element that simulates piston ring 5 is seated in piston block model 6 which is settled in the saddle 8 with force transducer 7.

Because in the apex seal motion there are no dead centers and the plate has finite length, the to-and-fro motion had to be substituted by a linear translational motion in the plate length limit. The next step was to reflect the apex seal kinematics was to replace analog steering method by its numerical equivalent. This allowed to specify any plate path and velocity profile, also the apex seal velocity profile. The additional advantage of numerical way of plate motion programming is that we can choose any part of apex seal track on trochoid cylinder liner to be simulated. The plate path determination consists in specifying its position for every of 4096 points. In case of reciprocating engine 4096 point mean full crankshaft turn and for the apex seal motion simulation this number is discrete division of any rotor rotation angle that has been chosen to be investigated. It was said that there are no death centers in the apex seal motion so it was important to make the acceleration and deceleration safe. Additional procedures were added to the main program which are responsible for initial and final phase realization of the plat motion. Initially authors assumed that with total plate length of 800mm only 500mm will be used to the apex seal motion simulation while 100mm will serve to acceleration and deceleration. This results from the necessity of minimization the inertia force that is generated during plate acceleration and deceleration. Actual view of the test stand is presented on fig. 11.

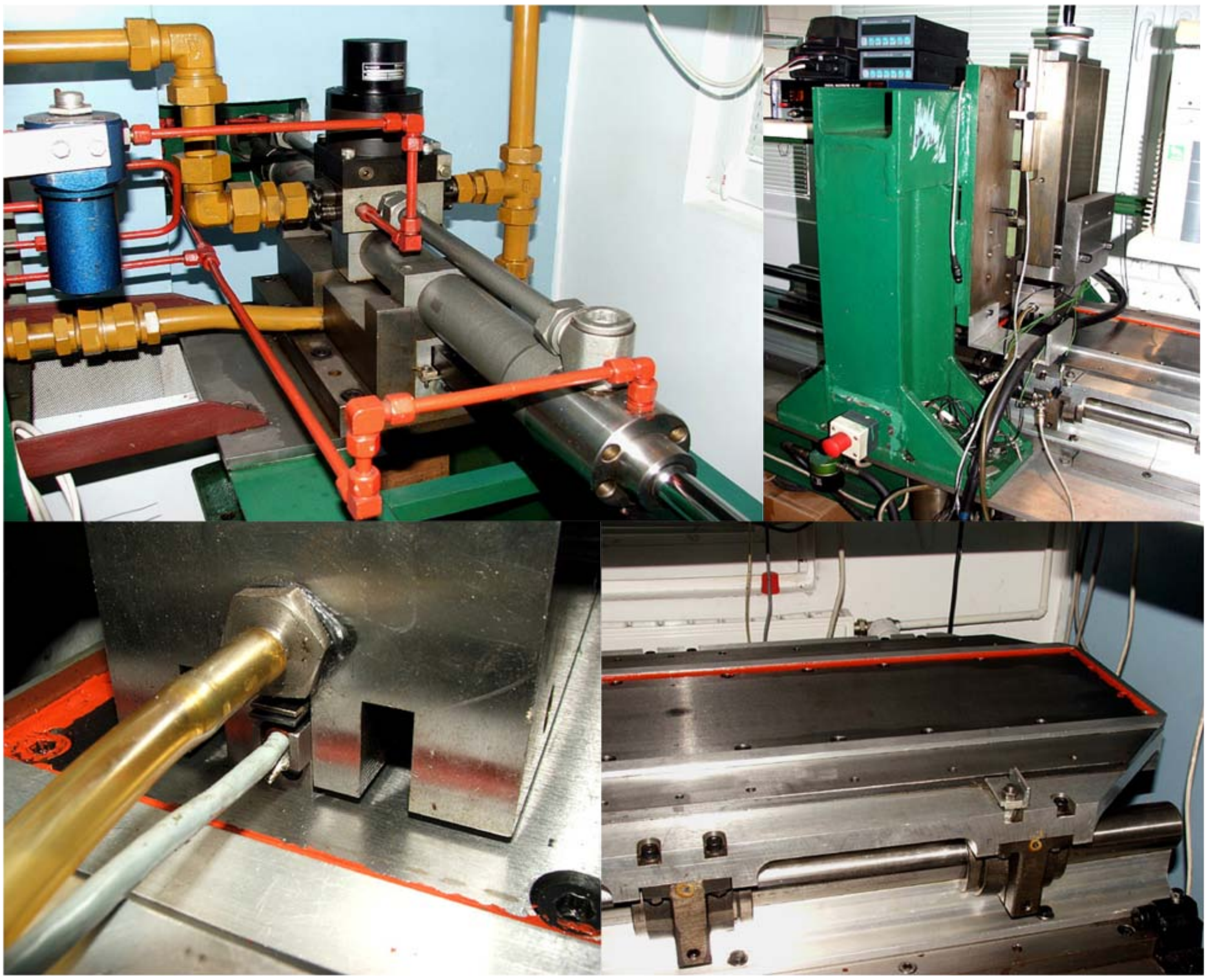

Fig. 11. View of the test stand, upper left corner – hydraulic drive, upper right corner – saddle with piston model attached, lower left corner – piston model with attached slat, lower right corner – the plate that simulates the cylinder liner surface

4. Conclusions

1. One of possible ways to improve the oil film parameters between the apex seal and trochoid sliding surface is to shape the apex seal profile as a two parabolic curve.
2. In most of the apex seal path, where the apex seal axis is perpendicular or almost perpendicular to the trochoid sliding surface, the oil film is generated by high curvature radius parabola while in the vicinity of shorter axis of trochoid the oil film is sustained by parabolic profile with lower radius of curvature.
3. The low radius of curvature profile part should be formed in such way that the convergent gap would not be less than 25% of total gap length between the apex seal and the trochoidal sliding surface.

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