ISSN: 2350-0328
International Journal of Advanced Research in Science, Engineering and Technology
Vol. 6, Issue 5 , May 2019

Modeling and simulation of an axial piston pump with differential pressure limiter

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ABSTRACT: A recent study of U.S. Department of Energy shows that the efficiency of fluid power averages 21 percent. This offers a huge opportunity to improve the current state-of-the-art of fluid power machines, in particular to improve the energy consumption of current applications. This paper describes the mathematical model of the flow generation unit made in industrial software Amesim. Pump working characteristics are studied with two cases, namely by varying orifice area with constant load; and constant orifice area with varying load. In both cases, it is seen that the pump generates a flow rate according to operator signal. As real valve VPA 40LS made by Danfoss is taken as object of study, it is dismantled and 3D model is made in SolidWorks. Finally, its mathematical model is studied in Amesim with constant load and varying operator signal, and with varying load with constant opeartor signal. A discussion is made on pump pre-set pressure margin, which is always set higher than necessary due to uncertainty and worst operating conditions that in turn leads to unnecessary energy losses.

KEY WORDS: Fluid Power, Hydraulics, axial piston pump.

I.INTRODUCTION

Due to the advantages of high power weight ratio and high load capability, fluid power technology has been used for long time in all sorts of mobile machinery, for example in construction, forestry and agricultural machinery [1]. Load Sensing systems are the industry standard at the moment. In those systems, it is crucial to determine design pump pressure margin. This margin is usually set to overcome the losses in the hoses and directional valve. Therefore, the worst working conditions should be taken as a reference to correctly choose the pre-set pressure margin ito satisfy requirements of all operating conditions. Recent studies in Electronic Load Sensing systems demonstrated the possiblity to reduce pressure losses in the directional valve by optimizing the setting of pressure margin s according to operating flow rate. Due to uncertainty of pressure losses in the hoses, in order to provide sufficient pressure drop across the directional valve, pump pre-set margin is always set to higher value than needed pressure drop at that operating point. But the question remains open – how much higher is sufficient? As a result of unnecessary pressure losses, energy efficiency of Fluid Power Systems leaves much to be investigated [2].

II. SIGNIFICANCE OF THE TOPIC

This paper addresses the mathematical model of the flow generation unit. It is made in Amesim and working characteristics are studied with two cases, namely varying orifice area with constant load; and constant orifice area with varying load. In both cases, it is seen that the pump generates a flow rate according to operator signal. In other words, pump flow rate is the function of the operator signal only. When the flow rate demand changes, pump reacts to it no matter what is the pressure at the load. Thus, the mathematical model of the flow generation unit correctly describes the real behaviour of the real valve VPA 40LS made by Danfoss.

This work is useful to study of Load Sensing systems, since they are the industry standard at the moment. In LS systems, it is crucial to determine design pump pressure margin. This margin is usually set to overcome the losses in the hoses and directional valve. Therefore, the worst working conditions should be taken as a reference to correctly choose the pre-set pressure margin ito satisfy requirements of all operating conditions.

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III. LITERATURE SURVEY

Due to the advantages of high power weight ratio and high load capability, fluid power technology has been used for long time in all sorts of mobile machinery, for example in construction, forestry and agricultural machinery [1]. Load Sensing systems are the industry standard at the moment. In those systems, it is crucial to determine design pump pressure margin. This margin is usually set to overcome the losses in the hoses and directional valve. Therefore, the worst working conditions should be taken as a reference to correctly choose the pre-set pressure margin ito satisfy requirements of all operating conditions. Recent studies in Electronic Load Sensing systems demonstrated the possiblity to reduce pressure losses in the directional valve by optimizing the setting of pressure margin s according to operating flow rate. Due to uncertainty of pressure losses in the hoses, in order to provide sufficient pressure drop across the directional valve, pump pre-set margin is always set to higher value than needed pressure drop at that operating point. But the question remains open – how much higher is sufficient? As a result of unnecessary pressure losses, energy efficiency of Fluid Power Systems leaves much to be investigated [2].
In figure 1, there is an ISO representation of displacement control of a pump via an absolute pressure limiter [3]. At rest the 3/2 position direction control valve (DCV) closes the pump outlet with the user and the user is connected to the tank. When 3/2 DCV is switched, the pump outlet is connected to the user.

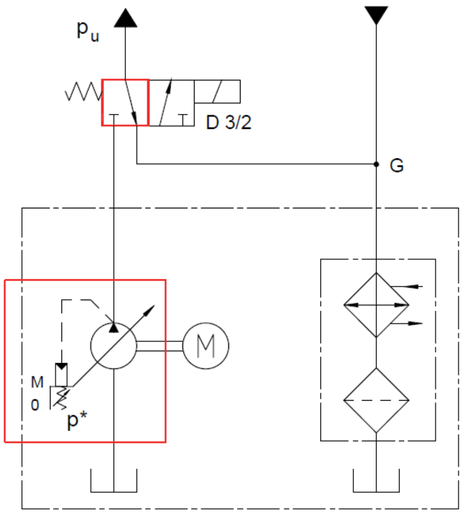

Figure 1. ISO scheme of direct acting absolute pressure limiter [3].

The actuator presents a piston mechanically connected to the pump. On the right of the piston there is a spring, while on the left there is a pressure of the pump outlet. So if the piston moves to the right, the displacement of the piston reduces the displacement of the pump. If the pump outlet pressure is less than the cracking pressure of the spring, the displacement of the pump is equal to the maximum displacement, a unity. Flow rate is imposed by the flow generation unit and the pressure is imposed by the load:

$$p_u < p^* \text{ and } \alpha = 1: \quad Q_u = Q_{max} \text{ and } M = V_{max} \cdot p_u \quad (1)$$

If the pump outlet pressure is equal to the cracking pressure of the spring, the piston regulates and the displacement of the pump is less than unity. Flow rate is imposed by the load, while the pressure is imposed by the power unit and equals to p*:

$$p_u = p^* \text{ and } \alpha < 1: Q_u = \alpha \cdot V_{max} \cdot \omega \text{ and } \quad M = \alpha \cdot V_{max} \cdot p_u \quad (2)$$

So the maximum pressure in the circuit is limited to a value of the cracking pressure of the spring p*. This kind of layout is made to work in the vertical chamber of the Q-p characteristic. The efficiency of the ideal pump is always a unity. In reality, this group is a variable displacement vane pump and the displacement is varied between 0 and 1 thanks to the stator movement. Now let us consider the similar layout but with a piloted absolute pressure limiter in figure 4.

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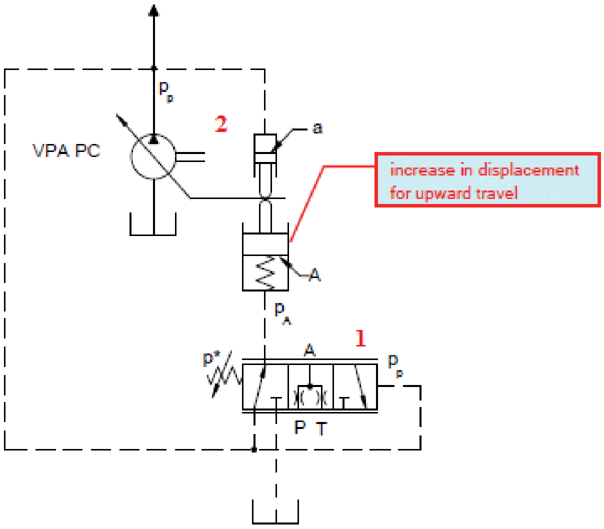

Figure 4: ISO scheme of a piloted absolute pressure limiter [3].

And there are two other components, namely a spool (1) and variable displacement pump (2). The spool is a continuous position 3/3 direction control valve with adjustable spring. At rest, pilot line pressure coming from the pump outlet acts on the actuator A. When the spool regulates, it throttles the pilot flow between P and A ports. At extreme right configuration, it connects the port A with the tank. The figure 5 below shows the valve spool.

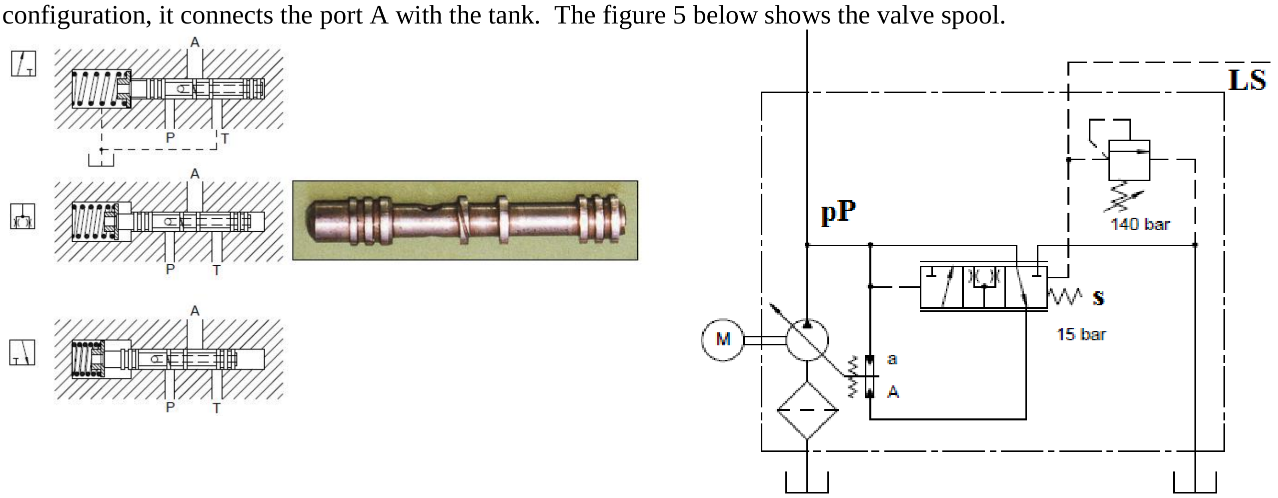

Figure 5: On the left, spool of the valve and on the right ISO scheme of differential pressure limiter given in [3].

The actuators are two pistons with different surface of influence. Both of them act on the radial piston pump stator. In the figure above, the upward travel of the horizontal bar increases the pump displacement. Let p_P be the pressure at the outlet of the pump.

We consider the cases in the following:

- If p_P < p*, then the spool is in rest position. $p_P = p_A$, and since a<A, $\alpha = \alpha_{max} = 1$
- If p_P = p*, the spool regulates. As a consequence, $p_A < p_P$. Also the actuators are in equilibrium. Therefore, pP is equal to p*.

IV. METHODOLOGY

A methodology to study the axial piston pump is straightforward. First, the real pump is taken and disassembled. Then, by measuring dimensions of each constituting component, its 3D model is made with SolidWorks CAD software. Fine measurements, such as a spool and differential pressure limter are made with optical 3D scanner Faro Arm ®. After completing 3D CAD model, pump is modeled by using Amesim software. All parameters used for Amesim is obtained from 3D CAD model. Discharge coefficient at the metering edge of spool is simulated by using SolidWorks Flow Simulation. Flow generation unit in our circuit is a variable displacement radial piston pump [4]. Its stator is seen in figure 7, where the right piston's surface of influence is A and the left one's is a. The actuator A acts to increase the pump displacement, while the other one decreases the displacement. Therefore, the pump displacement varies when these actuators are in balance.

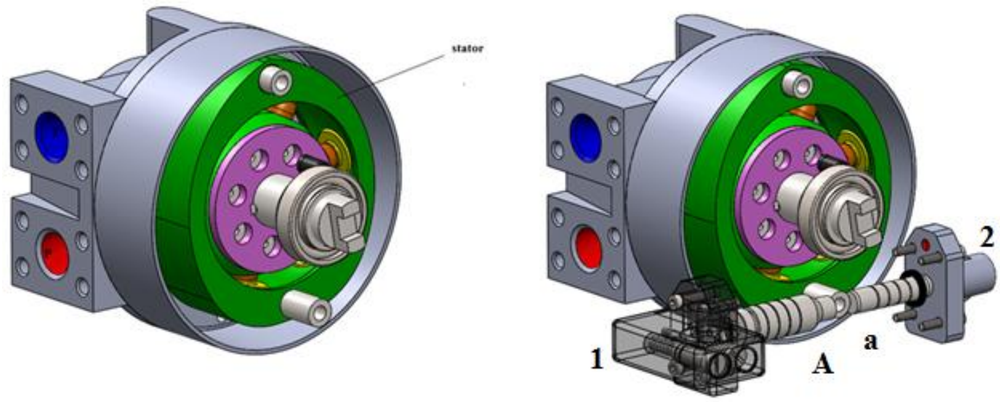

Figure 7: radial piston pump and actuators acting on its stator [4].

On both ends of the actuators, there are two assemblies (1) and (2). By changing the left module (1) as in figure 8, one can achieve different configuration of the pump. In figure 8 there is an isometric view of the module. There are 4 hydraulic ports connected to it (P, A, T, LS).

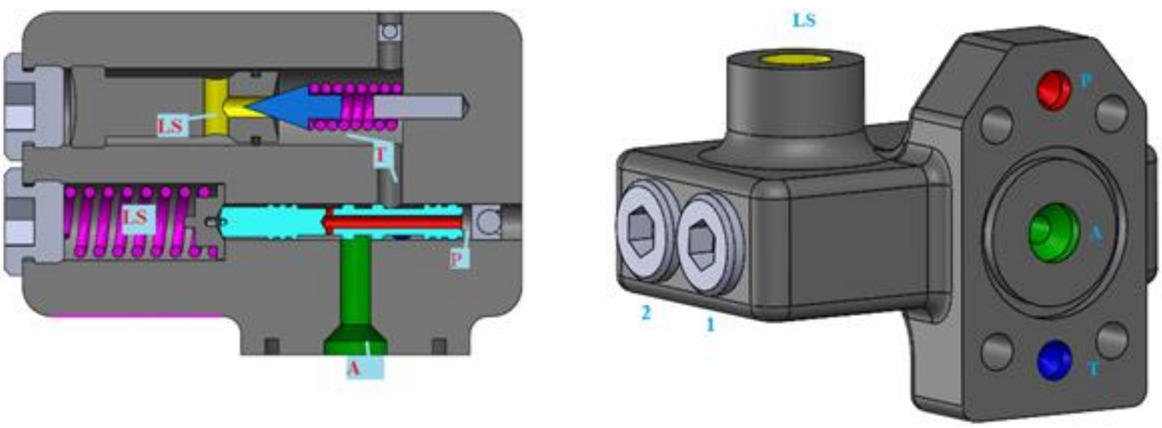

Figure 8: differential pressure limiter module

The module itself is mounted on the VPA40LS by means of the 4 bolts. There are two moving parts in the module – spool and the poppet of the pressure relief valve on the LS line, as labeled (1) and (2). While in figure 12, one can see the pressure relief valve in detail. If the pressure in the LS line increases and equals to the pressure setting of the poppet spring p** = 140 bar, the poppet regulates and limits pLS inside the module at that value thanks to the restrictor R (seen on top view below). One can notice the notch on the spool as in figure 5 ensures the minimum flow area between P and A when spool is in regulating condition. The spool has an axial channel and the radial channels. They are both

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International Journal of Advanced Research in Science, Engineering and Technology
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connected to each other and the axial channel makes a thru-hole on the right of the spool. The two shoulders on the middle are chamfered while the shoulders on the ends have groves to avoid blocking of the spool due to eccentricity.

V. EXPERIMENTAL RESULTS

Final model of flow generation unit with differential pressure limiter is is below.

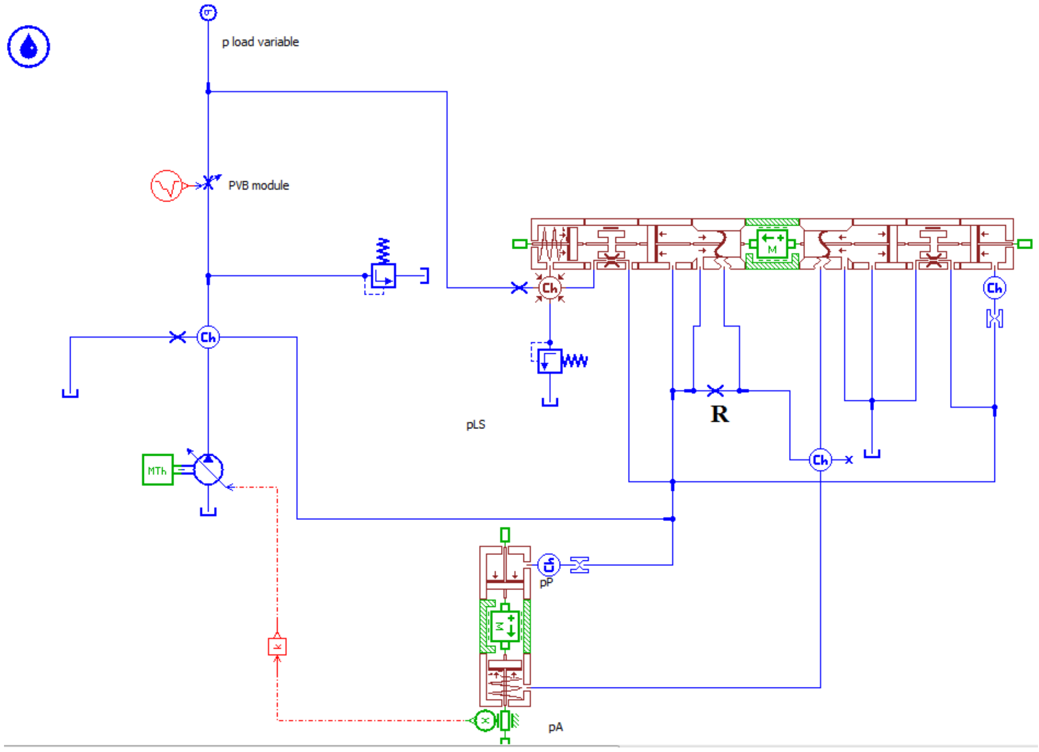

Fig 10: Amesim model of the differential pressure limiter

Case 1. Constant load with varying orifice area

In the first simulation, load pressure pLS is kept at constant 50 bar and flow area of the variable orifice *X(t)* representing the PVB module varied from 30% to 40%.

$$X(t) = \begin{cases} 30\%, & t < 5 \\ 40\%, & t \geq 5 \end{cases}$$

As seen in the figure, the pump is pressurized around 68 bars no matter how is the orifice signal. This is a value of *pLS* + *s*, where *s* = 18 bar and pLS = 50 bar. So the differential pressure limiter maintains constant pressure drop equal to ***s*** across the variable orifice PVB.

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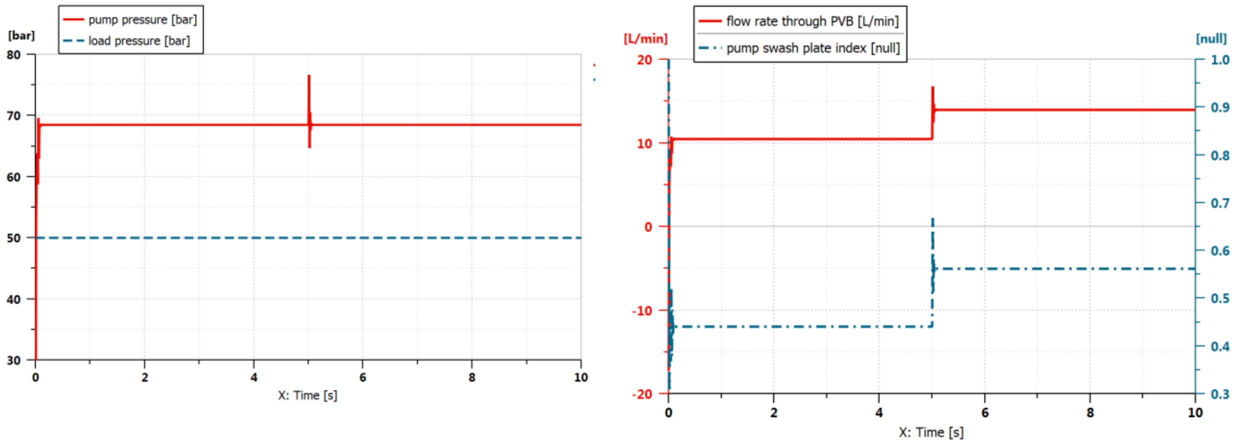

Figure 11. Pump pressure remains constant while orifice signals change (left figure). On the right, flow rate thorugh PVB increase (solid line) as a consequence of pump swash plate index increase (dotted line).

Case 2. Variable load with constant signal

In this case, load is varied while keeping the PVB orifice signal constant. If the mathematical model is correct, pump pressure should increase in order to keep the pressure drop across the orifice constant. As a consequence, flow rate generated by pump should remain constant whenever orifice signal is constant. To simulate this behaviour, the flow area of the variable orifice is kept constant at 40% and the load pressure pLS is varied from 10 bars to 100 bars.

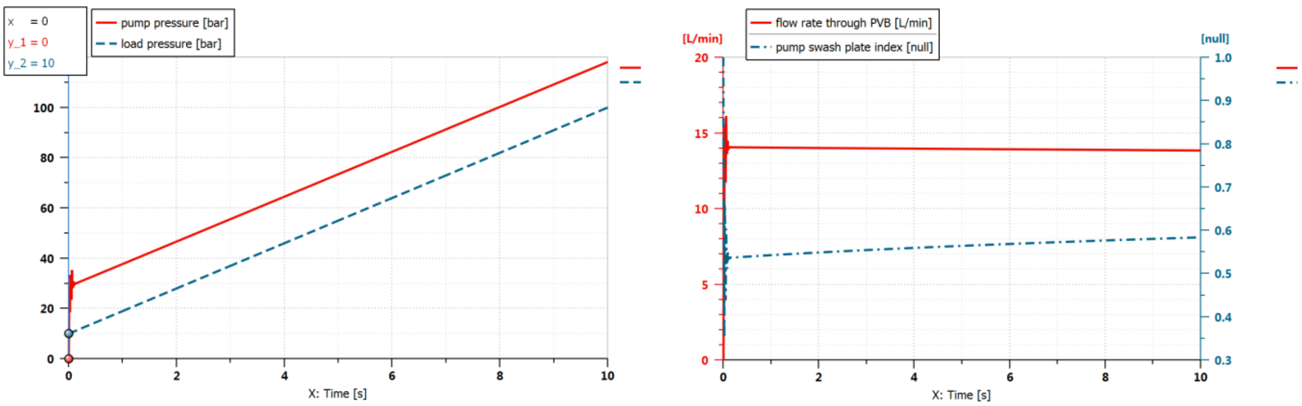

Figure 12. Pump pressure and load pressure (left). Flow rate and pump swash (right).

As seen before, the pump is pressurized at $pLS + s$. It is interesting to note that swash index of the pump increases. This is due to the increased leakages of the pump operating at elevated pressures, thus pump swash plate index increase to compensate internal leakages.

VI. CONCLUSION AND FUTURE WORK

This paper addresses the mathematical model of the flow generation unit. It is made in Amesim and working characteristics are studied with two cases, namely varying orifice area with constant load; and constant orifice area with varying load. In both cases, it is seen that the pump generates a flow rate according to operator signal. In other words, pump flow rate is the function of the operator signal only. When the flow rate demand changes, pump reacts to it no matter what is the pressure at the load. Thus, the mathematical model of the flow generation unit correctly describes the real behaviour of the real valve VPA 40LS made by Danfoss.

This work is useful to study of Load Sensing systems, since they are the industry standard at the moment. In LS systems, it is crucial to determine design pump pressure margin. This margin is usually set to overcome the losses in the hoses and directional valve. Therefore, the worst working conditions should be taken as a reference to correctly choose the pre-set pressure margin ito satisfy requirements of all operating conditions [6].

Moreover, this work can be useful to kickstart recent studies in Electronic Load Sensing systems, in which pressure losses in the directional valve can be reduced by optimizing the setting of pressure margin s according to operating flow rate [7]. Due to uncertainty of pressure losses in the hoses, in order to provide sufficient pressure drop across the directional valve, pump pre-set margin is always set to higher value than needed pressure drop at that operating point. But the question remains open – how much higher is sufficient? As a result of unnecessary pressure losses, this study is useful to determine pre-set margin in the future works.

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