Use of the numerical experiment at the design stage of piston pulp pumps

Yuri Kossenko-Belinsky

Volodymyr Dahl East-Ukrainian National University, Lugansk, Ukraine

Summary. The results of numerical experiment on kinematic mathematical models of piston Pulp pumps with various modifications of the crank-connecting rod drive mechanisms and the number of cylinders. The recommendations on the rational choice of their number, the angular pitch of cranks and the ratio of their length to the length of the rod, and eccentricity to the radius of the crank mechanism.
Keywords. Pulp piston pump, the kinematic model, numerical experiment, crank, slider crank, the uneven flow and pressure, the angular pitch of cranks.

INTRODUCTION

Piston pumps are used for hydraulic transportation of abrasive media (solid minerals and waste of its production, different mortars) [1-3], and the drilling of exploratory and development wells for oil and gas [4-8].

These are powerful, heavy and cumbersome machines of maximum power (to 1500 kW) and weight (to 65t) [1; 9-12]. Their research and operational development by means of the physical experiments on natural samples is difficult, expensive, and often impossible.

For faster and better solutions to the problems of existing pumps improvements and creating new prospective pump it is necessary to change the ratio of physical and numerical simulation for the latter. To do this, we developed [13], mathematical (kinematic and dynamic) models of the workflow pumps and software for the numerical simulation by computer.

OBJECT OF RESEARCH

In developing the conceptual design of pumps it is enough to research them on simple kinematic models, in which the motion of the fluid in the pipes is determined only by the number of chambers and the kinematics of the propellants, depending on the selected driver and ratios of linear and angular measurements to specific units. The numerical experiments allow us to find not only the impact of these factors on the kinematics of the fluid and on changing its flow rate in the inlet and outlet pipes, but also to identify each time the theoretical values of the coefficients of uneven flow and pressure pump, which is very important.

The very study of pumps on kinematic models allow us to determine feasibility of using one or the other pump design concept for future settlement of design development and implementation of a numerical experiment on a dynamic model.

RESULTS

First there studied piston pumps with single acting cylinders and traditional central slider-crank drive mechanisms. The independent variables in this case are the number of cylinders Z and ratio r / l r (the radius of the crank rod to length l). The result of each experiment could be obtained graphical dependences of the relative displacement, velocity and acceleration of the piston, the relative flow rates of the suction and discharge piping, and calculated the theoretical values of the coefficients of the uneven flow and pressure in them. The latter are determined by the expressions (1), the latter of which is valid at the pump for a short pipe with throttle load.

$$\Delta = \frac{Q_{max} - Q_{min}}{Q_{av}},$$

$$\delta = \frac{Q^2{}_{max} - Q^2{}_{min}}{Q^2{}_{av}} \quad . \qquad (1)$$

Some of the results are shown in table. 1. Their analysis shows:

for odd Z increasing r/l leads to significant increase Δ, especially δ;

for even Z with equal angular increments cranks value r/l has no effect on the coefficients of Δ and δ, the values of which are respectively more than the nearest pumps with odd Z.

In addition, it was found that for pumps with an even number of cylinders (6, 10, 12, etc.) that are divisible by an odd number, you can greatly reduce uneven flow in pipelines. To do this, all of the number of cylinders Z = m * k should be divided into m groups with an odd number of cylinders k in a group and equal angular cranks increments α = 2π / k, but the groups rotate relative to each other at an angle Δα, whose value is determined for each Z at each r / l [14]. For example, when Z = 6 * (step-uniform angular pitch of the cranks – the last two columns in table 1), m = 2, k = 3, α =120°, and Δα = π / Z = 30°, r / l = 0. With increasing r / l efficiency of the process decreases, i.e. Δ and δ worsen (increase). At r / l = 0, 1 Δα = 38,3°, and at r / l = 0,2 Δα = 44,3° [14].

It should be noted that, in contrast to the diameter d_s in the suction pipe, which depends only on the pump flow Q and pistons diameter D, which depends also on their number Z, the value of their stroke S and the number of double strokes n, effect, but in different ways, the ratios ψ = r / l, [13] and table 2. At equal angular step cranks [16]: for pumps with even Z>2 S and n are equal at different ψ and Q and the same Z; for pumps with both odd Z and with even Z, but having the step-uniform angular step cranks, when ψ increases, the piston stroke length is increased while n at the same time is decreased.

Dependences to determine the d_s, D, S, and n obtained in [14] with the durability of the cylinder-piston pair [17] and the suction capacity of the pump, but without the reliability of the valves in the high-viscosity fluids. Therefore, the found value of strokes n = 225,3 stroke / min (the third row of table. 2) should be treated with caution. To confirm that value an additional numerical experiment on a dynamic mathematical model is to be provided plus physical experiment.

Investigated pumps with traditional design schemes include slides to protect pistons and cylinder liners from the effects of shear forces – that increases their service life, but causes significant increase in the size and weight of the pumps.

Table 1. Dependences of Δ and δ on the ratio r / l

r/l	Z=3		Z=4		Z=5		Z=6		Z=7		Z=6*	
	Δ,%	**δ,%**	Δ,%	**δ,%**	Δ,%	**δ,%**	Δ,%	**δ,%**	Δ,%	**δ,%**	Δ,%	**δ,%**
0	14,03	27,41	32,5	61,9	4,97	9,87	14,03	27,4	2,52	5,03	3,45	6,85
0,1	19,09	36,54	32,5	61,9	6,2	12,23	14,03	27,4	3,1	6,16	8,87	17,3
0,2	25,1	47,27	32,5	61,9	7,55	14,82	14,03	27,4	3,73	7,39	12,54	24,4

Table 2. Technical Data Pulp pumps with different number of single-acting cylinders Z and ψ

Q, l/s	Z	d_s, m	D, m	ψ=0,0		ψ=0,1		ψ=0,2	
				S, m	n, str/min	S, m	n, str/min	S, m	n, str/min
28	3	0,205	0,168	0,223	112,7	0,246	102,4	0,268	93,9
	5		0,130	0,134	187	0,147	170	0,161	156,4
	6		0,119	0,223	112,7	0,223	112,7	0,223	112,7
	6*		0,119	0,112	225,3	0,144	174,3	0,173	145,9

To address this issue the constructive diagram of the double-shaft membrane-piston pump without the original slide was synthesized [15], as shown in fig. 1 (the kinematics of one of its mirrored halves – in fig. 2).

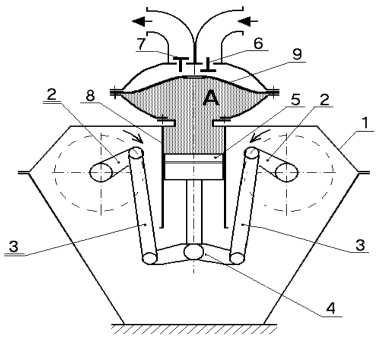

Fig.1. Half-constructed scheme of the pulp membrane-piston pump. No slide-block

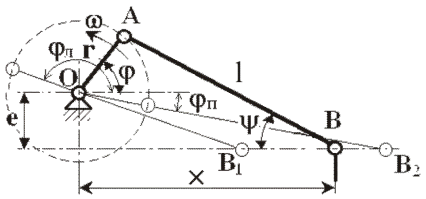

Fig.2. Kinematic scheme of the half of the drive mechanism

Two crank shaft 2 rotate synchronously in opposite directions, allowing the piston 5 reciprocates. Forces that arise in each half of the drive – normally to the direction of the piston motion – close and get mutually balanced on the beam 4. A cavity between the piston and the diaphragm is filled with oil that protects the cylinder-piston pair from abrasion and prevents the pumped liquid in the sump pump. Location of hydraulic parts inside the drive allows, as we will see, significantly reduces the weight of the machine.

Kinematic study of the drive mechanism [4] showed that at the **e / r > 0** and **r / l > 0**:

angles of the suction and discharge of the crank shaft may differ significantly from 180°;

so, dependences of the speed of the piston in the suction and discharge cycles and uneven fluid flow in the inlet and outlet pipes will be different;

the stroke of the piston S may be more than 2r, in contrast to the central drive mechanism, when e = 0 and S = 2r.

Comparing effect of the ratio dimensions of the drive r / l and e / r onto the change of the instantaneous pump flow is shown graphically in figure 3 and figure 4 according to its relative value for each cylinder and for both five-cylinder pumps (top – blowing, lower – absorption) at different **e / r** and **r / l**. Total relative feed pump differs not only by the oscillations amplitude, but by their frequency as well (for the second version – 2 times more).

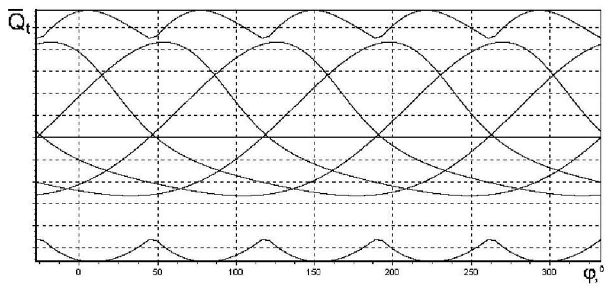

Fig. 3. The changes in the relative supply of the pump at z = 5, e/r = 2 and r/l = 0,3

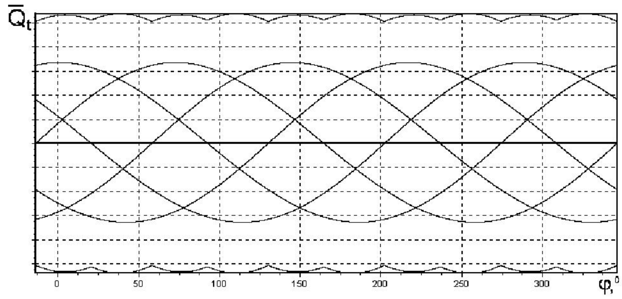

Fig. 4. Changes in the relative supply of the pump at z = 5, e/r = 2 and r/l = 0,05

The coefficients of unevenness of the flow and pressure rates at the inlet and outlet of each pump are shown in table 3. But in the second version they are several times less than in the first one as being respectively equal to the suction and discharge piping ($\bar{\Delta}_в = 3{,}22$; $\bar{\Delta}_н = 4{,}05$; $\bar{\delta}_в = 3{,}17$; $\bar{\delta}_н = 3{,}99$).

According to the table 3 data, for the off-center drivers **S > 2r** always is, what is most notably seen at higher meaning **e/r** and **r/l** (see line 3, table 3), which near of degeneration of the mechanism. But at that the discharge angle $\boldsymbol{\varphi_n}$ decreases sharply, while the coefficients of unevenness of the flow and pressure rates overly increases (as much as 5,2 and 15,7 – by comparison with version 2). Respectively the coefficients of unevenness of angular velocity and torque of the crank shaft increases, which is unacceptable. As for the step-uniform angular pitch cranks, even in machines with such a drive, it can also be effective (see fig. 5 and fig. 6), provided the rational choice of the values of **e/r** and **r/l**.

Table 3. Research results of the five-cylinder pumps without slider-blocks
With various ratios of the sizes of the drive components

Z	e/r	r/l	S/r	φ_d °	Δ_s,%	Δ_d %	δ_s %	Δ_d %
5	2	0,3	2,64	148,48	19,16	24,07	37,35	47,01
	2	0,05	2,07	178,52	6,04	6,04	11,97	11,97
	3	0,25	4,0	126,87	31,38	94,19	63,28	189,95

Table 4. Specifications of piston pumps with a crank drive for hydraulic coal with Q = 250 m3, p = 10 MPa and ŋp = 0.94

Design scheme of the machine	Z	n, str/min	D, mm	S, mm	Δ, %	δ, %	F, кН	L_{tr}, $^m/s$	M, t	$\bar{M}$, кg/кW
Duplex (double-acting cylinders)	2	68	240	400	44	85	450	0,9	38,2	55
Triplex and cup. valves **r/l = 0,15**	3	120	250	260	19,1	36,5	488	1,04	28,7	41,3
Triplex with ring. valves **r/l = 0,15**									24,3	35
Double-shaft triplex without slide-blocks **e/r=1, r/l=0,05**	3	125	250	240	17,6	34,7	488	1,17	15,9	22,7
Double-shaft without slide-blocks, with step-equal. angular. step cranks r/l = 0,05, e/r = 0,5	6*	157	200	150	11	21	292	1,17	17,1	24,4

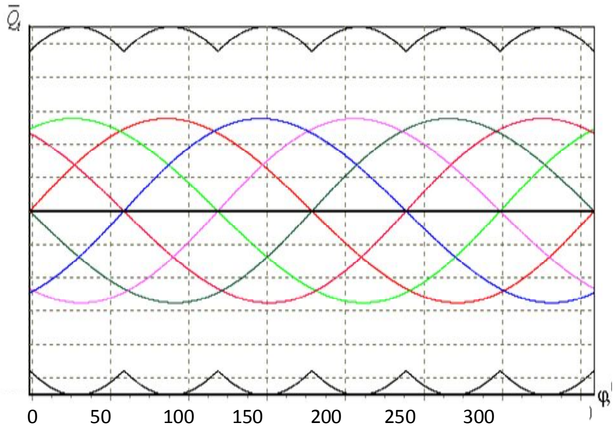

Fig. 5. Changes in the relative supply six-cylinder pump with a uniform angular step of cranks and r / l = 0,2

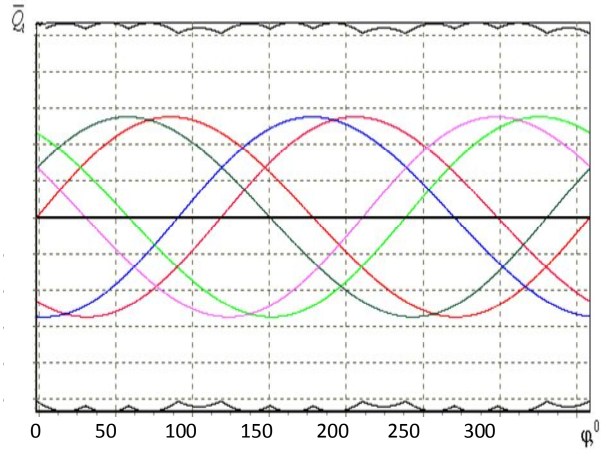

Fig. 6. Changes in the relative supply six- cylinder pump with e/r = 0,5, step-uniform angular step of cranks and r/l = 0,05

By the results obtained by numerical experiments were carried out five projects sketch of the pulp pumps at the same parameters of their work. The first three projects - the traditional design scheme (see table 4, lines 1-3), the last two - on the proposed scheme (see table 4, lines 4-5).

Specifications and performance of all the machines are listed in table 4. In a third embodiment poppet valves replaced with ring [14; 18], which led to a noticeable reduction in the weight of the pump. Besides, repeated decrease of the harmful volume will increase its delivery rate, especially when pumping gas-containing solutions, which is almost always the case. Versions 4 and 5 were performed with annular valves and seals.

In the table 4: η_d – delivery rate, F_p - the pressure on the piston, L_{tr} – road of pistons friction at a time, M – mass of the pump, $\bar{M}$ – the mass-to-pump-power.

Obviously, proposed technical solutions are superior to the now produced traditional design Pulp pumps by the parameters of uneven flow and pressure, especially the mass

CONCLUSIONS

The numerical experiment on the effect on the kinematics of the fluid and the degree of irregularity of its costs and the pressure in the inlet and outlet pipes piston pumps, the relative magnitude of the geometric dimensions of the main parts for the drivers of cars with conventional and synthetic design schemes.

The results obtained and executed settlement and design work can be recommended for development of advanced Pulp pumps adopts the proposed original design scheme. This should give preference to vehicles with an odd number of cylinders or even number of cylinders z = 6 *, but with a staggered-uniform angular step of cranks and values relations $e / r \leq 1$ and $r / l \leq 0{,}05$. If there is not any need to use an off-center of the drive crank mechanism ($e > 0$), it is better to use the center, which has $e = 0$.

Desire to increase stroke, significantly more than twice the radius of the crank through the use of non-central ($e > 0$) of the drive mechanism is futile due to a sharp deterioration in this unevenness of the pump and torque on its shaft, deterioration of the pump suction capacity due to a sharp increasing the speed of the fluid in the inlet pipe, and therefore the inertia pressure.

REFERENCES

1. Valume pumps for pumping of highly concentrated of hydromixture.// Review information. Series pulpbiuilding. – Moscow: CINTIHIMNEFTEMASH. - 1987. 48 p.
2. **Gurevich E.S. 1982.:** Possibilities research of piston pumps using for hydrotransport on long distance. Moscow: Machinebuilding industry,. - 112 p.
3. **Yunisov R.M. 1984.:** Research and substantiation of working characteristics of pulping pumps for main hydrotransport. Ph.D. dissertation. Moscow.
4. **Verzilin O.I. 1987.:** Modern drilling pumps. – Moscow: Mashinebuilding.– 256 p.
5. http://www.ronglii.com/indexru.html
6. www.lukoil.ru
7. www.derrick.ru
8. www.vzbt.ru/products/S
9. www.upetgroup.ro
10. http://www.rongliicom/indexru.html
11. uhkirf.com
12. http://www.compair.com
13. **Kossenko-Belinsky Yu. A, 2011.:** Piston Pulp pumps: monograph. / Lugansk: Izd VNU them. Dahl, - 211s.
14. **Kossenko-Belinsky Yu, 1997.:** Piston pumps for abrasive slurries: monograph. / Lugansk: Publishing House of VSU,. - 142s.
15. Pat. 21029 (Ukraine). Piston unit. Yu. A. Kossenko-Belinsky. 2007
16. **Roman Ray, Miroslava Ray, 2010:** The method for the definition of reduced coefficient of frictionin crank-connecting mechanism of Pump. Polish Academy of Sciences Branch in Lublin, Lublin University of Technology, Volodymir Dal East – Ukrainian National University of Lugansk. Teka: Commission of Motorization and Power Industry in Agryculture, Lublin University of Technology, Volodymir Dal East- Ukrainian National University of Lugansk. Volume XB. Lublin 2010. P. 125-128.
17. **Taras Zamota, Alexander Kravchenko, 2010.**: Improvement of tribotechnical characteristics of piston ring surface at running in. Polish Academy of Sciences Branch in Lublin, Lublin University of Technology, Volodymir Dal East – Ukrainian National University of Lugansk. Teka: Commission of Motorization and Power Industry in Agryculture, Lublin University of Technology, Volodymir Dal East- Ukrainian National University of Lugansk. Volume XB. Lublin. P .323 - 330.
18. **Kossenko-Belinskiy Yu., 2010.:** Determination of characteristics of circular distributor valve piston pulp pump. Polish Academy of Sciences Branch in Lublin, Lublin University of Technology, Volodymir Dal East – Ukrainian National University of Lugansk. Teka: Commission of Motorization and Power Industry in Agryculture, Lublin University of Technology, Volodymir Dal East- Ukrainian National University of Lugansk.Volume XA. Lublin. P.294-302.

ИСПОЛЬЗОВАНИЕ ЧИСЛЕННОГО ЭКСПЕРИМЕНТА НА СТАДИИ ПРОЕКТИРОВАНИЯ ПОРШНЕВЫХ ПУЛЬПОВЫХ НАСОСОВ

Юрий Косенко-Белинский

Аннотация. Приведены результаты численного эксперимента на кинематических математических моделях работы поршневых пульповых насосов с различными модификациями кривошипно-шатунных приводных механизмов и числом цилиндров. Даны рекомендации по рациональному выбору их количества, углового шага кривошипов и отношения их длины к длине шатуна, а также эксцентриситета механизма к радиусу кривошипа.

Ключевые слова. Поршневой пульповый насос, кинематическая модель, численный эксперимент, кривошип, ползун, шатун, неравномерность подачи и давления, угловой шаг кривошипов.