Experimental studies of the working processes of a reciprocating hybrid power machine with a slot seal of step-type with various working fluids

V E Shcherba, V V Shala, E Yu Nosov, G S Averyanov, M E Linkov

Omsk State Technical University, 11, Mira Ave., Omsk, 644050, Russian Federation

Abstract. The article considers results of experimental studies of a piston hybrid power machine with a slot seal of step-type. While experimental studies, the following parameters were taken as independent parameters: the type of cooling liquid (I-20A oil and MGE-46B oil were used), the injection pressure in the compressor section, the discharge pressure in the pump section, the crankshaft speed.

The experimental studies determined that using of low-viscosity oil I-20A made possible to reduce the temperature of the cylinder-piston group and the temperature of the intake gas.

1. Introduction
Reciprocating hybrid power machines of volumetric action (RHPM) are distinguished by their high compactness, low mass, improved cooling of the cylinder-piston group of the compressor section, high efficiency and performance of as a compressor section as a pump one [1, 2]. With a high ratio of discharge pressure in the compressor $p_{dc}$ to discharge pressure in the pump section $p_{dp}$ gap seals with different bandwidth are applied [3, 4].

As shown by the studies [5, 6], with a high ratio $p_{dc}/p_{dp}$, one of the most effective designs is a piston hybrid power machine with a slotted seal of a stepped type [7, 8].

So, it seems expedient to carry out experimental studies on the effect of cooling liquids on the working processes of RHPM with a slotted seal of a stepped type.

2. Formulation of the problem
To carry out experimental studies, an experimental sample and a test bench were created, figure 1 (photo).

The experimental sample, figure 2, had a crosshead pattern. The cavity above piston 15 serves to compress and to move gas, the cavity below the piston serves to compress and to move fluid. The piston seal is slotted stepping.

The piston hybrid power machine with a stepped seal had the following basic geometric parameters:
- the diameter of the cylinder is 0.05 m;
- piston stroke is 0.05 m;
- total working length of the cylinder is 0.1 m;
- length of the upper sealing part is 0.055 m;
- length of the lower sealing part is 0.045 m;
- clearance between the piston and the cylinder in the upper part is 14 mkm;
- clearance between the piston and the cylinder in the lower part is 61.5 mkm;
- length of the piston is 0.049 m.

The cylinder, the piston and rod are steel, made of steel 45, heat treatment is hardened HRc 35...40.
In carrying out the experimental studies, there were measured static pressures in the compressor section and pump section, the temperatures of the cylinder-piston group, the temperature of the intake gas, gas and fluid flow, and the amount of the fluid carried into the compressor discharge line per unit time. Measurement of static pressure was carried out with manometers, class 1 of type MPZ-UUHL1. To measure temperatures, there were used semiconductor thermistors of DTS 035-50m.B3.80 type. To measure airflow in the compressor section, two methods of investigation were used: volumetric and thermo anemometric, using flow meters of the volumetric type “Vector-04” and an air flow sensor SMCPF2A, respectively. The fluid flow rate was measured by a turbine fluid flow transducer TPR8-1-1B. The measurement of the fluid amount in the injected gas was carried out by a volumetric method, its essence is to install a discharge line for the oil-moisture separator containing a package of brass nets with a cell of 0.5 – 1.0 mm or the Raschig rings. When the oil-moisture separator is full up to about 1/4…1/3, the fluid is poured into a measuring vessel to settle and then the height of the oil column and concentrated fluid is to be determined. Knowing the time of the experiment, it is easy to determine the discharge rate of the fluid. The error in measuring gas and fluid flow rates was (1...3) %. The error in measuring pressure and temperature was not more than 1%. In carrying out experimental studies, independent parameters were chosen:
1. Working fluids in the pump section.
2. Discharge pressure in the compressor section is $p_{dc}$.
3. Discharge pressure in the pump section is $p_{dp}$.
4. Angular speed of cranked shaft rotation is $n_{rev}$.

The experimental studies assumed that the suction pressure in the compressor section was uniformly the injection pressure in the pump section and equal to 1bar. When planning the experiment, a classic plan with fractional replicas was used. The following ranges of variation of independent parameters were used:
4 bar $\leq p_{dc} \leq$ 7 bar
4 bar $\leq p_{dp} \leq$ 11 bar
250 rpm $\leq n_{rev} \leq$ 350 rpm.
There were used oil I-20A and oil MGE-46V as working fluids. Industrial oil I-20A is general-purpose oil, a distillate or a distillate mixture with a residual of sulphurous and low-sulfur oils of selective purification.

Figure 1. General view of the experimental stand and the piston hybrid power machine with step-type slot seal
Figure 2. Section view of the piston hybrid energy machine with step-type slot seal.

1 is crankcase, 2 is blind joint, 3 is the bearing lid, 4 is the bearing, 5 is stuffing box, 6 is a drive shaft, 7 is a crank, 8 counterweight, 9 is a crankcase cover, 10 is a crosshead guide, 11 is a crosshead, 12 is a gudgeon, 13 is a rod, 14 is a stem, 15 is a piston, 16 is a stuffing box, 17 is seal, 18 is a cylinder, 19 is a valve box, 20 is vent line, 21 is back rest.

MGE-46V oil is mainly used for hydrostatic transmissions operating at pressures up to 35 MPa in the temperature range from -10 up to 80 °C. The oil is produced on the basis of industrial oils with antioxidant, anti-wear, and dispersancy and antifoam additives.
The kinematic viscosity of the I-20A oil at 40 °C is 25…35 mm²/s, and the kinematic viscosity of oil MGE-46B at the same temperature is 41.4…50.6 mm²/s, i.e. oil MGE-46V is more viscous than oil I-20A in about 1.5 times. Specific heat of oil I-20A at t = 40 °C is 1.67…2.5 kJ/kg·K, and the specific heat of MGE-46B oil at the same temperature is 1.95 kJ/kg·K, the specific heat of the oils is approximately the same.

3. The results discussion
As the injection pressure in the compressor section increases, the amount of fluid carried out decreases. This is due to an increase in the amount of fluid flowing from the compressor section to the pump section. It should be noted that the amount of MGE-46B oil having a higher viscosity comes from the pumping section to the compressor section less than the I-20A oil. This in turn leads to the fact that the volume of oil MGE-46B in the injected gas is less than the volume of oil I-20A (see figure 3). In accordance with the results presented in Fig. 3, the volume of the extracted oil MGE-46V is approximately 1.5 times less than the I-20A oil. So pressure losses may be reduced during the injection process, but, on the other hand, also cooling of the injected gas is worsened. Increasing $p_{dc}$ difference in fluid content in the oil in the pumped for I-20A oil and MGE-46B is increasing.

![Figure 3. Dependence of the discharge of discharge liquid from the injected gas on the discharge pressure in the compressor section for different working bodies in the pump section: 1. I-20A oil; 2. MGE-46B oil](image)

Increasing the viscosity of the oil reduces the flow of coolant from the pump section to the compressor section, and it leads to deterioration in the cylinder-piston group, and the compressed gas cooling. Thus, when using more viscous oil (MGE-46V), the average cylinder temperature increases by about 2 K over the entire range of variation $p_{dc}$, and the lid temperature is about 1K. Presented in figure 4 results conclude that the suction temperature into a compressor section increases with increasing $p_{dc}$. Using of less viscous oil (I-20A) reduces the temperature of the intake gas by 2.5…2.0 K in the range $p_{dc}$ from 4 bar up to 7 bar. As the discharge pressure in the pumping section increases, the amount of liquid carried out of the compressor section increases, and it is caused by an increase in the amount of fluid coming from the pump section to the compressor section.

As the viscosity of the fluid increases, the amount of fluid leaking decreases and it leads to a decrease in the amount of fluid to be carried out and to deterioration in the cooling of the cylinder-piston group, the intake and injected gas. Thus, when $p_{dc} < 10$ bar difference in the amount of the staked fluid into discharge line in the compressor section with more viscous fluid will be 20mm/min. As the discharge pressure in the pump section increases, mass transfer between the working cavities of the compressor
and pump sections increases, and it improves cooling of the cylinder-piston group and the compressed gas. When using low-viscosity fluids, cooling is improved. Thus, when \( p_{dp} = 5 \) bar difference in average temperature of the cylinder using a low viscosity oil is 2.5 K (see figure 5). By increasing \( p_{dp} \) up to 11 bar, difference is reduced to 0.5 K. This is due to the large amount of fluid coming from the pump section to the compressor section. It should be noted that the difference in the temperatures of the intake gas has a similar nature. Thus, when \( p_{dp} = 5 \) bar it is 1.5 K, and at \( p_{dp} = 11 \) bar it is 0.5 K. The difference in average temperature of lid compressor section remains essentially constant over the entire range of variation \( p_{dp} \) and it is 1 K.

Figure 4. Dependence of the intake air temperature on the discharge pressure in the compressor section for different working bodies in the pump section: 1. I-20A oil; 2. MGE-46B oil

Figure 5. Dependence of the average cylinder temperature on the discharge pressure in the pump section for different working bodies in the pump section: 1. I-20A oil; 2. MGE-46B oil
Increasing the number of revolutions (angular velocity) of the crankshaft rotation, the amount of fluid carried to the consumer of the compressed gas increases. When using a low-viscosity fluid (I-20A oil), the amount of fluid to be carried out is increased in comparison with a high-viscosity fluid. This increase is 10…18 ml/min. It should be noted that as the angular velocity increases, the difference in the amount of fluid carried out increases from 10ml/min at \( n_{rev} = 250 \) rpm up to 18 ml/min at \( n_{rev} = 350 \) rpm. As the angular velocity increases, the time for removing the heat of compression in the compressor section decreases, and it leads to an increase in the temperature of the cylinder-piston group and the intake gas.

Presented in figure 6 results concluded that using a low-viscosity coolant the average temperature of the cylinder remains essentially constant, while using MGE-46B oil it increases by 2 K with increasing \( n_{rev} \) with 250 rpm up to 350 rpm. It should be noted that at \( n_{rev} = 250 \) rpm the average temperature of the cylinder, when using I-20A and MGE-46V oils, is the same. The temperature of the lid of the compressor section increases with increasing speed. If more viscous fluid is used, the temperature of the lid is higher than when using low-viscosity fluid throughout the entire angular velocity variation. This increase is from 1 K to 3 K with an increase in the angular velocity from 250 rpm up to 350 rpm. A similar picture is observed for the temperature of the intake gas. It should be noted that the difference in the temperatures of the intake gas is approximately the same when using oil I-20A and MGE-46B throughout the range of variation \( n_{rev} \) and it is 2.5 K.

4. Conclusions
The experimental studies of RHPM with step-type slot seal proved that when using more viscous oil in the pump section, the cooling of the cylinder-piston group and the intake gas are improved in the entire range of variation in the operating parameters while increasing the amount of fluid in the injected gas.

![Figure 6. Dependence of the average cylinder temperature on the angular velocity of the crankshaft for different working bodies in the pump section: 1. I-20A oil; 2. MGE-46B oil](image)

References

[1] Shcherba V E Bolshtyansky A P Kaygorodov S Yu Kuzeeva D A 2015 *Analysis of the main advantages of combining compressors and large-scale pumps into a single unit* (Bulletin of Machine Building) No 12 pp 15-19
[2] Shcherba V E Bolshtyansky A P Shalai V V et.al. 2013 Pump-compressors. Work processes and design basics (Moscow: Mechanical engineering) p 367

[3] Kondyurin A Yu Shcherba V E Shalay V V et al. 2016 Calculation of the liquid flow in the slit seal of the pump-compressor, made in the hydro diode form (Chemical and oil and gas engineering) №4 pp 30-34

[4] Shcherba V E Nesterenko G A Pavlyuchenko E A Vinichenko V S 2014 Calculation of the piston seal of the pump-compressor, made in the form of a concentric slot with a detachment groove in the body of the piston (Chemical and oil and gas engineering) No 2 pp 25-29

[5] Shcherba V E Lysenko E A Nesterenko G A Grigoryev A V Kondyurin A Yu Bazhenov A M 2016 Development and investigation of a piston seal, made in the form of a smooth slit of a step type, for a piston hybrid power machine of volumetric action (Chemical and oil and gas engineering) № 4 pp 45-48

[6] Bazhenov A M Shcherba V E Grigoryev A V Kondyurin A Yu Paramonov A M 2016 Analysis of the influence of the ratio of direct and reverse fluid flows in the slot seal of a piston hybrid power machine on the ratio of injection pressures in the pump and compressor cavities (Omsk Scientific Bulletin of OmSTU) No 6 (150) pp 45-49

[7] Shcherba V E Shalai V V Grigoryev A V Kondyurin A Yu Bazhenov A M 2018 Adaalysis of the influence of the injection pressure in the compressor section on the working processes and characteristics of a piston hybrid power machine with a slotted seal of a stepped type (News of higher educational institutions. Mechanical engineering) №4 (697) pp 49-57

[8] Bazhenov A M Shcherba V E Grigoryev A V Kondyurin A Yu Blinov V N. 2016 Analysis of the influence of eccentricity on the ratio of mass flow of liquid in the forward and reverse directions in a piston-type slot seal of a step-type type of a piston hybrid power machine of volumetric action (Omsk Scientific Bulletin of OmSTU) No 6 (150) pp 49-53