THE OIL FILM PARAMETERS OF THE WANKEL ENGINE APEX SEAL IN ASPECTS OF DURABILITY OF MATING ELEMENTS

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Abstract

The Wankel engine is one of only few alternatives to the reciprocating engines. The advantages such as good value of maximum engine power to its mass ratio are still present and can have great sense in selected fields of application, for example General Aviation. Nevertheless the disadvantages of the Wankel engine design have never lost its importance and still pose an obstacle to wider use of the Wankel engine. One of the main drawbacks is the rotor sealing system design, especially the apex seal where single slat has to fulfil a purpose of conventional engine piston rings pack. Moreover the unfavourable changes of the apex seal angle of attack causes that optimal shape of the apex seal sliding surface, in aspects of the oil film parameters, can not be achieved. These results in worsening of apex seal elements mating conditions, reducing its effectiveness and durability. In the paper authors present results of simulation researches where oil film parameters, for example oil film thickness, which determines apex seal durability, were obtained for different engine operating conditions and various shapes of apex seal sliding surface geometry. The results indicate possible directions during apex seal designing process which should provide oil film continuity in whole rotor angle position range and for wide engine operation conditions range.

Keywords: *combustion engines, Wankel engine, apex seal, oil film parameters*

1. Introduction

In the year 2003 sensational information was that Mazda company introduces new sport car model propelled with latest version of the Wankel engine which thanks to new constructional solutions development can be a turning point in its history. The most important change was moving exhaust port from trochoidal cylinder liner to side housing, very similar to the intake ports. The feature that deserves a special attention is the fact of fulfilling EURO 4 standard. As a consequence of this the new RENESIS engine won the title of „Engine of the Year 2003".

The characteristic feature of the Wankel engine is the way of rotor sealing. Each of three rotor apexes possesses single slat which has to meet all tasks that are also set for whole piston rings packet of the reciprocating engine, for example combustion chamber sealing, oil distribution on cylinder liner at a uniform rate, heat abstraction. This kind of single slat tightening has always caused many problems for engineers. This imperfection of rotor sealing has become the reason of regression in Wankel engine development and nowadays the only company that still continues researches on the rotary engine is Mazda. Unfortunately with new perspectives in the rotary engine development known problems of the apex seal are still present. Authors decided to use basics of the hydrodynamic lubrication theory and numerical analysis to determine the oil film parameters in the apex seal. In the beginning working parameters like velocity course, load distribution, possible sliding surface geometry had to be defined. In the next step defined parameters which have influence on the oil film parameters were taken into account during computer simulation researches. The basics of the hydrodynamic lubrication theory are not going to be included in this paper as they are well described in accessible and approved scientific literature [3-5].

2. Kinematics of the apex seal

The obvious difference between the Wankel engine and reciprocating engine is kind of piston movement. Both kinds of movements have fundamental influence on sealing elements kinematics. For the Wankel engine it is characteristic that there is no zero velocity of apex seal. This feature makes it different from the reciprocating engine in which piston with piston ring packet velocity for TDC and BDC is zero. In this place the advantage of the Wankel engine has to be pointed out because it is proven that the main reason of sustaining the oil film between cooperating components is the slide effect but the necessary condition is non zero relative velocity. This means that the apex seal meet the main requirement of sustaining the oil film during complete rotor turn which is very favourable. The squeezing out and the inclination effects without sliding effect because of the basic of its formation can not sustain the oil film in infinite time. The comparison of the apex seal velocity and the piston velocity has been shown on Fig. 1.

The other factor that differs from the apex seal from the piston ring is the angle of attack which for the Wankel engine is defined as an angle between normal line to the trochoidal cylinder liner and rotor midperpendicular in tangency point of apex seal with cylinder liner. On Fig. 2 the angle of attack is denoted as a β angle. Tangent of angle of attack is represented by dependence (1) [1]:

$$tg\beta = \frac{3\sin 2\varphi}{Z + 3\cos 2\varphi}, \tag{1}$$

where:

φ - the angle of turn,

Z - characteristic parameter of trochoid.

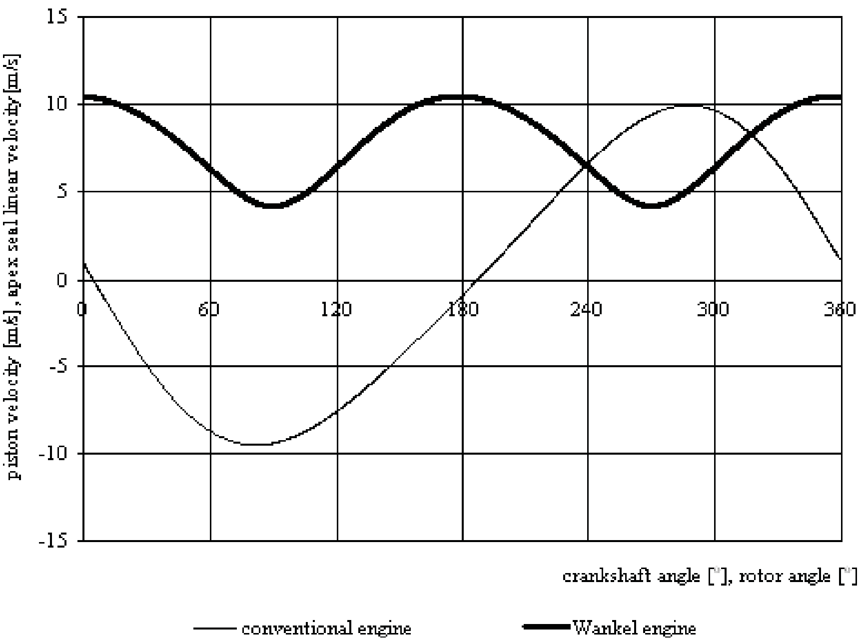

Fig. 1. Speed curves of piston and apex seal for full revolution of the crankshaft and Wankel engine rotor

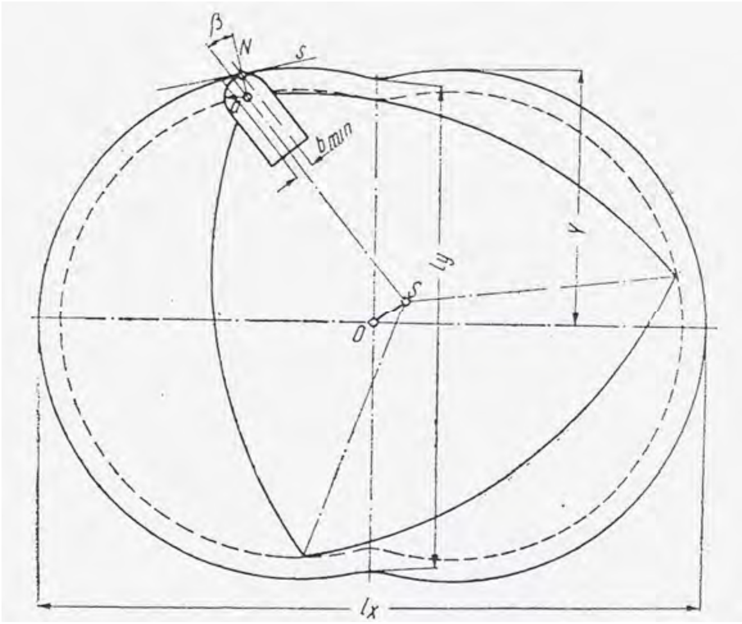

Fig. 2. β attack angle of the apex seal [1]

On base of above formula it is possible to determine the apex seal angle of attack for every rotor position. For characteristic parameter Z = 7, which is used in Mazda's RENESIS engine, minimum and maximum values of angle of attack are adequately -25.38º and 25.38°. The leaning amplitude value of apex seal is several grades higher than for compression ring in classic reciprocating engine which results from the piston transverse movements. Course of apex seal angle of attack changes is shown on Fig. 3.

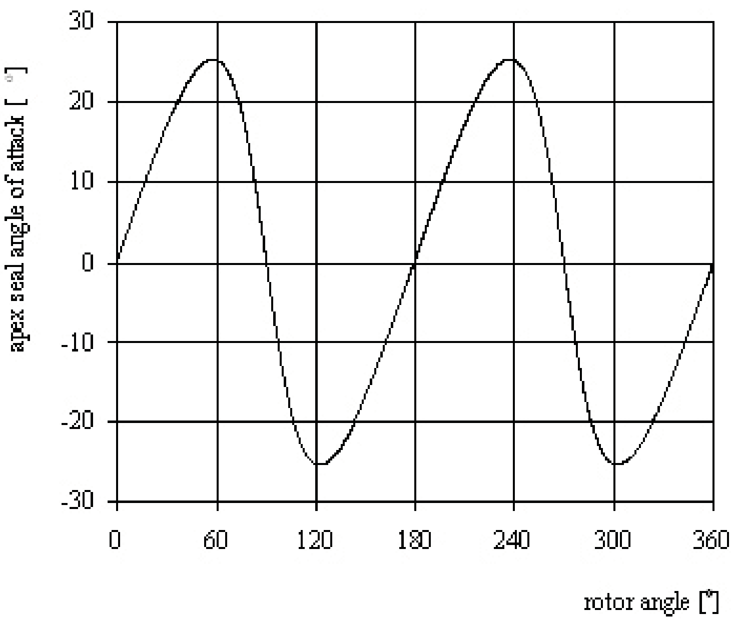

Fig. 3. Attack angle values of the apex seal for instantaneous positions during full rotor revolution

Such high variation of the attack angle force constructors to use slat with small curvature radius which is not the optimal one considering oil film parameters. The shape of apex seal working surface is much different from the working surfaces of compression rings where the barrel deflection is only few μm. Comparison of barrel compression ring and apex seal is shown on Fig. 4.

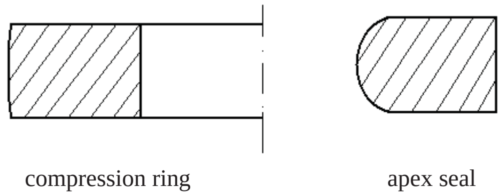

Fig. 4. Comparison of working surface shapes for the piston sealing ring and apex seal

3. Load distribution in the apex seal

To determine the oil film parameters the relation of apex seal and trochoidal cylinder position has to be specified. Apex seal position results from changes of its angle of attack and from loads that act on the slat. The state of loads of the apex seal is significantly different from the sealing piston ring state of loads used in conventional engines but there are also some similarities like the fact that the main force pressing the apex seal slat down to the cylinder surface is the gaseous pressure force. The apex seal slat separates two adjacent chambers where two different strokes of the four stroke engine proceed. These strokes are repeatable and they are out of phase by 120 degrees of the rotor angle of rotation, what is shown on Fig. 5. On the axis of abscissas segmentation of the full revolution of the rotor to 24 equal parts is presented.

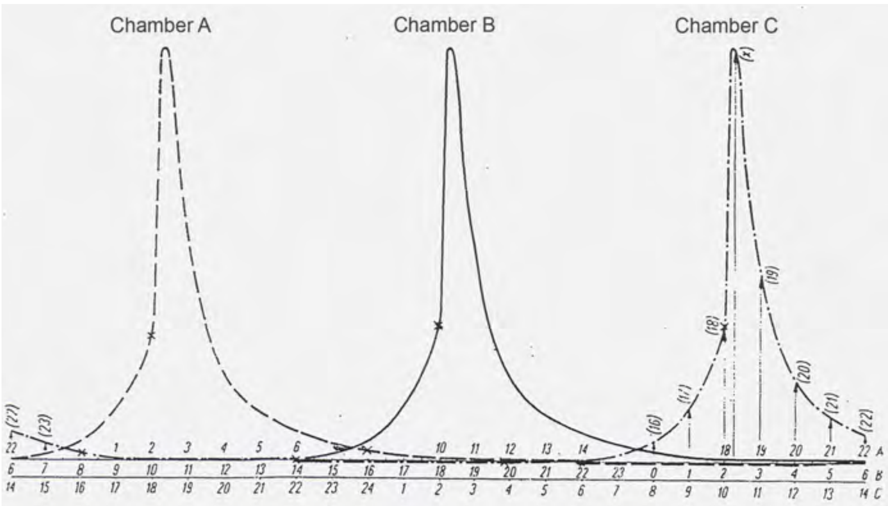

Fig. 5. Developed indicator diagram of the Wankel engine [1]

The important component of the resultant radial force is the radial component of the centrifugal force of inertia F_{by}. F_{by} force is variable not only for value but also for vector sense in the engines where Z parameter value is less than 9, which means that this force is alternatively pressing down the apex seal slat to the cylinder surface and pulling it off from the cylinder surface. The course of the radial component of the centrifugal force of inertia F_{by} is expressed by formula (2) [1]:

$$F_{by} = -\left(\frac{R}{9} - e\cos\frac{2\alpha}{3}\right)\omega_k^2 \cdot m_L\,, \tag{2}$$

where:
R - radius of the rotor,
e - eccentricity of the engine shaft,
α - angle of shaft rotation,
ω_k - shaft angular velocity,
m_L - mass of the apex seal slat.

The course of the radial component of the centrifugal force of inertia F_{by} determined by formula 2 for the following values:
- R = 105 mm,
- e = 15 mm,
- ω_k = 523.60 rad/s (5000 rpm),
- m_L = 0.004 kg,

is shown on Fig. 6.

To avoid the situation where the apex seal slat detaches from the cylinder surface, which in obvious way leads to gas blow-by effect between adjacent chambers, it is necessary to apply a plate spring. The characteristic of the spring must be adequately selected so in the shorter axis of the trochoid area the apex seal slat will not detach from the cylinder surface and in the longer axis of the trochoid area it will not cause too much of pressing down force.

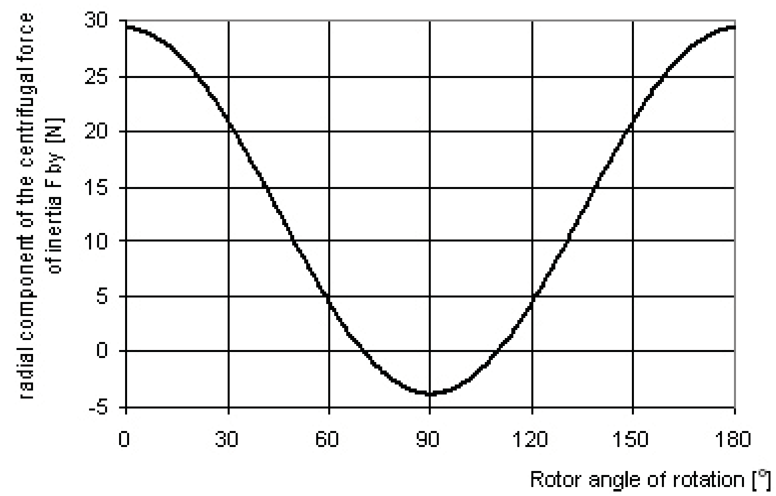

Fig. 6. The example of radial component of the centrifugal force of inertia F_{by} course

4. Numerical simulations results

The only Wankel engine being produced nowadays in series is the Mazda's Renesis engine, which drives the RX-8 model. According to this state the authors decided to take to the simulations the following dimensions of Renesis engine:
- R = 105 mm,
- e = 15 mm,
- Z = 7,
- V_s = 654 cm^3,
- i = 2,
- V_{ss} = 1308 cm^3,
- $\varepsilon = 10$.

Also different shapes of the apex seal sliding surface, show on Fig. 7 were taken into consideration.

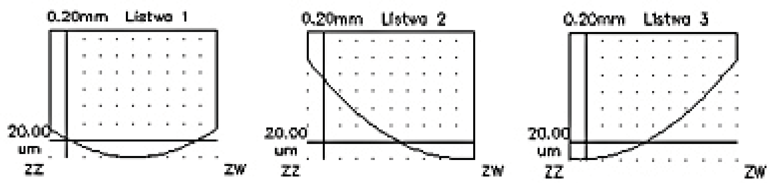

Fig. 7. Shapes of the apex seal sliding surfaces used during numerical simulations

Because of limited paper volume only selected example results can be presented. Whole packet of researches includes series of input parameters combinations.

As a first the results for 750 rpm and full load of the engine are presented on Fig. 8. The shape of sliding surface is shown on the right side of the figure.

Three courses represent oil film thickness (thick line), thickness of oil layer left after slat pass (medium line) and friction force (thin line). The starting oil layer thickness on the cylinder surface was 1.5 μm. As expected, the lowest oil film thickness occurs in the area of the shorter trochoid axis as a result of the lowest linear velocity of the apex seal slat, and as it is proven, the linear velocity is the main factor that allows maintaining the oil film. The pulling off force for this engine speed is low and no influence on oil film thickness can be noticed.

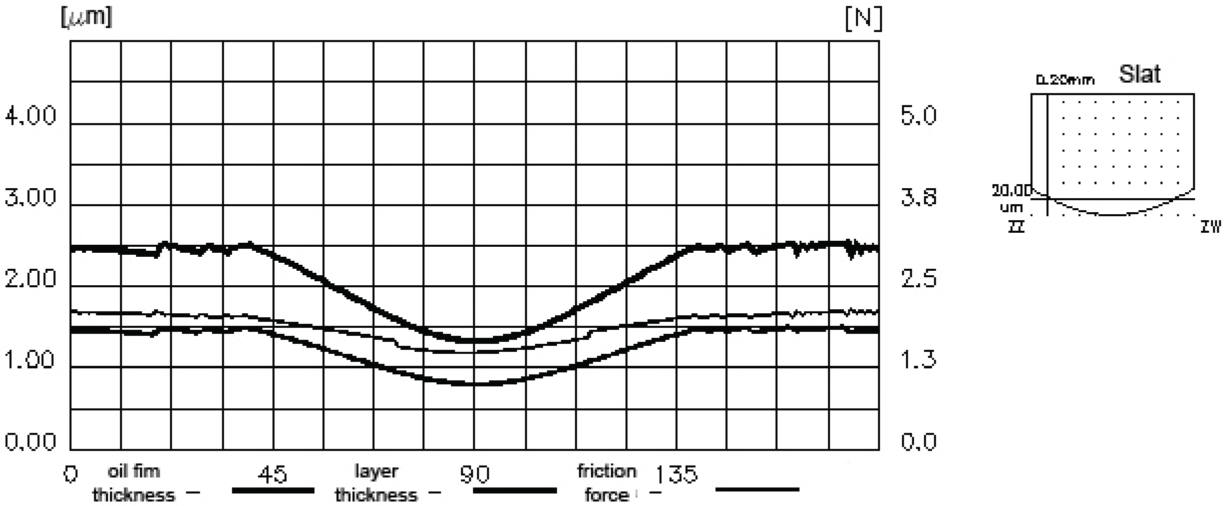

Fig. 8. Oil film parameters courses for 750 rpm

Also in the same area the oil film layer is being scraped by slat which reveals as a lower oil layer thickness than 1.5 μm after slat pass. The friction force is connected with two factors: velocity and sliding surface area that is covered with oil film. In the area of shorter trochoid surface friction force is less that in the other areas, but this decline is not as distinct as velocity decrees would suggest. To explain this phenomenon diagram of sliding surface covering area has to be analyzed. Proper diagram is presented on Fig. 9.

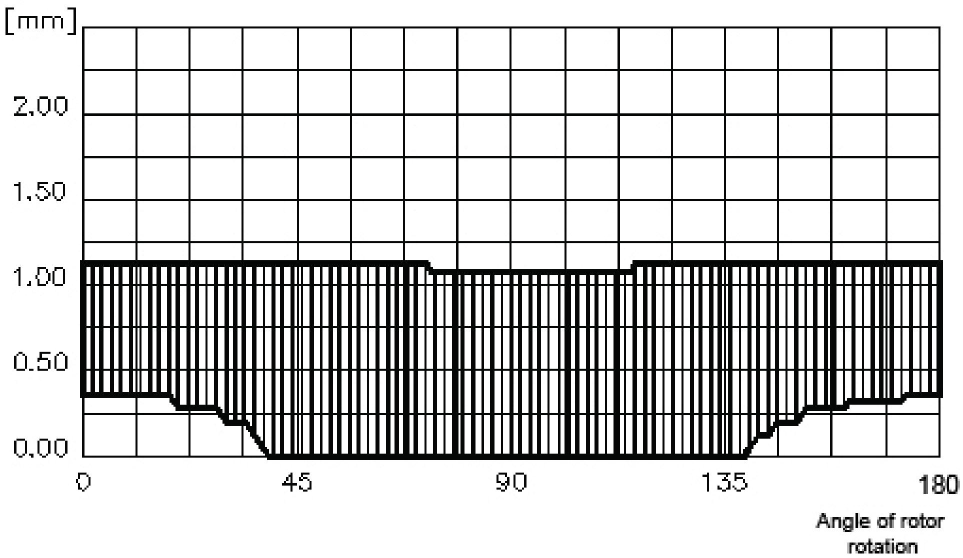

Fig. 9. Area of apex seal slat sliding surface covered with oil film

On the axis of ordinates represents slat width which total is 2 mm. 0 mm value is the attacking surface and 2 mm value is run-off surface. For the zero rotor position only about 40% of the total slat width is covered with oil film and because the attacking edge is not covered with oil film there is no oil scraping effect. With rotor movement the area of sliding surface covered with oil film increases what explains why the friction force do not decrease so much as firstly expected. When the oil film reaches front edge of the slat oil gets started to being scraped, which could be also seen on Fig. 12. In the range of 135-180 rotor angular positions the covered area once more decreases, the scraping effect is again not present, and connected with growing linear velocity friction force slightly increase.

Authors will omit results obtained for 3000 rpm and will present results for 9000 rpm as a representation of extreme operating conditions of the apex seal. The results are presented on Fig. 10. The starting oil layer thickness on the cylinder surface was 4 μm and as it can be read from the following diagram for whole rotor revolution that there is no scraping effect and layer thickness is nearly constant at the starting level of 4 μm. Very interesting phenomena is that the oil film thickness is also almost unchanged and is only two times thicker than in case of 750 rpm. This is much caused by inertia force. In the longer axis area inertia force pressing the slat against the cylinder surface is much higher than for 750 rpm and this makes impossible to achieve much higher values of the oil film thickness, whereas in the shorter trochoid axis area the inertia force pulls off the slat from the cylinder surface and results in higher values of the oil film. It is worth to notice that the maximum friction force is five times higher than for 750 rpm, which will cause drastic increase of friction power.

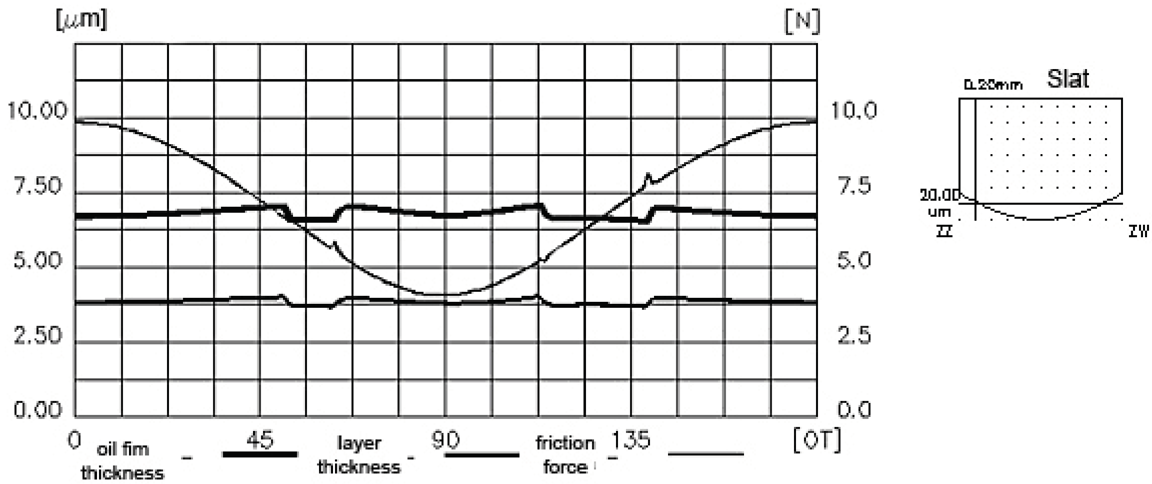

Fig. 10. Oil film parameters courses for 750 rpm

5. Conclusions

The apex seal is much more problematic type of rotor sealing than piston rings in conventional reciprocating engines. Disadvantageous shape of the sliding surface of the apex seal and changes of inertia force sense makes troubles to obtain sufficient lubricating conditions in the apex seal for whole range of the engine operating conditions. This leads to shorter service life of the apex seal and to impairing the engine working indicators like maximum output power or fuel consumption. It seems that area of shorter trochoid surface is the critical one for the apex seal durability especially during low speed and high load of the engine operation.

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