CSŐVEZETÉKEK ARMATURÁINAK CSÚSZÓ-ORSÓS, REDUKTOROS VILLAMOS-HAJTÁSAI ÉS A FEJLESZTÁS FŐBB IRÁNYAI

REDUCING GEARINGS OF ELECTRIC DRIVES OF PIPELINE FITTINGS WITH SLIDING SPINDLE AND PRIORITY DIRECTIONS OF CREATION OF THEM

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ABSTRACT

The article considers standard serial constructions of russian and foreign electric drives, and comparative analysis of their technical levels. It is created a gear electric drive of new technical level. The article also describes its distinctive signs and formulates a new concept of creation of multiturnaround electric drives.

INTRODUCTION

When creating first samples of pipeline fittings with electric drives with the power of less than 5 kW, in the 60-s of the previous century, companies AUMA (Germany), BIFFI (Italy), ROTORK (England), Tulaelectroprivod, Cheboksarsky works of electromechanical equipment (Russia) and others mostly paid their attention to reducing worm-gear drives. Their merits are: noiseless and smooth work; relatively good efficiency coefficient with gear ratios < 40; possibility of self-braking; compactability and low specific metal content (0,05…0,08) kg/Nm; well-developed methodology of synthesis, designing and producing using the standard machine-tool equipment, etc. Then a very important role was played by technically and technologically easy way of producing shaft of a worm wheel hollow, as well as a constructive opportunity to transmit drive moment from hollow shaft to the input of isolation valves.

Sustainable growth of installed power per employee and widening of torque of electro drives of pipeline fittings created several difficulties in improving the worm-type gears. There appeared some technical solutions where momentum from the worm-type wheel is transmitted into output cone or cylinder gear, thus creating combined worm-cone or worm cylinder motor-reducers. They allowed to increase installed power and to widen the limits of changing of torque and frequency of the drive rotation, but they also demanded constructional development, new division of gear ratio between input worm gear and output cylinder or cone steps, aimed at decreasing of worm gear ratio in the general constant gear ratio. In the time being they started to use new transfers: planetary [1], spiroid [2] and wave-like gears with flexible wheel or intermediate rolling elements [3]. However with introduction these transfers the problem of creation of electric drives didn't lose the urgency.

It speaks that the majority of the mentioned above power transmissions poorly adapt for structure of electric drives of new generation or don't adapt absolutely. The drive is still designed by a principle: «To each technological machine – the independent option». Therefore there is a big variety of designs of independent electric drives in branch (about 300).

The article considers driving equipment based on a new principle: «The group of identical or various technological machines with identical connecting forms and the sizes of flanges of a drive, but different options of parameters of its output torque and speed is served by one universal unified multiturnaround drive with multiple kinematic communications in one dimension». Such possibility appeared in connection with creation of a new version of gearings – multithread gearings with multipair convexo-concave contacts and non-inphase working gearings in power streams [4, 5]. These transfers at a design stage guarantee: multiple execution in one dimension; the highest reliability; any beforehand set resource of work and optimum dimensional and mass indicators (the specific weight can be lowered to

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$q_м$ = 0,01...0,03 kg/N · m and less at efficiency ≥ 0,95), – inaccessible to all other types of power transmissions.

COMPARATIVE ASSESSMENT OF TECHNICAL SOLUTIONS, PUT IN SERIAL REDUCING GEARINGS OF ELECTRIC DRIVES

When assessing different types of reducing gears, we should pay attention to the following factors: market demands and price limits of the electric drive produced; possibility to make qualitatively new mechanics on the operating serial machine tool and equipment; existence of techniques for design and production of new mechanics; desire of the management to increase technological and productive level; import substitution of foreign equipment on the domestic; the power - and the cost-effective use of resources and so on.

The comparative assessment of world market production is given in tab. 1 and 2 on some indicators of its technological level.

Table 1 Assessment of driving equipment on specific weight

Producing company	**Model of electric drive**	**Type of reducing gearing**	**Specific weight,** kg/Nm
GZ Elektroprivod, Russia	ГЗ-А (-Б, В, Г, Д) ГЗ-ВА (-ВБ) ГЗ-ВВ (-ВГ,-ВД)	Worm	0,05…0,1
Electro drive works «TOMZEL», Russia	ЭПЦ15000/20000/50000; «Angstrem»	Multithread planetary-rolling transmission with intermediate rolling elements	0,024…0,04
«ZPA Pecky», Pechky, Check Republic	*MODACT MOA*; *MODACT MOA OC*	Two-lined planetary reducer	0,086…0,1
CJSC «Tulaelectro-privod», Russia	ЭП4 (scheme 41)	Worm	0,113
AUMA, Germany	*SA*10.1	Worm-type power transmission	0,045...0,070
LLC «Sibirsky mashinostroitel», Russia	Tomprin	Wave three-row rolling reducer	0,024…0,05
GREATORK, China	*AVA* и *ZJ Series*	Conic	0,056
		Worm	
Bernard, Tecofi, France	*SRA* и *ST Series*	Conic	0,072
		Worm	
АБС, «ЗЭиМ», Cheboksary, Russia	ПЭМ-А, ПЭМ-Б, ПЭМ-В	Combined	0,092
Rotork, England	*MTV Series*	Worm	0,09
	IB Series	Conic	
Biffi, Italy	*ICON2000EC*	Worm –conic type	0,191

Table 2
Assessment of driving equipment on efficiency depending on type of power transmission and its gear ratio

№	**Kind of gear transmission**	**Efficiency coefficient at gear ratio**					
		4	7	10	15	40	≥ 100
1.	Cylinder (one or two step)	0,95	0,94	0,93	0,92	0,88	0,85
2.	Conic (one or two step)	0,94	0,93	0,92	0,90	0,85	0,80
3.	Hypoid (one or two step)	0,88	0,83	0,80	0,77	0,68	0,64
4.	Worm-like	0,74	0,75	0,72	0,66	0,45	0,25
5.	Spiroid	0,86	0,83	0,79	0,68	0,45	0,24
6.	Planetary « $2k-h$ » performed in three satellites (one or two steps) [3]	0,98	0,97	0,96	0,96	0,90	-
7.	Planetary « $3k$ » performed in three satellites [3]	-	-	-	-	0,9	0,85
8.	One-lined planetary gear « $k-h-v$ » [3]	-	-	-	0,7	0,6	-

9.	Multithread planetary-rolling transmission with intermediate rolling elements [5]	-	-	-	0,95	0,95	-
10.	Multithread gearings «$3k-2g-h$» with multipair convexo-concave contacts and non-inphase working gearings [1, 2]	-	-	-	0,95	0,95	0,95
11.	Combined:						
	– cylinder-worm	-	-	-	-	0,43	0,24
	– cylinder-planetary	-	-	-	-	0,85	0,8
	– worm-cylinder	-	-	0,7	0,64	0,44	0,24
	– worm-conic	-	-	0,7	0,64	0,44	0,24
	–cylinder- multithread	-	-	-	0,95	0,95	0,94
	–worm doublestaged	-	-	-	-	-	0,15

The analysis of the given materials shows, that leading developers of pipeline fittings with electric drives reach competitive indicators on specific weight only due to unduly high load of mating profiles in gear cinematic pairs, when real surface stress amounts to (1500 … 2000) MPas and there is high possibility of small-cycle fatigue gear damages. It is inadmissible for a reason of safety of pipeline [2]. In multiturnaround electric drives of new generation contact tension in the gear pairs shouldn't exceed 1000 MPas, and flexural – 600 MPas. This condition isn't carried out by all known drives containing one-line power transmissions (tab. 1 and 2).

MULTITHREAD REDUCING GEARING WITH MULTIPAIR NON-INPHASE WORKING GEARINGS

According to Government Decree of Russian Federation № 218 Tula State University in collaboration with JSC "Michurinsky Plant "Progress" developed electric drive 7МРЭП-88-10/115-00.000 for all-purpose pipe fitting with the use of multithread planetary transmission of «$3k-2g-h$» type as power transmission [4]. Kinematic communications of reducing gear of that type are shown on the figure 1. Ease and durability of a design of the multiturnaround electric drive of the third dimension are visible.

Power transmission «$3k-2g-h$» due to non-inphase working gearings doesn't limited by the value of torque and speed outputs and it allows making a number of target parameters on speed and the torque in one preset third dimension ($a_w = 80$ mm) with repeated reservation of the torque. That isn't provided by any other contemporary power transmission [4, 5]. In table 3 are provided a technical characteristics of a reducer of this electric drive.

Efficiency coefficient of transmission equals 0,95 for all gear ratio indicated in the table 3 at the module of 1 mm.

Teeth number of gear units in two drive stages of multithread transmission due to non-inphase working gearings shouldn't be divisible by the number of satellite gears and shouldn't have common multiples, herewith sums of teeth number of central gear wheels of two drive stages should always be equal among themselves and be divisible by the number of satellite gears under condition of its common assembling [1, 2].

Fig. 1. A two-step planetary reducer of small module with 12 streams of type « $3k-2g-h$ » with an open cover of a reducer of the multiturnaround electric drive of the third dimension with a connecting flange type B (□ 200×200 mm)

Table № 3
Technical specifications of planetary reducing gear « $3k-2g-h$ » [5]
of the multiturnaround electric drive of the third dimension

Parameters	Units of measurement	Implementations				
		00/115	01/77	02/58	03/46	04/38
The capacity of the electric motor	kW	3,0				
Interaxal distance	mm	80,0 (88,0)				
Nominal frequency of rotation of a rotor of the electric motor	min^{-1}	1820				
Gear ratio of power transmission	–	115	76,6	57,5	46	38,3
Gear ratio of manual drive	–	220,5	134,5	102	82	69
Nominal output torque	Nm	1725	1150	865	700	580
Maximum impermanent output torque	Nm	4250	2800	2100	1700	1400
Nominal frequency of rotation of output shaft (nut)	min^{-1}	16	24	32	40	48

The value of safe modulus of straight tooth gearings of multithread transmission is defined by bending durability of its teeth in critical section defined by State Standard Ru № 21354-87 "Gear cylindrical evolute transmissions of external toothing. Stress analysis".

The least strong on a bend are satellite gear teeths g_2 of output drive stage « $a_2-g_{2i}-b_2$ » due to a particular form and size in critical section (fig. 1). It is smaller gear in two working gearings « a_2-g_{2i} » and « $g_{2i}-b_2$ »).

Formula for defining of safe gearing modulus of transmission « $3k-2g-h$ » is received in [4] and looks like:

$$m_F \geq K_m \sqrt[3]{\frac{T_{b_2} Y_{F_{g2}}}{z_{g_2} z_{b_2} a_c a_p \varepsilon_\alpha \psi_{bd} [\sigma_{F_{g_2}}]}},$$

wherein: K_m – integrated coefficient considering average conditions of operating gearing formation, that equals to $K_m = 14$ for a straight tooth transmissions $K_m = 11{,}2$ – for a helical gear transmissions; T_{b_2} - torque moment of transmission output element; Y_F – coefficient of tooth form recommended to define by dependence of teeth number z_{g2} и coefficient of displacement of original profile of an instrument during their grooving; z_{g_2}, z_{b_2} – number of teeth of satellite gear and output gear; a_c – number of satellite gears in a row; a_p – number of satellite gear rows in a drive stage; ε_α - transverse contact ratio; ψ_{bd} - coefficient of satellite gear face g_2; $[\sigma_{F_{g2}}]$ – allowed stress of gear tooth material of satellite gear, $[\sigma_{F_{g2}}]$ =400…600 MPa.

Obtained value of gearing modulus rounds off to its standard larger value defined by State Standard № 9563-80.

Value of operating gearing modulus of multithread transmission obtained from conditions of gear teeth flexing strength should be tested for durability by contact stress of its circumpolar surface. As far as contact stress in conjugate tooth profiles of two units are the same both for small gear and large one, the analysis is made for a gear unit which working stress $[\sigma_H]$ is less, in other words it's made for an output central gear it's made of less durable material in comparison with satellite gear.

Formula for design calculation of gearing modulus by surface endurance obtained with consideration for contact stress in circumpolar surface of mating teeth of operating gearing has this form [4]:

$$m_H \geq K_d \sqrt[3]{\frac{T_{b_2} K_{H_\alpha} K_{H_\beta} K_{H_v} (z_{b_2} - z_{g_2})}{\psi_{bd} a_c a_p \varepsilon_\alpha z_{g_2}^2 z_{b_2}^2 [\sigma_{H_{b_2}}]^2}},$$

wherein: $K_d = \sqrt[3]{2(Z_E Z_H Z_\varepsilon)^2}$ – integrated design coefficient which in design calculations equals $K_d = 770$ for a steel straight tooth gear and $K_d = 675$ for a helical gear; Z_E – coefficient considering mechanical characteristics of gear units' materials (elastic modulus and Poisson's ratios), for steel wheels $Z_E = 190$ MPa; Z_H - coefficient considering a form of teeth mating surfaces in gearing pole; Z_ε – coefficient considering total length of contact lines; $K_{H\alpha} K_{H\beta} K_{Hv} = K_H$ – load coefficient considering specific character of its imposition including external and internal dynamics and load distribution through the length of contact line; $[\sigma_{H_{b_2}}]$ - allowed stress of gear tooth material of output gear b_2.

From these formulae it follows that for decreasing of tensions and deformations in highest centroid pairs of kinematic elements of multithread transmission it's necessary to increase the number of power streams, contact ratio in operating gearing and teeth number of output central gear (to the maximum) and satellite gears (to the minimum). It ought to be mentioned that coefficient of gear wheels' width ψ_{bd} should be taken up within limits of $\psi_{bd} \cong 0{,}3...0{,}35$ which guarantee embed ability of lower kinematic pair of third class in satellite gears' crown.

The results of these formulae evaluation for different values of loading torques and numbers of power streams under the same parameters of gear units and allowed stress in all cases give almost the same value of modulus that testify to strength balance of gear units against teeth breakdown and pitting corrosion of its contact destruction [4]. It is important that there is an application possibility of using in drives with large torque gears with small modules $m = 0{,}8; 1{,}0$ и $1{,}25$. That implies also that only multithread power transmissions due to coaxiality of an input and output don't place limitations on dimensions of reduction gear motors. All other power transmissions don't have such important advantage.

Universality and unification of multiturnaround electric drive with multithread transmission «$3k-2g-h$» are proved by analytical parametrical series of output parameters by speed and torque in one specified dimension (table 3), which defined by dimension with a connecting flange.

General view of the electric drive of third dimension based on analyzed reducing gear with a gear ratios 230; 115; 77; 58; 46 and 38 is demonstrated on the figure 2 and specifications of such drives enumerated in the table № 4.

CONCLUSION

The analysis of production which represented on the world market by the producers of electric drives for pipeline fittings allows singling out the next factors characterizing technical level of reducing gears.

1. Companies which are working at the world market for a long time, traditionally use worm gearings as power transmissions. Usually transmissions' upgrading is connected with usage of combination of worm gearing with cylindrical, planetary or cone gearing and with dangerous increase of allowed tension in contact pairs of teethes for the purpose of minimization of dimensions and weight.
2. Companies which were established in recent decades produce reducing gears with spiroid or harmonic gearings with intermediate rolling elements. This fact is connected not so much with best specifications of those gearings as with a decrease in tension in contact pairs of teethes to 1000…1500 MPa and with scientific interest of developers of this production. These transfers concede on indicators of a technological level to multithread drives with non-inphase multipair convexo-concave gearings.
3. Reducing gear with planetary transmission of «$3k-2g-h$» type developed by Tula State University in collaboration with JSC "Michurinsky Plant "Progress" compares favorably with reducing gearings of producing electric drives by the fact that of coaxial arrangement of input and output. It doesn't limit dimensions of electric drives, practically covering every dimension-type by only five dimensions with the connecting sizes АК, Б, В, Г and Д or with connecting flanges of F10; F12; F30 and F35 on ISO 5210.
4. Universality and unification of such reducing gear are confirmed by corresponding analyses and electric drive test [6].
5. Tula State University and JSC "Michurinsky Plant "Progress" joint project of new generation electric drive for pipeline fitting of general purpose permits reduction of working expenses as this project was commenced with a focus on the current technical process and on engineering manpower.

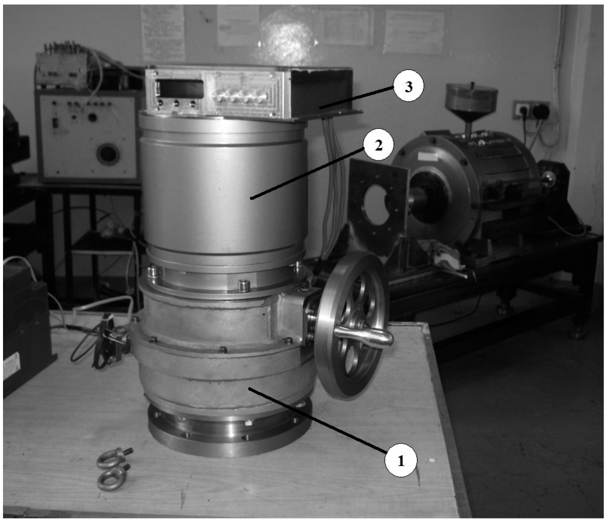

Figure 2.
General view of multiturnaround multithread electric drive of third dimension 7МРЭП-88-230.00.000 with discrete change of gear ratio: 1 – multithread reducing gear; 2 – engine; 3 – control unit

Table 4
Specifications of electric drive 7МРЭП-88-230.00.000

<table>
<tr><th rowspan="2">№</th><th rowspan="2">Parameters</th><th rowspan="2">Unit of mea-sure-ment</th><th colspan="14">Implementations</th></tr>
<tr><th colspan="2">-00/115</th><th colspan="2">-01/77</th><th colspan="2">-02/58</th><th colspan="2">-03/46</th><th colspan="6">-04/38</th></tr>
<tr><td>1</td><td>2</td><td>3</td><td colspan="2">4</td><td colspan="2">5</td><td colspan="2">6</td><td colspan="2">7</td><td colspan="6">8</td></tr>
<tr><td>1</td><td>Engine type</td><td>-</td><td colspan="14">ДБМ 150-4-1.5-3</td></tr>
<tr><td>1.1</td><td>Power</td><td>kW</td><td colspan="14">≤ 2.18</td></tr>
<tr><td>1.2</td><td>Nominal frequency of rotation of a rotor</td><td>min^{-1}</td><td colspan="14">1820</td></tr>
<tr><td>1.3</td><td>Nominal torque</td><td>Nm</td><td colspan="14">15.0</td></tr>
<tr><td>1.4</td><td>Starting torque</td><td>Nm</td><td colspan="14">37</td></tr>
<tr><td>2</td><td>Transmis-sion type</td><td>-</td><td colspan="14">7РПМ-88/230</td></tr>
<tr><td>2.1</td><td>Gear ratio of power transmis-sion</td><td></td><td colspan="2">115</td><td colspan="2">76.6</td><td colspan="2">57.5</td><td colspan="2">46.0</td><td colspan="6">38.3</td></tr>
<tr><td>2.2</td><td>Gear ratio of manual drive</td><td>-</td><td colspan="2">200.5</td><td colspan="2">134.35</td><td colspan="2">102</td><td colspan="2">82</td><td colspan="6">69</td></tr>
<tr><td>1</td><td>2</td><td>3</td><td colspan="2">4</td><td colspan="2">5</td><td colspan="2">6</td><td colspan="2">7</td><td colspan="6">8</td></tr>
<tr><td>2.3</td><td>Nominal output torque</td><td>Nm</td><td colspan="2">1725</td><td colspan="2">1150</td><td colspan="2">865</td><td colspan="2">700</td><td colspan="6">580</td></tr>
<tr><td>2.4</td><td>Output maximum imperma-nent torque</td><td>Nm</td><td colspan="2">4250</td><td colspan="2">2800</td><td colspan="2">2100</td><td colspan="2">1700</td><td colspan="6">1400</td></tr>
<tr><td>2.5</td><td>Nominal frequency of rotation of output shaft (nut)</td><td>min^{-1}</td><td colspan="2">17.4</td><td colspan="2">26.1</td><td colspan="2">34.8</td><td colspan="2">43.5</td><td colspan="6">52.2</td></tr>
<tr><td>3</td><td>Type of isolation valve</td><td>–</td><td colspan="14">wedge with sliding spindle</td></tr>
<tr><td>3.1</td><td>Pipe di-ameter,</td><td>mm</td><td>250</td><td>500</td><td>300</td><td>400</td><td>100</td><td>400</td><td>150</td><td>300</td><td>80</td><td>100</td><td>200</td><td>200</td><td>250</td><td>350</td></tr>
<tr><td>3.2</td><td>Pressure,</td><td>MPa</td><td>6.3</td><td>1.6</td><td>6.3</td><td>4.0</td><td>25.0</td><td>1.6; 2.5</td><td>16.0</td><td>1.6-4.0</td><td>16.0</td><td>16.0</td><td>4.0</td><td>6.3</td><td>2.5; 4.0</td><td>1.6</td></tr>
<tr><td>3.3</td><td>Spindle diameter</td><td>mm</td><td>50</td><td>50</td><td>50</td><td>50</td><td>44</td><td>40</td><td>40</td><td>40</td><td>30</td><td>30</td><td>30</td><td>40</td><td>40</td><td>40</td></tr>
<tr><td>3.4</td><td>Spindle movement</td><td>mm</td><td>288</td><td>526</td><td>344</td><td>435</td><td>120</td><td>426</td><td>185</td><td>342</td><td>84</td><td>120</td><td>222</td><td>222</td><td>288</td><td>378</td></tr>
<tr><td>3.5</td><td>Spindle thread (trapezoid)</td><td>-</td><td colspan="2">Tr50x8</td><td colspan="2">Tr50x8</td><td>Tr44x8</td><td>Tr40x6</td><td colspan="2">Tr40x6</td><td colspan="3">Tr30x6</td><td colspan="3">Tr40x6</td></tr>
</table>

3.6	Nut’s turning number during valve shift	Revol.	36	66	43	54	15	71	31	57	14	20	37	37	48	63
3.7	Time of fittings’ opening/ closure	sec.	216	396	129	162	30	142	47	86	17	24	45	45	58	76
3.8	Attachment flange’s dimension ISO-5210	-	F25	F25	F25	F25	F16	F16	F16	F16	F14	F14	F14	F16	F16	F16
3.9	Required torque moment on nut	Nm	850	990	1500	1460	880	800	700	780	230	280	290	460	510	450
3.10	Required frequency of rotation of nut	min^{-1}	14	24	21	24	39	36	48	40	50	50	51	51	48	48
3.11	Real factor of safety by torque moment	-	2.03	1.74	0.8	0.8	0.96	1.06	1.0	0.9	2.48	2.04	1.96	1.24	1.1	1.27
3.12	Required boost coefficient by speed	-	0.9	1.5	0.9	1.0	1.23	1.13	1.2	1.0	1.05	1.05	1.07	1.07	1.0	1.0

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