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Problem of Non Proportional Flow of Hydraulic Pumps Working with Constant Pressure Regulators in Big Power Multipump Power Pack Unit in Open System

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Abstract: The non-proportional flow problem of hydraulic pumps with constant pressure regulator in case of simultaneous work in Power Pack Unit with advanced multipump structure in open supply system is presented in the paper. Main parts of hydraulic power pack unit with multipump structure, mounted on board of modern product and chemical tankers and pump constant pressure regulator DP type, are described. Coefficients of non-proportional pump flow phenomena are defined. Analysis of non-proportional flow problem referring to four hydraulic axial piston pumps simultaneously working with pressure constant regulator in Power Pack Unit in open system is presented. The idea of non-proportional pump flow phenomenon reduction is discussed. Conclusions are presented.

Keywords: constant pressure regulator; hydraulics; multipump structure; open system; power pack unit; product and chemical tankers

1 INTRODUCTION

In marine technology hydraulic central loading systems are used very often [5, 6, 14]. As example are big power hydraulic central loading systems, mounted on board of modern product and chemical tanker or oil platforms, for powering of submerged cargo and ballast pumps [7, 8, 11, 12, 13, 18]. The power of these hydraulic systems reaches a level of 1000 kW or more. On product tanker B578-I 'Helix' (built by Szczecinska Shipyard from Poland for SHELL Company), the hydraulic power pack unit, consisting of five hydraulic axial piston pumps A4VSO500 Rexroth Bosch type [19], has total power of 2200 kW installed (Fig.1a). On described ship every cargo tank was equipped with the separate submerged FRAMO cargo pump with hydraulic drive [12, 13, 17]. The hydraulic drive was installed as safer in comparison to solutions with electric motion, from the reason of working in dangerous space, especially in respect to explosive.

For supplying twenty-two independent FRAMO cargo pumps, the hydraulic central loading system with three main lines was built: the main pressure line P, return line R, and leakage line L, running along the all open deck. All receivers of the hydraulic energy, the cargo group, ballast pumps, mooring and anchor mooring winches, the deck crane, and also the bow thruster (with power of about 1300 kW) including, were connected to the hydraulic central main lines in parallel. Hydraulic power packs used for supplying such hydraulic central loading systems must have big power. The structure of such hydraulic power packs should be adapted to the working conditions [9]. Many hydraulic energy receivers, supplied from the central system are exploited at the load or flow less than nominal performance conditions, defining the structure size. This depends on the destination of devices with the hydraulic drive and on the coefficient of the simultaneousness of work of each receiver, supplied from the central loading system. Described situation is typical for most of industrial applications. The power and flow of the largest hydraulic axial piston pumps produced in the world is limited [19]. Therefore, the structure of these hydraulic power packs of large power must be designed as multi-pump structure. This fact requires the pump unit designer to make a decision. Is it better to make a pump unit with smaller number of hydraulic main pumps but with greater individual each power or vice versa with possibly more pumps but with less individual each power. In the case of large power pump units (e.g. greater than 100 kW), the presented problem is important from technical and financial side [6]. There are several criteria to assess the problem. One of the most important is the criterion of efficiency of the hydraulic power packs at the normal exploitation conditions of the hydraulic central loading system. Next problem is loading conditions of hydraulic pumps working simultaneously in one big power pack [20].

There is very limited literature describing the non-proportional flow problem of hydraulic pumps working parallel and simultaneously in one hydraulic power pack unit. Several investigators have presented the analysed hydraulic central power pack with multipump structure as a single-pump with capacity equal to simple multiplication of some smaller standard pump [4, 16, 21]. Such simplified models of the power packs move to the analysis of the single big hydraulic pump, not as group independent smaller pumps. In the same method, Mayr [4] and Darling [16] described the central Power Packs mounted on decks of sea vessels. Nollau and Helle [21] have tried to investigate the dynamics problems of central loading systems. However, in the paper are reduced the model of hydraulic central power pack unit to simply form with one hydraulic pump structure. Later, Banaszek et al. [5-9, 12, 13] have described problems with control of big power hydraulic power pack units with multipump structure. He presented solutions of constant pressure central loading systems mounted on board of modern product and chemical tankers. Author has analysed central power pack with some smaller pumps inside, equipped with constant pressure flow regulators.

In practice different adjustments of regulators move to different flows of individual pumps during simultaneous work in one power pack unit. The non-proportional flow problem creating in such power pack structure and having the influence on total output flow and power is discussed in present paper.

2 PRESSURE CONSTANT PUMP REGULATOR

In hydraulic central loading systems on modern product and chemical tankers, hydraulic main pumps as a part of hydraulic central power pack are equipped as a rule into pressure constant regulators. Their function consists in such adapting of the current flow of the regulated pump in relation to the current consumption of oil by supplied hydraulic central loading system, to keep constants, settled by the operator of the system, the service pressure on the way of the change of the geometrical volume of the working hydraulic pump [1-3]. A typical example of above mentioned elements are regulators of the DP type produced by Bosch-Rexroth Group/Germany, determining one from a standard equipment of axial piston pumps with variable displacement, A4VSO. Pumps of this type are often used in central loading systems of large power. The schematic diagram of the constant pressure regulator is shown in Fig. 1b. The mechanism of the inclination of the pump swash plate disk consists of the master servomotor 1 with shaft, supported by the additionally mechanical spring 9 and by the pressure, generated by the hydraulic pump. On non-piston, side of the servo motor cylinder acts the pressure of oil swimming from the control system by the four shaped master distributors 2. The pressure on the inflow of the distributor is fixed by the set of the ruff 3 and the relief valve 5, fulfilling the adjustment of the pressure controller rule p_{str}. The pressure this affecting the slider shaft in the extreme area of the distributor together with the spring 6 causes such steering of the master distributor 2 that the shaft of the master servo motor 1 will bend the resisting disk of the pump up to here, till the pressure generated by the pump will not draw up with the adjusted value of the pressure on the relief master valve controlled of a adjustment of the regulator [10, 15].

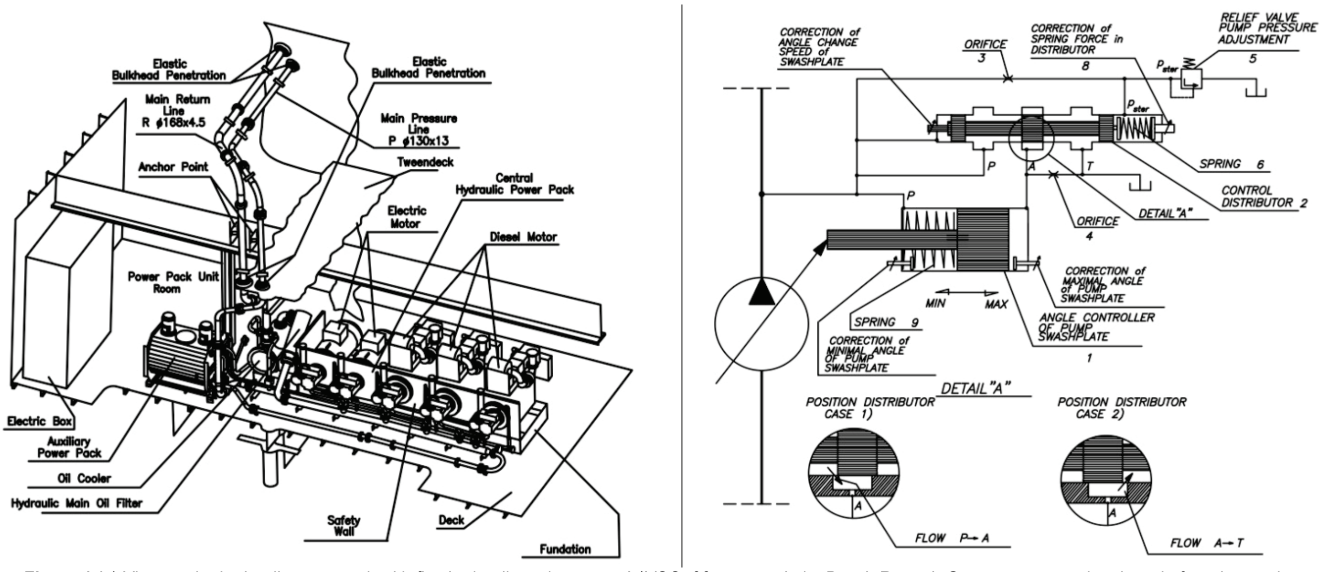

Figure 1 (a) View on the hydraulic power unit with five hydraulic main pumps A4VSO500 type made by Bosch Rexroth Germany mounted on board of product tanker 'Helix'; (b) Pressure constant pump regulator DP type made by Bosch-Rexroth Germany

However, in practice on ships deck, when we use the large power pump, very seldom we have a situation in which the operator of pump system possesses the possibility of measurement of the real pump driving moment, the rotational speed, even the flow of the pump. Therefore to our calculations one founded that performance characteristics of constant pressure regulators of pumps No. 1, No. 2, No. 3 and No. 4, comply with diagrams presented in Fig. 2. All hydraulic pumps were connected parallel according to the schema in Fig. 3. In compliance with the procedure of pumps regulation, common pressure adjustments of all constant pressure regulators became settled at the working pressure of all pumps:

$$p_p\big|_{\alpha=14,5^\circ} = 24.8 \text{ MPa} \tag{1}$$

It complies with results of pressure adjustment of central hydraulic power pack working pressure installed on product tankers B573-I/II class.

3 COEFFICIENTS OF NON-PROPORTIONAL PUMP FLOW IN MULTIPUMP POWER PACK UNIT

For the purpose of analysis of the flow of particular pumps in the multi pump type power pack unit problem was necessary to defining of measures describing above mentioned occurrence. Accordingly one found that analysed multipump power pack unit was composite from hydraulic pumps of the type axial piston about the variable displacement, the same type and the nominal size, equipped into constant pressure regulators "multipump" power pack unit was created from axial piston pumps with variable displacement, the same type and the nominal size like equipped into constant pressure regulators[12]. The above foundation complies with the structure of most multipump power pack units of thelarge power, mounted on deck of modern ships. For the purpose of generalizing of led out dependences and equations, one introduced the following undimensional parameters:

- the coefficient of pump relative load $\overline{p}_p$:

$$\overline{p}_p = \frac{p_p - p_{p0}}{p_{pnom}} = \frac{\Delta p_p}{p_{pnom}} \tag{2}$$

where: p_{pnom} – the nominal pump working pressure

- the current geometrical volume adjustment of the working volume pump e_p:

$$e_p = \frac{q_{pt}}{q_{pt\,max}} \approx \frac{q_{pgeom}}{q_{pgeom\,max}} \approx \frac{\alpha_p}{\alpha_{p\,max}} \tag{3}$$

where: q_{pt}, q_{pgeom} – the current theoretical and geometrical working volume of hydraulic pump, referred to one turnover of the drive pump shaft; $q_{pt\,max}$, $q_{pgeom\,max}$ – the maximum theoretical and geometrical working volume of hydraulic pump, referred to one turnover of the drive pump shaft; α_p, $\alpha_{p\,max}$ – the current and maximum swashplate angle of the axial piston pump variable displacement type the relative pump flow $\bar{Q}_p$:

$$\bar{Q}_p = \frac{Q_p}{Q_{pt\,max}}, \tag{4}$$

where: $Q_{pt\,max}$ – the theoretical pump flow at $e_p = 1.0$ and at the nominal speed of the electric motor at the zero pressure drop between the pump inflow and the outflow:

$$Q_{pt\,max} = q_{pt\,max} \cdot n_p \left| \begin{matrix} e_p = 1.0 \\ \Delta p_p = 0 \end{matrix} \right., \tag{5}$$

where: $n_p \left| \begin{matrix} e_p = 1.0 \\ \Delta p_p = 0 \end{matrix} \right. = n_{p0}$ – the rotational speed of the drive electric motor, at the zero pressure drop between the pump inflow and the outflow and full adjustment of the working geometrical volume $e_p = 1.0$.

At the foundation of the simultaneous work of these pumps, the average flow of every working pump at the pressure p_p of the multipump power pack unit carries out:

$$\bar{Q}_{pp}(p_p) = \frac{\sum_{i}^{m_p} Q_{pi}(p_p)}{m_p}, \tag{6}$$

where: $\bar{Q}_{pp}(p_p)$ – the average real flow of the single hydraulic pump in the power pack unit; $Q_{pi}(p_p)$ – real flow of i – pump; m_p – number of hydraulic pumps in power pack unit.

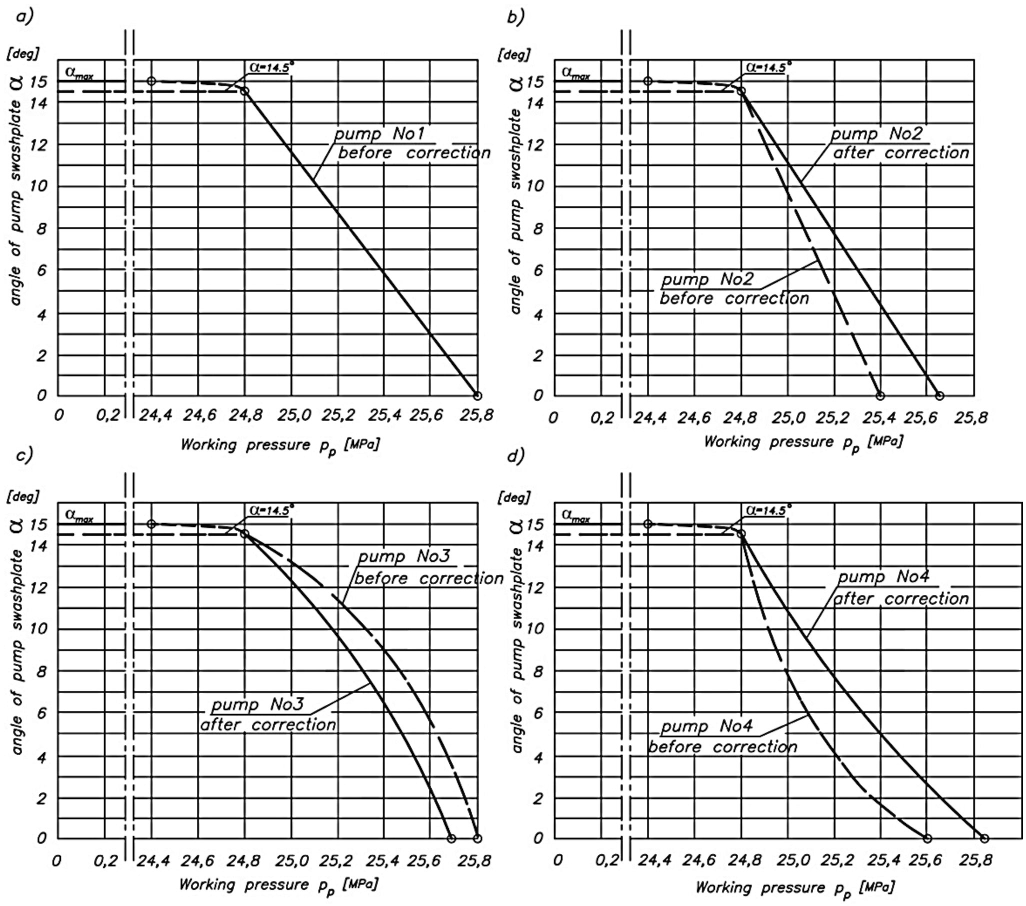

Figure 2 Performance characteristics of A4VSO500 (Made by Bosch Rexroth/Germany) axial piston pump pressure regulators DP type, before and after correction, mounted in analysed four-pump hydraulic power pack unit (a) Pump No. 1, (b) Pump No. 2, (c) Pump No. 3, (d) Pump No. 4.
The source: the own elaboration

In consequence of different performance constant pressure pump regulators, at the given common working pressure of the power pack unit, each pump can work at different adjusted angles of swashplates and values of their theoretical geometrical volume and the theoretical flow. So both the value of average the flow of the single pump in the power pack and flow of each pumps can differences themselves in relation to themselves in the dependence from the working pressure of the power pack and adjustments of the constant pressure regulators.

Based on given foundations can be defined the coefficient of the non-proportionality of the flow of the single pump in multipump power pack unit δ_{pi} as follows:

$$\delta_{pi}(p_p) \overset{def}{=} \frac{\left|\bar{Q}_{pp}(p_p) - Q_{pi}(p_p)\right|}{\bar{Q}_{pp}(p_p)} \cdot 100\% \quad , \tag{7}$$

The value above mentioned coefficient determines the measure of the non-proportionality of the current flow of the given pump in the pump unit, i.e. the measure of the deviation of her flow from the average flow definite for the all power pack unit.

A measure of non-proportionality phenomena of the flow of all power pack unit will be the value of the coefficient of the non-proportionality of the flow of the feeding whole pump unit δ_{pzp} defined in the following form:

$$\delta_{pzp}(p_p) \overset{def}{=} \frac{\sum_{i=1}^{m_p}\left|\bar{Q}_{pp}(p_p) - Q_{pi}(p_p)\right|}{m_p \cdot \bar{Q}_{pp}(p_p)} \cdot 100\% \overset{def}{=} \frac{\sum_{i=1}^{m_p}\delta_{pi}(p_p)}{m_p} \tag{8}$$

4 ANALYSIS OF NON-PROPORTIONAL FLOW PROBLEM OF FOUR HYDRAULIC AXIAL PISTON PUMPS SIMULTANEOUSLY WORKING WITH PRESSURE CONSTANT REGULATOR IN POWER PACK UNIT IN OPEN SYSTEM

For the purpose of the non-proportional pump flow problem in multipump Power Pack Unit analysis, one accepted the hydraulic Power Pack Unit with four variable displacement axial piston type pumps (the same type and the nominal size), equipped into constant pressure regulators DP type. It is a standard for big Power Packs mounted on board product and chemical tankers, especially built in Szczecin Shipyard/Poland [20]. The analysis was executed on the way of the numerical simulations, accepting DP constant regulators performance characteristics of each pump No. 1-4 according to Fig. 2(a-d). For simulations one founded the parallel work of all pumps in the open system (see Fig. 3). For the purpose of the analysis execution one introduced the following physical general suppositions:

- All physical processes in pumps and in the hydraulic system are pseudo stationary,
- All processes in hydraulic system are isothermal or isentropic,
- Kinetic energy of the fluid in hydraulic system is neglected,
- Whole hydraulic system, including all pumps and equipment is fully deaerated and during work cavitation processes are not present,
- Working fluid (hydraulic oil) conditions comply with requirements according to the Newton Formula, during the flow process the kinematic viscosity of the fluid is constant and equal to nominal value, recommended by hydraulic pump maker,
- All hydraulic pumps are the same size, axial piston, variable displacement type , with constant pressure DP regulators according to Fig. 2 performance characteristics (A4VSO500 DP made by Bosch Rexroth Germany,
- Main safety relief valve is installed close to the hydraulic pumps in Power Pack Unit, it is tight, pressure drops between pumps including Relief valve are neglected,
- Pressure drops on suction side of hydraulic pumps are neglected.

Hydraulic diagram of analysed Multipump Power Pack Unit is shown in Fig. 3. All hydraulic pumps (No.1, No. 2, No. 3, No. 4) work simultaneously and parallel in Power Pack Unit comply with performance characteristics according to Fig. 2(a-d). As a result one received diagrams of dependencies of relative real flow of in function of working pressure p_p for whole Multipump Power Pack Unit at simultaneously work all hydraulic pumps No. 1-4 (see Fig. 4).

Above mentioned dependence was compared with analogical Multipump Power Pack Unit diagram consists of four ideal same hydraulic pumps with linear performance characteristics according to Pump No. 1 (see Fig. 2a). We can observe that despite apparently approach together both Power Packs flow characteristics (ideal 4 × pump No. 1 and real) are between differ [14]. It is the result of different adjustments of swashplate of pumps according to performance characteristics of pressure constant regulators during the parallel work. Therefore, flow of pumps because of different adjustments of the geometrical working pumps volume are differ (at angles of pump swashplates smaller as common adjustment α_1, α_2, α_3, α_4 < 14.5°). Therefore already at the working pressure of the analysed Power Pack Unit p_p = 25.2 MPa (Point B at diagram Fig. 4), the main pump No. 4 creates only just 14.2% flow of the whole Pump Unit. Apparently results refer to theoretical flows of remaining pumps are as follows: pump No. 3 till 39.2%, pump No. 1 - 30.0% and pump No. 2 - 16.6% (see Tab. 1). It is clearly shown that pumps No. 1 and No. 3 are more loaded in generation of common flow referring to remaining pumps. Phenomena of non-proportional flow of particular pumps in Pump Unit are more clearly visible at other working pressures. For example at working pressure p_p = 25.6 MPa. Then proportional flow of pumps No. 1 and No. 3 carries out properly: No. 1 - 33.7% and No. 3 - 65.9%, at the flow of pump No. 4 only 0.4%. In this time hydraulic pumps No. 2 adjusted on the zero value of the geometrical working volume (α = 0) and does not take the participation in generating of the common flow of the Power Pack Unit [15]. Above mentioned observations are visible referring to values of non-proportional flow coefficiences in function of working pressure all analyzed pumps and whole Power Pack Unit. For working pressure of the analysed Power Pack Unit p_p = 25.2 MPa (Point B at diagram Fig. 4), the coefficient of the non-proportionality of the flow in multipump Power Pack Unit is apparently as follows: pump No. 1 δ_{p1} = 19.87%, pump No. 2 δ_{p2} = 33.43%, pump No. 3 δ_{p3} = 56.86% and pump No. 4 δ_{p4} = 43.29%. It creates the coefficient of the non-proportional flow in completely multipump power pack unit δ_{pzp} = 38.36%. We can observe in Tab. 1, values of the non-proportional flow referring to pump No. 3 are distinct higher as others and can have values higher even 100% level. It is result of situation (in range of working pressure p_p = 25.4 ÷ 25.7 MPa), where at working hydraulic pumps No. 2 and No. 4 with null or strongly low adjustments of geometrical working pump volume α_2, α_4 ≈ 0°) , distinct part of Power Pack Unit flow is generated by only pump No. 3. For example at working pressure p_p = 25.6 MPa the coefficients of the non-proportional flow in multi pump Power Pack Unit are apparently: pump No. 1 δ_{p1} = 34.8%, pump No. 2 δ_{p2} = 100.00%, pump No. 3 δ_{p3} = 163.44% and pump No. 4 δ_{p4} = 98.26%. It creates the coefficient of the non-proportional flow in whole multipump Power Pack Unit δ_{pzp} = 99.13%.

Results of investigation of non-proportional flow of pumps during simultaneous work are shown inTab.1.

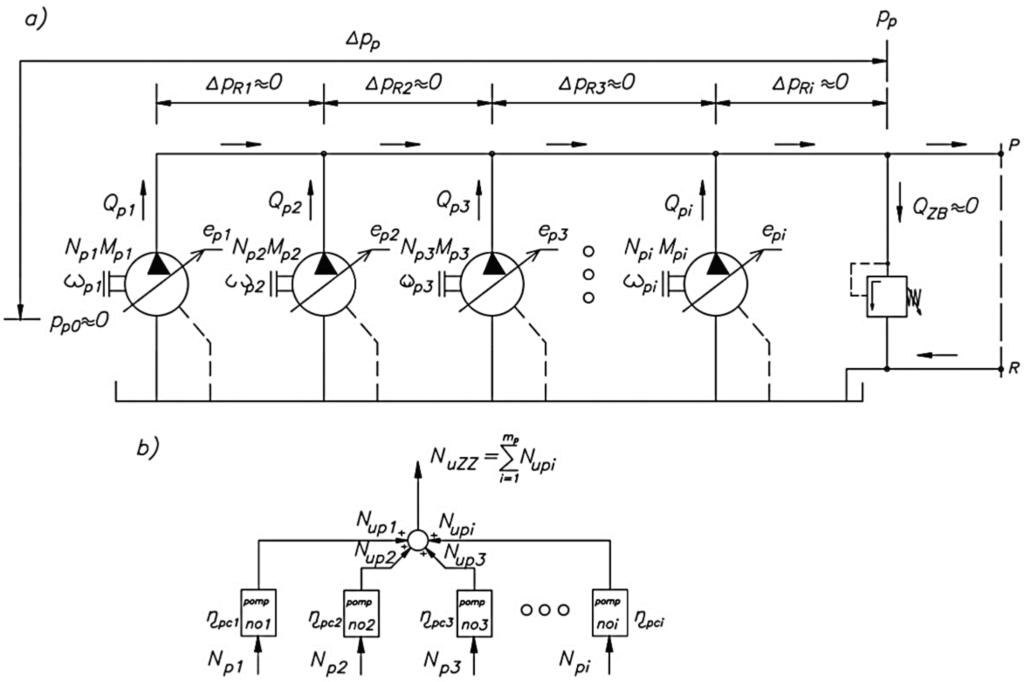

Figure 3 Simplified diagram of multipump type Power Pack Unit working in open system: (a) hydraulic diagram (b) block

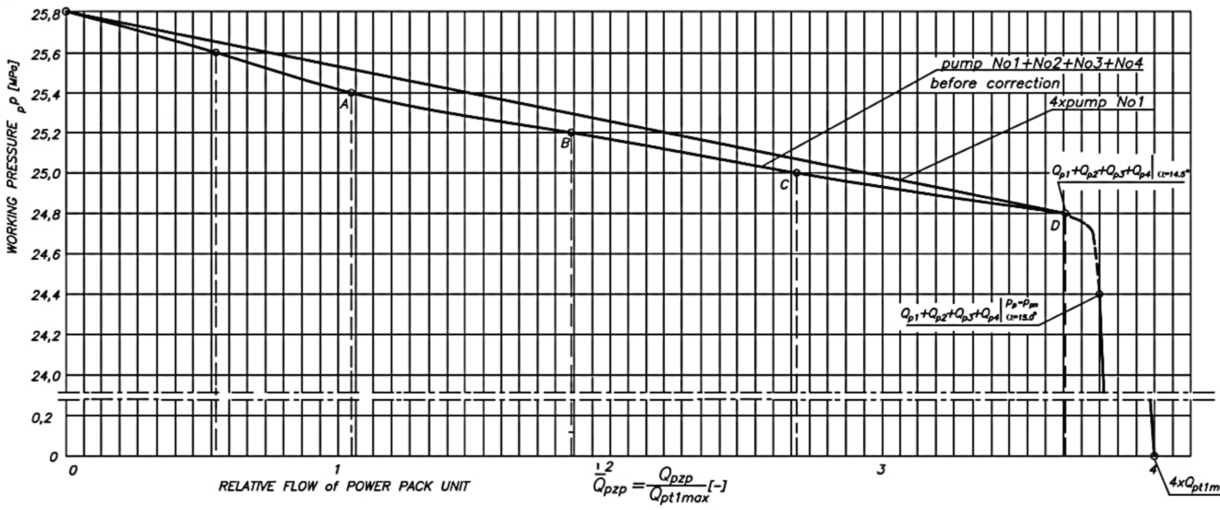

Figure 4 Generalized performance characteristics of Hydraulic Power Pack Unit consisted of four analysed hydraulic axial piston pumps equipped into constant pressure regulators DP before correction

5 DP PRESSURE CONSTANT REGULATORS RANGE OF THE TOLERANCE ADJUSTMENT IDEA

An important conclusion from the analysis presented in last point of the paper is the ascertainment that at the selection of the hydraulic pumps, recommended is so to choose them size so that during the long work of Power Pack Unit at typical conditions of the charge, should be used their possibly full geometrical working volume, i.e. so that their current geometrical volume adjustment of the working volume pump e_p should be possibly greatest [10, 14]. For analysed hydraulic pumps optimum in respect of energetic efficiency the work happens at adjustment of geometrical working volume e_p = 1.0 and relative loading p_p =1.0.

To obtain the best possible it effect at simultaneously work of particular pumps one should have to limit differentions in adjustments and performance characteristics of constant pressure pump controllers DP installed on particular pumps. Therefore one decided to introduce the restriction of the performance characteristics of constant pressure controllers adjustment tolerance range of the of pumps working in the Power Pack Unit $\Delta\alpha$ or all the working range working pressure of the Pump Unit (see Fig. 5). For our case, we found the correction limit of DP performance characteristics according to the following criterion:

$$\Delta\alpha < 4° \tag{9}$$

For so formulated criterion one executed corrections of performance characteristics of DP regulators of each pump (see Fig. 2). To remember here, the base of adjustment relation for regulations of all DP regulators was the performance characteristic of the pump No. 1 DP regulator, which one left without any correction. Performance characteristics of pumps No. 2, No. 3 and No. 4 became properly corrected according to the criterion (9).

Consequently, one received diagram of dependence of the relative real flow in the function of the working pressure p_p of the whole Power Pack Unit, at the established parallel work of all hydraulic pumps simultaneously. Comparatively to analogous diagram prepared for the Power Pack Unit without the correction of DP performance characteristics it is visible that the defection outstanding of result common flow characteristic of the whole Power Pack Unit from the standard flow characterization of the Power Pack consisted of four hydraulic pumps with absolutely the same characteristics according to pump No. 1, sharply surrendered to the improvement [20].

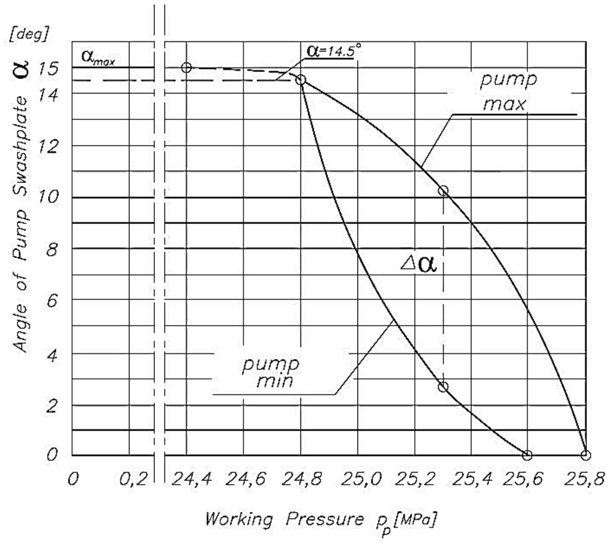

Figure 5 The range of the tolerance idea of the performance characteristics of constant pressure regulators adjustment of pumps working in the Power Pack Unit.

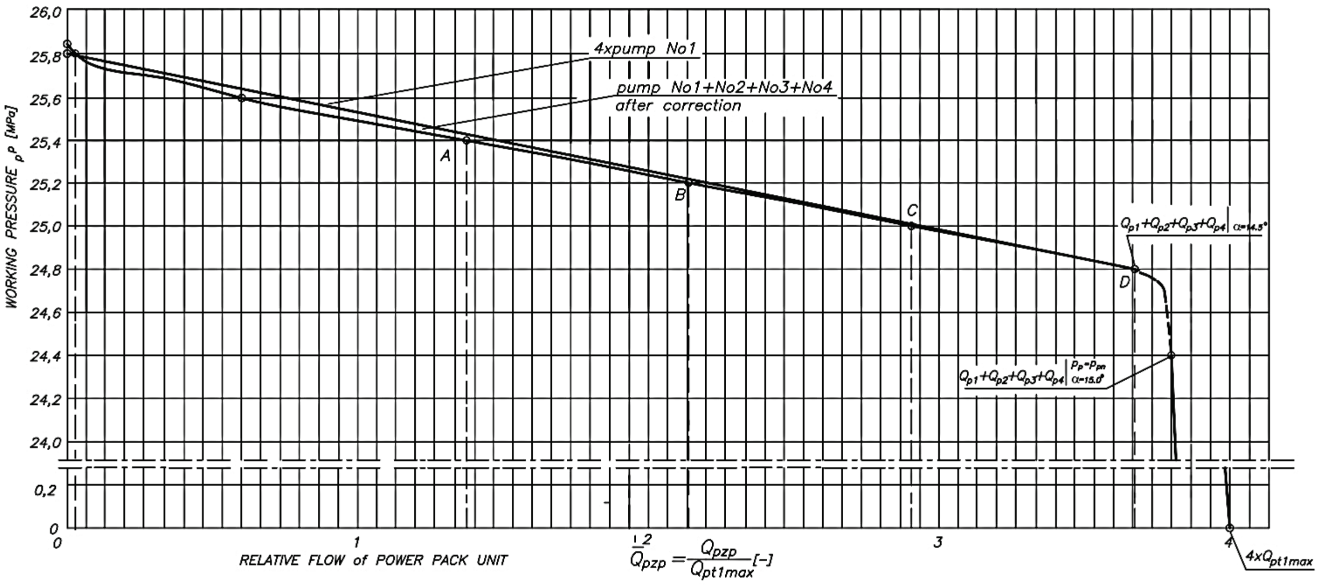

Figure 6 Generalized performance characteristics of Hydraulic Power Pack Unit consisted of four analysed hydraulic axial piston pumps equipped into constant pressure regulators DP after correction

Table 1 Results of percentage participation of hydraulic pumps in generating the Power Pack Unit common flow before and after correction of pump constant pressure DP regulators with value of non-proportional flow coefficients for selected values of working pressure p_p

Working pressure = 25.0 MPa										
	Pump No. 1		Pump No. 2		Pump No. 3		Pump No. 4		δ_{pzp}	Required pump flow
Before correction	27.4%	δ_{p1} = 9.78%	22.9%	δ_{p2} = 8.54%	31.2%	δ_{p3} = 24.91%	18.5%	δ_{p4} = 26.16%	17.35%	25.0%
After correction	25.3%	δ_{p1} = 1.21%	24.2%	δ_{p2} = 3.04%	26.8%	δ_{p3} = 7.36%	23.6%	δ_{p4} = 5.53%	4.28%	25.0%
Working pressure = 25.2 MPa										
Before correction	30.0%	δ_{p1} = 19.87%	16.6%	δ_{p2} = 30.43%	39.2%	δ_{p3} = 56.86%	14.2%	δ_{p4} = 43.29%	38.36%	25.0%
After correction	25.8%	δ_{p1} = 3.01%	22.9%	δ_{p2} = 8.54%	28.6%	δ_{p3} = 14.43%	22.8%	δ_{p4} = 8.90%	8.72%	25.0%
Working pressure = 25.4 MPa										
Before correction	35.2%	δ_{p1} = 40.88%	0.0%	δ_{p2} = 100.0%	54.8%	δ_{p3} = 119.15%	10.0%	δ_{p4} = 60.03%	80.01%	25.0%
After correction	26.9%	δ_{p1} = 19.45%	20.0%	δ_{p2} = 39.99%	30.0%	δ_{p3} = 10.08%	23.2%	δ_{p4} = 30.48%	25.00%	25.0%

For engineers without enough experience, analysing above mentioned Power Pack Unit flow characteristic it can seem that one obtained the almost ideal flow characteristic. During that time, the occurrence of the non-proportionality flow of the hydraulic particular pumps working simultaneously remained in consequence of passed correction limited but not eliminated. In Tab. 1 were taken down values of the percentage participation of each hydraulic pump in generating common flow of the Power Pack Unit for selected values of working pressure of the expression before and after passed correction [20]. As is visible, after correction (see Fig. 6), at working pressure p_p = 25.4 MPa, succeeded to raise the proportional flow participation of the pump No. 2 from the zero value to 20.0% and limitations of the percentage participation of the pump No. 3 from 54.8% to the level 30.0%, where ideal recommended level is the value 25.0%. Simultaneously after the introduction of DP pump regulator corrections, the deviation of the proportional participation of each buoyancy pump in represented range of working pressures p_p = 25.0 ÷ 25.4 MPa in generating the common Power Pack Unit flow from the required value 25.0% did not exceed values ±5%, what was impossible for the Power pack Unit without introduced corrections. In result introduction of corrections are permission on the reduction of the non-proportionality flow problem in Power Pack Unit, but not his elimination to the end.

6 CONCLUSIONS

Executed in paper analysis referring to changes of flow pumps simultaneously working parallel in big power multipump Power Pack Units allows preparation of the following conclusions:

1. In multipump type Hydraulic Power Pack Units each pump can often work at same working pressure with different swashplate angles and with different values of current geometrical working volumes e_p, in time of simultaneously parallel work within the framework of one Power Pack Unit,
2. For limitation of non-proportional flow of pumps phenomena in multipump Power Pack Units it is recommended to increase adjustment accuracy of constant pressure regulators on the way of decrease the range of performance characteristics of above mentioned pump regulators each pumps for decreasing of pump swashplate angle differents under of their parallel work in the Power Pack Unit in the all range of working pressures,
3. Non-proportionality flow of pumps phenomena in multipump Power Pack has influence on non-proportional Power loading of particular pumps working simultaneously and parallel in one Pump Unit. As a result, this problem can lead to non-proportional wearing of not only parts and elements of particular hydraulic pumps but prime motors of above mentioned pumps including (t.e. electric or diesel motors).

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