SHIP BALLASTING PROCESS TIME CALCULATION WITH USE OF SUBMERGED BALLAST PUMPS WITH HYDRAULIC DRIVE SUPPLIED FROM CONSTANT PRESSURE HYDRAULIC CENTRAL LOADING SYSTEM ON MODERN PRODUCT AND CHEMICAL TANKERS

Andrzej Banaszek [1], Radovan Petrovic [2*], Sasa Botocanin[3]

[1] *West Pomeranian University of Technology Szczecin / Maritime Research and Transport Faculty, Street Al. Piastów 41, 71-065 Szczecin, Poland , E-mail: andrzej.banaszek@pzut.edu.pl*

[2] *The University "Nikola Tesla" in Belgrade, Faculty of Information Technologies and Engineering, 11158 BEOGRAD SERBIA, Cara Dusana 62-64, e-mail: radovan4700@yachoo.com*

[3] *PPT-Namenska AD – Company Wall,Cara Dušana 101, 37240 Trstenik , Serbia, e-mail:* ***sasa.batocanin@ppt-namenska.rs***

* Corresponding author: e-mail address: Tel. +38 1637580104 radovan4700@yachoo.com

ABSTRACT

Ballast systems are among one of the most important installations installed on board modern product and chemical tankers. They have a significant impact on the safety of the ship by determining its stability on the sea wave and the strength of the hull. Logistically, the efficiency of these systems is crucial in planning the loading and unloading times of a given tanker. Due to the explosion-hazardous zone, ballast pumps of the submersible type are often equipped with hydraulic drive. The purpose of the article is to present a methodology for calculating the time of ballasting process of a modern chemical tanker and the flow control of submerged ballast pumps with hydraulic drive, supplied from the hydraulic constant-pressure central loading system. The results of the calculations are important for the correct determination of the liquid cargo loading time of a tanker at the fuel terminal, the organization of the logistics and service system for the ship and for the entire loading port. In addition, the paper presents the construction of a submersible ballast pump with hydraulic drive with a description of the hydraulic system installed on board a modern tanker. The methods of controlling the flow of submerged ballast pumps are described, along with a new concept of using constant torque regulators to control the pump flow. Experimental results and theoretical calculations are presented.

Keywords: Balalst system , ballast pumps, hydraulic drive, hydraulic central loading system, product and chemical tankers

1. INTRODUCTION

Specialized vessels - product and chemical tankers - are used to transport liquid petroleum cargoes by sea. Ballast systems installed on their decks are among the most important service installations.... They have a major impact on the safety level of the ship. The size of the water ballast determines the depth of the ship, the position of the center of gravity and the metacentric height. The above technical parameters have a significant impact on the ship's wave behavior and wave stability. Often, poor ballasting of such a ship can lead to overturning of the ship on a wave in rough sailing conditions and a maritime disaster. The speed of the ship's captain's reaction in case of stability problems is therefore extremely important in the problem of safe navigation. Many times, on modern product and chemical tankers, the technical capability of the ballast system also determines the total handling time in port, which is extremely important from the logistical and business side. Newly built tankers, according to the requirements of the International Maritime Organization (IMO), must have a hull structure of the "Double Hull" type[14]. In a hull design of this type, each cargo tank must be separated from the ship's outer side by ballast tanks or an empty cofferdam. This arrangement protects the ship from the outflow of liquid cargo in the event of a collision or if the ship enters an underwater obstacle. An example of this type of ship is the B573-I/2 class product tanker m/t "Simunye", built at Szczecin Shipyard S.A. for the shipowner Unicorn Tankers Ltd. of South Africa (see Fig.1). The ship's side ballast tanks, created the possibility of mounting directly inside two (2) submerged ballast pumps with hydraulic drive. This allowed the creation of a new type of ballast system, with the ballast pumps fully installed inside the ballast tanks (as opposed to the traditional solution with ballast pumps installed in the engine room). This saves a lot of space inside the ship's engine room, while reducing the total length of ballast pipes and flow resistance in the ship's ballasting operation. This article presents the idea of a new type of

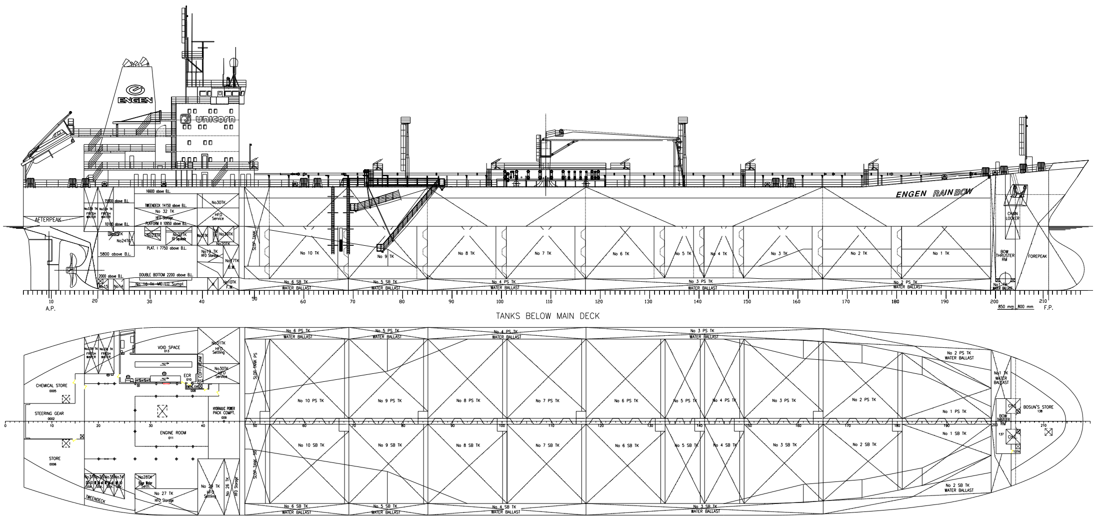


Figure 1. Product tanker B573-I/2 class m/t 'Simunye', built by the Shipyard Szczecinska S.A/Poland for the ship-owner -Unicorn Tankers from Republic of South Africa [16]

ballast system with submerged ballast pumps fed from a constant-pressure hydraulic central loading system. Also described are ways of constant-pressure control of the capacity of submerged ballast pumps using constant Torque controllers and their influence on the time of ship ballasting operations. Experimental results and theoretical calculations of the flow of a submerged ballast pump for different drop pressure settings in a hydraulic drive motor are presented In the literature, due to the specificity of the maritime subject, there are no studies on the above topic. We can only note the of the paper authors : (Banaszek [1]), Banaszek, Petrovic [2][4], Banaszek,Urbanski [3], describing the use of a hydraulic central loading system to drive submerged ballast pumps, as well as the works of Gorski-Perepeczko [10][9], Kutyrkin-Postnikov [12] and the instructions of pump manufacturers Framo [5], Hyunday [6], Shinko [7] and also other authors [17][18]. In this article, the theoretical calculations carried out are supplemented by the presentation of exemplary solutions of this type of hydraulic systems installed on board product and chemical tankers built in Szczecin Shipyard / EU-Poland.

2. DESCRIPTION OF THE STRUCTURE OF THE BALLAST SYSTEM WITH SUBMERSIBLE HYDRAULICALLY DRIVEN BALLAST PUMPS

The hull design of "Double Hull" types on modern product and chemical tankers and the resulting side ballast tanks made it possible to install submerged ballast pumps directly in them (see Fig.2.). Submersible ballast pumps are generally 1-stage, centrifugal pumps. According to Lloyd's Register regulations [14], this is a hazardous area due to the possibility of explosion. For this reason, the most common way to drive them on modern tankers is to use hydraulics, as safer than the alternative solution with electric motors. Total length of submerged ballast pumps

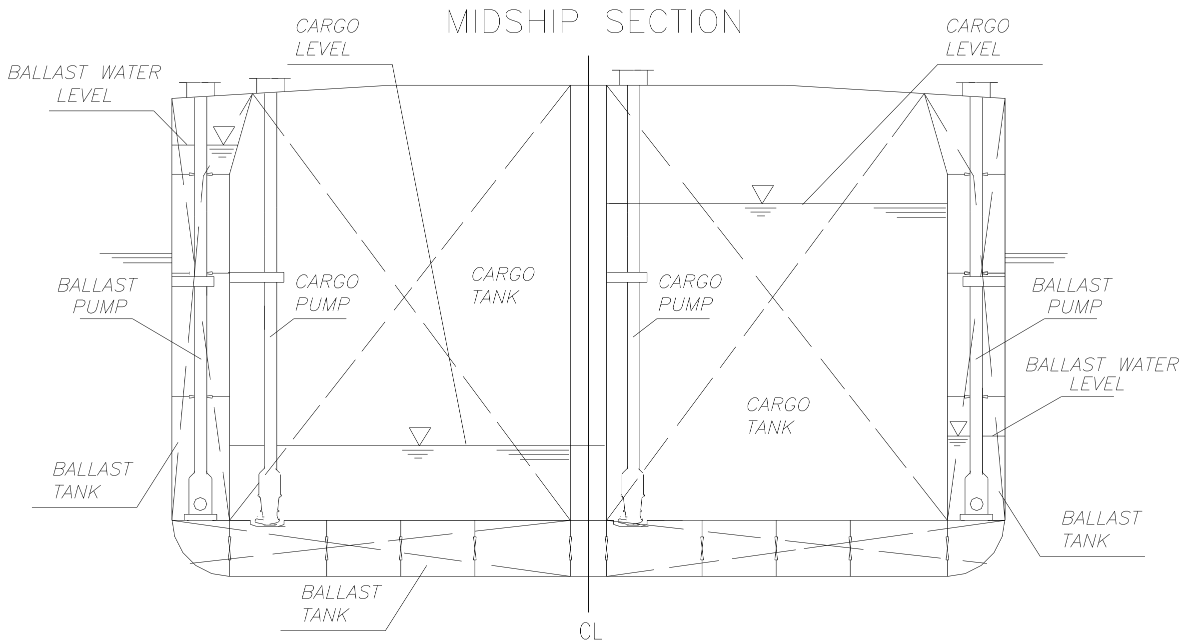


Figure 2. Submerged ballast pumps montage inside of side ballast tanks

must be adjusted each time to the situation on the ship - to the current height of the ballast tanks (see Fig.2). As a result, the height of the submerged pumps reaches 14-20 [m]. Older solutions with a long drive shaft and electric motor often experience vibration problems during the operating period. This is often the result of the unbalance of the long drive shaft. Mounting the hydraulic drive motor at the bottom, directly in the ballast pump head, reduces the length of the shaft and avoids excessive vibration problems to a minimum (see Fig..3a.). Due to the difficulty of service access to the head during normal operation of the pump (see Fig.3b), a fixed-pitch

A2FM-type axial piston hydraulic motor from Bosch-Rexroth/Germany (recognized as one of the most reliable hydraulic motors in its class [8]) was used as the drive motor. On the top plate, located directly on the deck coaming on the open deck, there is a hydraulic block designed to control the flow of oil supplying the hydraulic motor. Connected to the above block are hydraulic branches running from the hydraulic central cargo system. Hydraulic oil flows from the block to the hydraulic engine through a system of concentric lines, where there is a pressure line in the center, around the return line area, insulated on the outside with a glycerine filled cofferdam. In the event of loss of tightness through the hydraulic lines - the leaking

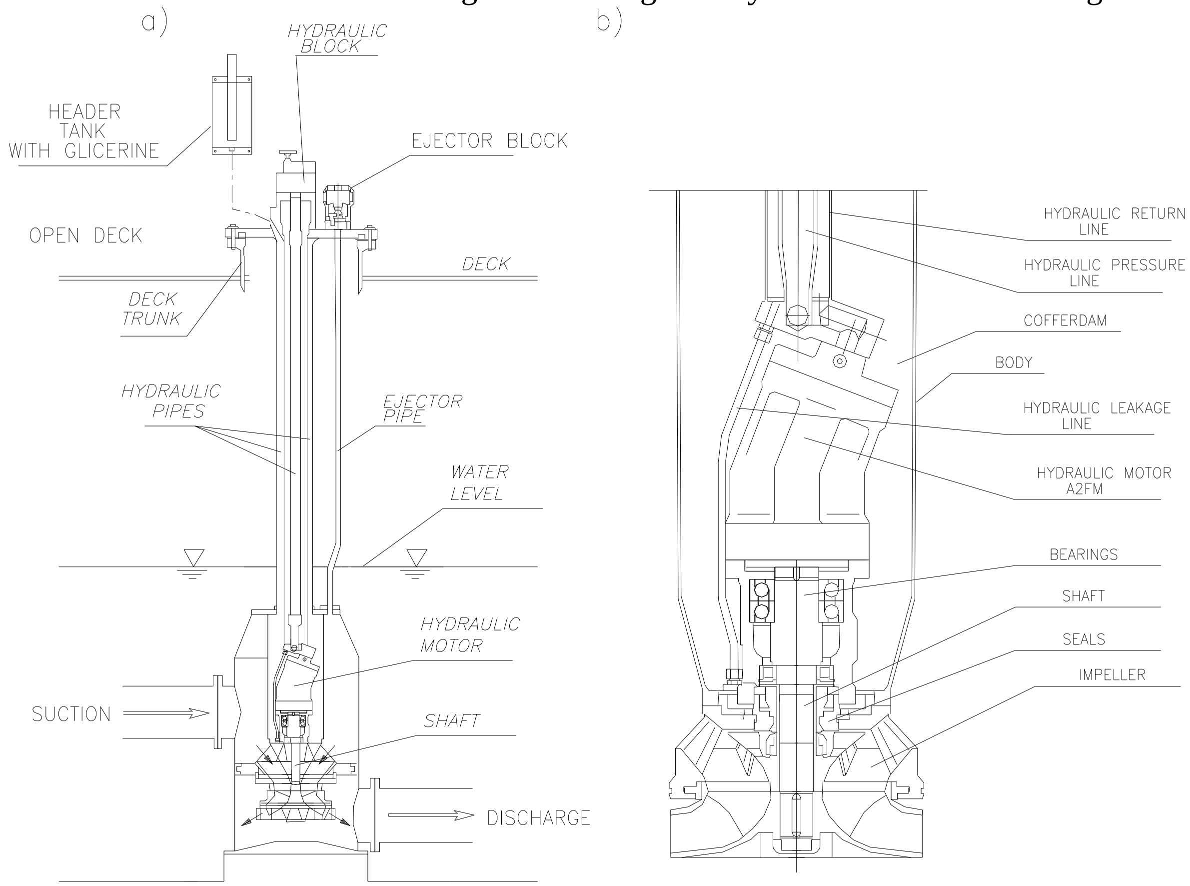


Figure 3. Typical submerged ballast pump with hydraulic drive (Framo/Norway) [5]

oil pushes the glycerin out of the tank to the outside alerting the service to the damage of the pump (see Fig.3a). Glycerin is fully safe for the natural marine environment, as it is fully biodegradable in seawater. Fig.4 shows the construction of a typical new-type water ballast system with submerged ballast pumps, installed on board modern product and chemical tankers. It consists of two ballast buses to which all ballast tanks are connected by separate pipe branches. A system of remotely controlled ballast valves allows any configuration of such a system. In the event of damage, it is possible to cut off the damaged pump and continue pumping operations by a second ready ballast pump. Ballast piping on typical ships is generally long, running along the entire double bottom of the ship. This situation can cause cavitation problems during emptying operations of ballast tanks, especially remote from the pumps. For this reason, each ballast pump is equipped with a semi-draining system with an air separator. Two parallel ballast trunk lines with two separate and independent ballast pumps increase the reliability of this important ship system in the process.

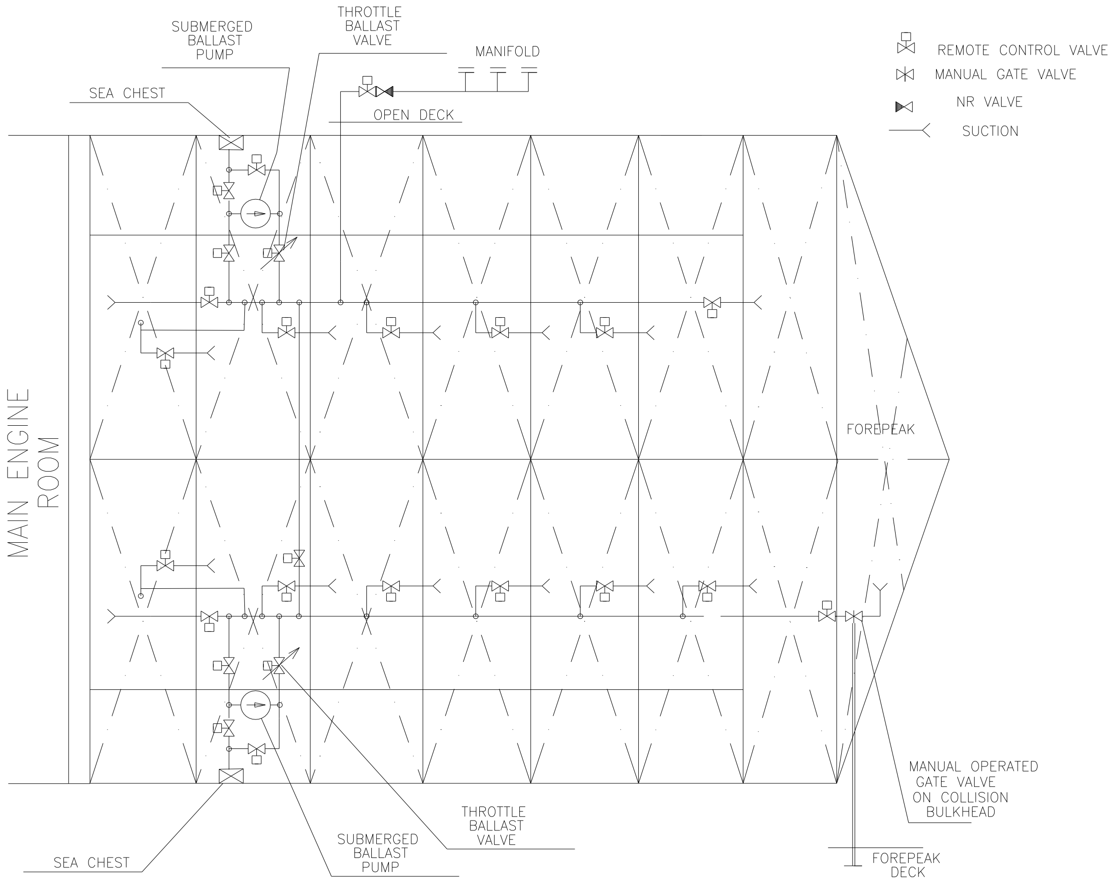
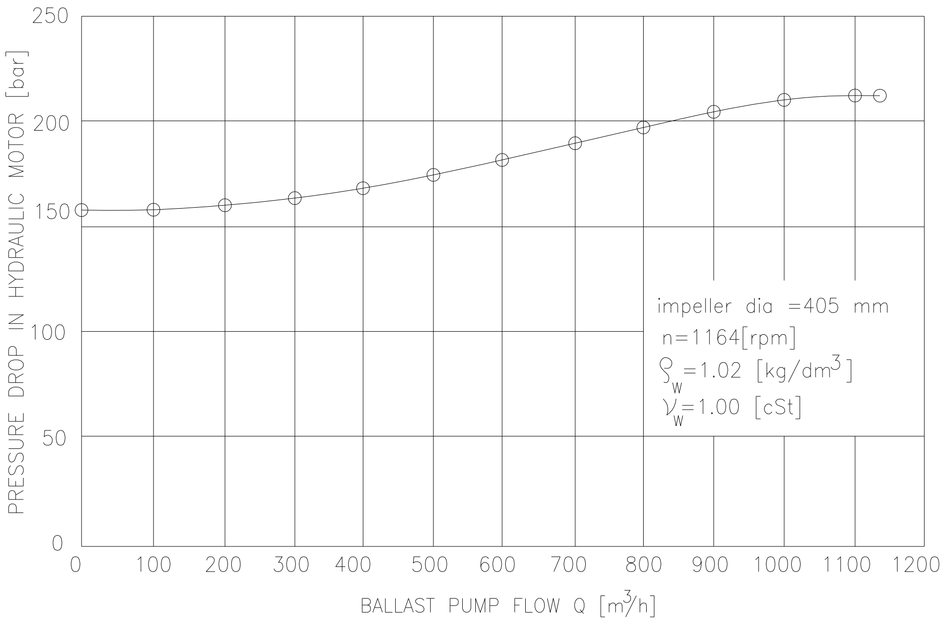


Figure 4. The water ballast system diagram with submerged ballast pumps

3. HYDRAULIC DRIVE SYSTEM WITH CONSTANT-TORQUE FLOW CONTROL

On board the B 573-I/2 class product tanker m/t "Simunye"[16] were 2 (two) submerged ballast pumps of the SB300 (Framo) type, with a nominal discharge flow of Q_{BPO} = 800 each, obtained at discharge pressure H_{BPO} =25 [mlc], When pumping seawater with a density of ρ_W=1,02 and kinematic viscosity ν_W =1,0 [cSt]. A hydraulic controller fulfilling the principle of a hydraulic oil flow controller was installed on the supply hydraulic line running from the hydraulic central main line. The controller set a constant value for the flow of hydraulic oil supplying the hydraulic pump motor, regardless of its load. Hydraulic motors of the A2FM axial-piston type (Bosch Rexroth [8]), installed in ballast pumps, are characterized by high volumetric efficiency. The value of volumetric leakage does not exceed the level of 5% of the total flow demand. Therefore, it can be approximated that the value of the flow demand of a hydraulic drive motor is proportional to the speed of the motor shaft. Figure 5 shows the flow and drive characteristics of the SB300 type ballast pump at nominal speed. The nominal value of the motor flow demand Q_S =240[l/min], which corresponds to the speed of the pump impeller n_S =1164 [rpm].



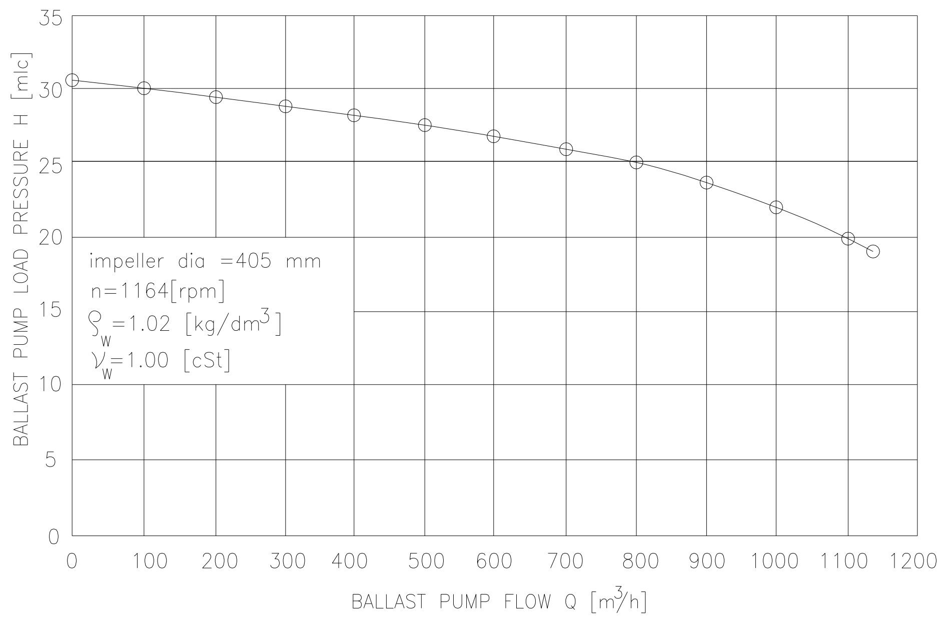


Figure 5. Technical characteristics of submerged ballast pump SB300 type (Framo) [5]

In order to better stabilize the flow value of the ballast pump, constant-torque controllers were used for capacity control. Its task is to maintain a constant oil pressure drop in the hydraulic drive motor. The staff supervising the operation of the hydraulic central loading system on board the described tanker can control the following technical parameters of the hydraulic power system:

- p_G - the delivery pressure of the main power pack and hydraulic central loading system
- Δp_S - the value of hydraulic oil pressure drop in the hydraulic motor of the investigated device connected to the hydraulic central loading system

The value of the first parameter can be adjusted using the p=const controller installed on the main power pack in the Power Pack Room. In the adjustment procedure, it should be remembered that the value of the discharge pressure in the hydraulic system should not be less than the total pressure drop of the hydraulic oil in the hydraulic motor of the most loaded hydraulic consumer and the pressure drop in the hydraulic system, including all hydraulic control valves in the path of hydraulic oil flow. In the case of ballast pumps, the value of the pressure drop Δp_S =const is regulated by a hydraulic controller in the pump's hydraulic block. In Fig 6. one showed the structure the pressure drops of hydraulic oil in the supply process of analyzed ballast pump. Hydraulic oil discharged by the central power pack is moved by the main pressure line of hydraulic central loading system and the master valve (constant-torque controller) to the driving motor of ballast pump and then comes back by the main return line to the power pack

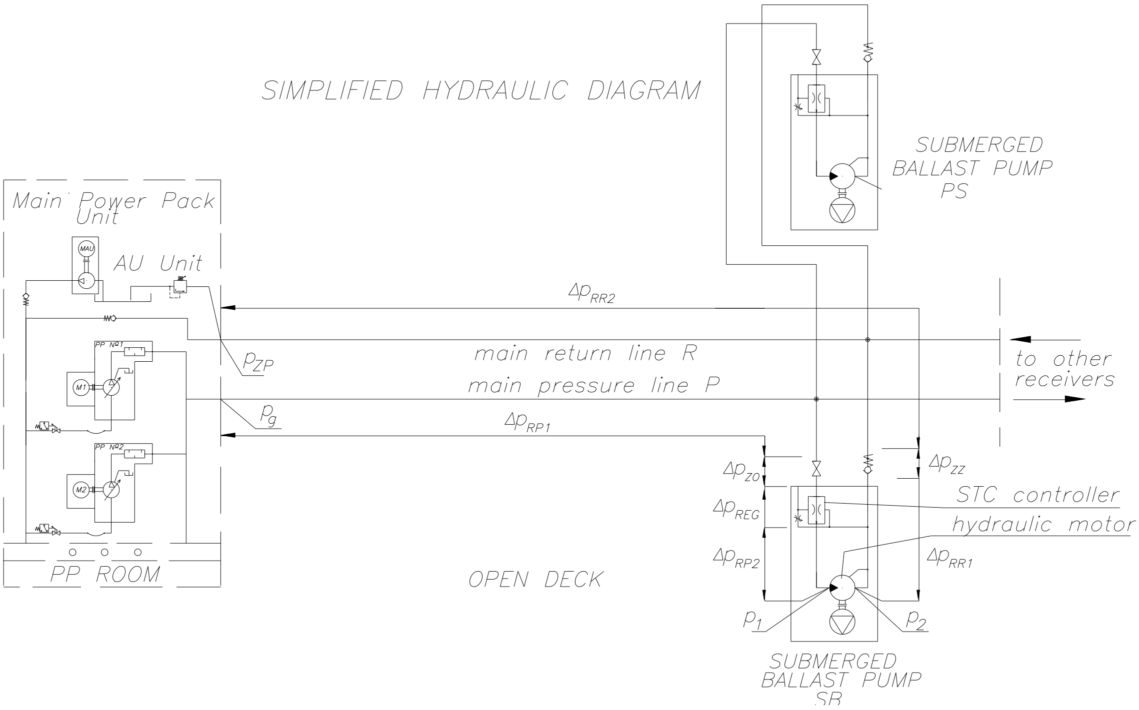


Figure 6. Hydraulic oil pressure drops in supply process of ballast pump from hydraulic central loading system

In compliance with diagram, one can write that pressure drop Δp_S in hydraulic motor of ballast pump is equal:

$$\Delta p_S = p_1 - p_2 \tag{1}$$

where : p_1, p_2 - hydraulic motor inlet and outlet pressure,

The minimum working pressure of the hydraulic power pack p_G, must be not less than:

$$p_G = p_1 + \Delta p_{RP} + \Delta p_{ZO} + \Delta p_{REG} \tag{2}$$

where: Δp_{RP} - pressure drop in hydraulic main pressure line

$$\Delta p_{RP} = \sum_{i=1}^{n} \lambda_i \cdot \frac{l_i}{d_i} \cdot \rho \cdot \frac{\upsilon_i^2}{2} + \sum_{j=1}^{m} \zeta_j \cdot \rho \cdot \frac{\upsilon_j^2}{2} \quad ; \qquad \upsilon_{i,j} = \frac{4Q_{i,j}}{\pi \cdot d_{i,j}^2} \tag{3}$$

where: $d_{i,j}$, $Q_{i,j}$, $v_{i,j}$ -pipe diameter , oil flow, oil velocity, Δp_{ZO} - ball valve pressure drop, Δp_{REG} - pressure drop in hydraulic constant-torque controller

Hydraulic motor outlet pressure can be calculated from the equation :

$$p_2 = p_{ZP} + \Delta p_{ZZ} + \Delta p_{RR} \tag{4}$$

where: - Δp_{RR} hydraulic main return line pressure drop

$$\Delta p_{RR} = \sum_{i1=1}^{n1} \lambda_{i1} \cdot \frac{l_{i1}}{d_{i1}} \cdot \rho \cdot \frac{\upsilon_{i1}^2}{2} + \sum_{j1=1}^{m1} \zeta_{j1} \cdot \rho \cdot \frac{\upsilon_{j1}^2}{2} \quad ; \qquad \upsilon_{i1,j1} = \frac{4Q_{i1,j1}}{\pi \cdot d_{i1,j1}^2} \tag{5}$$

- Δp_{ZZ} - NR valve pressure drop, p_{ZP} - adjustment pressure in support relief valve of main hydraulic power pack unit (see diagram in Fig.6)

In typical hydraulic systems of central loading, the value of the pressure p_{ZP} carries out 2-10 [bar] in dependences from the size of the system and the auxiliary hydraulic unit. In hydraulic central loading system installed on board product tanker m/t „Simunye" adjusted value of the opening support relief valve carried out: $p_{ZP} = 4$ [bar] (6)

Taking into account equals (2) and (4), the minimum value of the delivery pressure of the central hydraulic power pack should be not less than:

$$p_{G\min} \geq \Delta p_s + \Delta p_{RP} + \Delta p_{ZO} + \Delta p_{REG} + p_{ZP} + \Delta p_{ZZ} + \Delta p_{RR} \tag{7}$$

For the purpose of avoiding of problems with an unstable acting of the p=const. controller, the adjusted value p_{Gmin} in effect is accepted on the higher level:

$$p_G = \Delta p_s + \Delta p_{RP} + \Delta p_{ZO} + \Delta p_{REG} + p_{ZP} + \Delta p_{ZZ} + \Delta p_{RR} + \Delta p_{SS} \tag{8}$$

where Δp_{SS} - pressure drop safety surplus, typically $\Delta p_{SS} = 10$ [bar]

In analyzed case, as was above described, the hydraulic controller STC -90 adapts the value of the pressure drop Δp_{REG} in this way that the size the pressure drop in the hydraulic motor must be constant (control algorithm formula): Δp_S = const (9)

4. MATHEMATICAL MODEL

The drive torque on the impeller drive shaft of ballast pump (is described by means of the Schloesser formula [15]:

$$M_S = M_{ST} - \Delta M_{mf} - \Delta M_{m\mu} - \Delta M_{mh} - \Delta M_C \tag{10}$$

where: - M_S - actual torque on hydraulic motor drive shaft

- theoretical torque on motor drive shaft

$$M_{ST} = \frac{Dm}{2\pi} \cdot \Delta p_S$$

- Dm - hydraulic motor stroke displacement
- ΔM_{mf} - torque losses caused by mechanical friction

$$\Delta M_{mf} = C_{mf} \cdot \frac{Dm}{2\pi} \cdot \Delta p_S \tag{11}$$

C_{mf} - mechanical friction coefficient dependent on pump construction

- $\Delta M_{m\mu}$ - torque losses caused by viscosity friction

$$\Delta M_{m\mu} = C_{m\mu} \cdot \mu \cdot n_S \cdot Dm \tag{12}$$

$C_{m\mu}$ - viscosity friction coefficient dependent on pump construction and oil parameters,

μ- dynamic viscosity of the hydraulic oil, n_S - ballast pump rotation velocity

- ΔM_{mh} - torque losses caused by hydraulic losses in a motor

$$\Delta M_{mh} = C_{mh} \cdot \frac{\rho \cdot n_S^{\ 2}}{4\pi} \cdot \sqrt[3]{q_S^{\ 5}} \tag{13}$$

C_{mh} - hydraulic losses coefficient , **ρ**- hydraulic oil density, ΔM_C - constant loss torque in hydraulic motor

The dependence (10) can be write in the simplified form as:

$$M_S = \frac{Dm}{2\pi} \cdot \Delta p_S - \Delta M_S = M_{ST} - \Delta M_S \tag{14}$$

where: ΔM_S - total torque losses in hydraulic motor

$$M_S \approx M_{ST} = \frac{Dm}{2\pi} \cdot \Delta p_S \tag{15}$$

This means that the constant value of the hydraulic oil pressure drop in the ballast pump driving motor, holding consequently quasi-constant value of the impeller driving torque.

In compliance with the dynamic similarity theory of the centrifugal pumps (Łazarkiewicz-Troskolański [11], Jedral [13]), the change of the rotation speed of the pump impeller with relation to the nominal speed causes the change of main technical parameters of the pump according to following formula:

$$\frac{Q_{BPO}}{Q_{BP}} \cong \frac{n_n}{n_S} \tag{16}$$

$$\frac{H_{BPO}}{H_{BP}} \cong \left(\frac{n_n}{n_S} \right)^2 \tag{17}$$

where: H_{BPO}, Q_{BPO}, n_n - technical parameters of ballast pump at nominal pressure drop in a motor

H_{BP}, Q_{BP}, n_S - technical parameters of ballast pump at nominal pressure drop in a motor

In result ,from equations (17-19), the corrected values of the ballast pump flow characteristics can be calculated from the following simplified equations:

$$H_{BP} = H_{BPO} \cdot \frac{\Delta p_S}{\Delta p_{S0}} \tag{18}$$

$$Q_{BP} = Q_{BPO} \cdot \sqrt{\frac{\Delta p_S}{\Delta p_{S0}}} \tag{19}$$

where: Δp_S [bar], Δp_{S0} [bar]- nominal and actual pressure drop in hydraulic motor

5. EXPERIMENTAL AND CALCULATION RESULTS

To verify the flow control system of the pump with staomomentum control, experimental measurements of the flow of submerged ballast pumps of the SB300 type were made. The measurements were carried out at a special measuring station in the laboratory of the pump manufacturer FRAMO / Norway. Flow measurements were made when pumping seawater with a density of $\rho_w = 1025[kg/m^3]$ and kinematic viscosity $\nu_w = 1.0[cSt]$. The diameter of the ballast pump impeller was 405 [mm]. Axial piston hydraulic motor type A2FM200 with fixed displacement/stroke $Dm = 200[cm^3/rot]$ [12] was used as hydraulic motor. HLP-46 grade hydraulic oil with kinematic viscosity was used to drive the hydraulic motor. $\nu_H = 46.0[cSt]$ in temperature measurements $T_o = 50[^oC]$. The above data were in accordance with the manufacturer's recommendations. Pump flow measurements were made at 7 pressure drop settings on the STC-90 constant torque flow controller: $\{\Delta p_S\} = \{250, 225, 200, 175, 150, 125, 100\}[bar]$

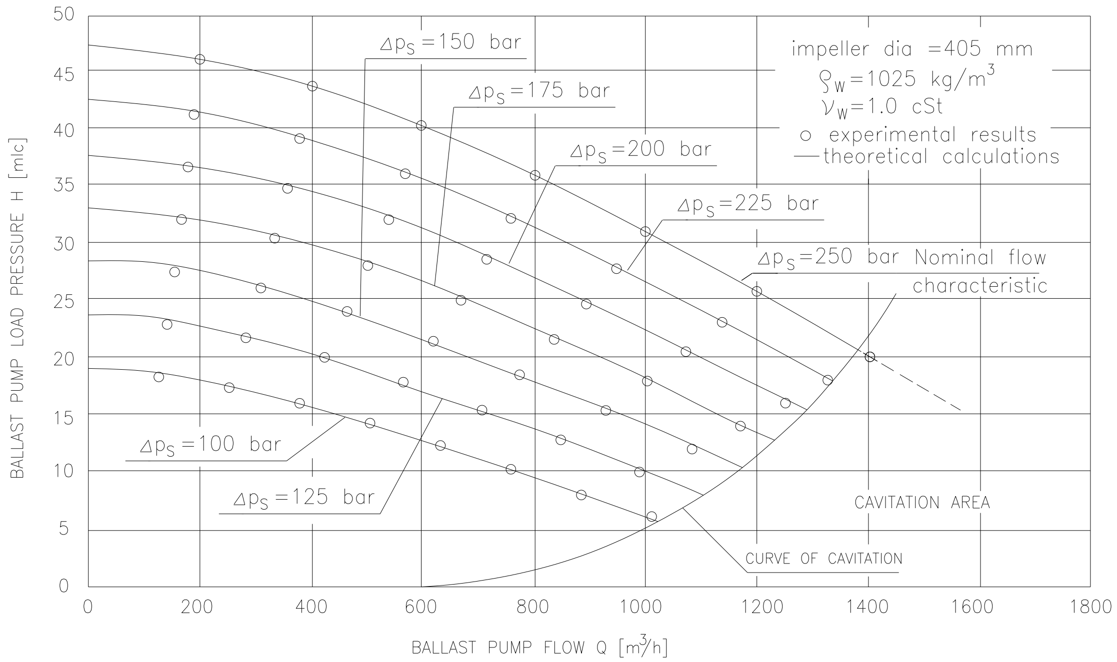


Figure 7. Experimental and calculation results of submerged ballast pump SB300 type, with the hydraulic drive and constant-torque flow control for seven (7) different drop pressure adjustments Δp_s

The results of experimental measurements of the described ballast pump are shown in Figure 7. They were compared with the results of theoretical calculations. Good agreement was obtained between the above results, confirming the usefulness of the mathematical model formulated in the paper. Using the proposed mathematical model, an example calculation of the time to carry out the ballasting process on the product tanker m/t "Simunye" was carried out for the following data : Volume of pumped water ballast: 12 000 cubic m, average flow resistance in the ballast system - 21 [mlc], setting constant torque controller STC-90 - Δp_{ss} = 200 [bar]
As a result, the time of total ballasting of the ship was obtained $t_{ballast}$= 11 hour 36 min. This result was consistent with the ship's actual ballasting time with an error of 1%, confirming the utility of the computational model.

6. CONCLUSIONS

Ballast pumps of the submersible type on board modern product and chemical tankers are the most important part of the ballast system. They are installed directly in the side ballast tanks, which are, according to the regulations of the calibration institutions, treated as a hazardous, explosive zone. Therefore, the ballast pumps described are usually equipped with a hydraulic drive, fed from a hydraulic central supply system. On the B573-I/2 class product tanker m/t "Simunye" it was decided to use the existing hydraulic central power system on board to drive the pumps. Hydraulic constant-torque controllers were used to control the flow of the ballast pumps for better stabilization of the discharge of the delivery pump with changing flow resistance in the ballast system and the water level in the individual ballast tanks. The paper presents the results of measurements of flow characteristics of ballast pumps operating at different from nominal values of hydraulic oil pressure drop in the engine, set in the constant-torque controller. The paper presents a computational model of ballast pump performance with hydraulic drive and control by a constant-torque controller. The results of experimental measurements of the ballast pump flow were compared with the results of theoretical calculations. Good agreement between the two results was confirmed. Therefore, the article is utilitarian in nature and can be helpful to ship ballast system designers, tanker fleet management personnel and logisticians responsible for the port fuel terminal.

7. LIST OF NOTATIONS

p_G	1 the working pressure of the main power pack and hydraulic central loading system	bar
Δp_S	the value of hydraulic oil pressure drop in the hydraulic motor	bar
p_1	inlet pressure to the hydraulic motor	bar
p_2	outlet pressure from the hydraulic motor	bar
Δp_{RP}	pressure drop in hydraulic main pressure line	bar
$d_{i,j}, Q_{i,j}, \upsilon_{i,j}$	diameter in pipe, oil flow, oil velocity	$m, dm^3/\min, m/s$
Δp_{RR}	pressure drop in hydraulic main return line	bar
p_{ZP}	adjustment pressure in support relief valve in filling up system	bar
M_S	performance torque on hydraulic motor drive shaft	Nm
Dm	hydraulic motor stroke displacement	cm^3/rot
n_S	cargo pump rotation velocity	rpm
ρ	hydraulic oil density	kg/m^3
$C_{mf}, C_{m\mu}, C_{mh}$	proportional coefficients experimentally determined	-
ΔM_{mf}	torque losses caused by mechanical friction in a motor	Nm
$\Delta M_{m\mu}$	torque losses caused by viscosity friction in a motor	Nm
ΔM_{mh}	torque losses caused by hydraulic losses in a motor	Nm
ΔM_C	constant loss torque in hydraulic motor	Nm
M_{ST}	theoretical torque on motor drive shaft	Nm
ΔM_S	total loss torque in hydraulic motor	Nm
T_o	temperature of oil	^oC
ρ_w	density of sea water	kg/m^3
ν_w	kinematic viscosity of sea water	cSt
ν_H	kinematic viscosity of hydraulic oil	cSt
μ	dynamic viscosity of the hydraulic oil	cP
Q_{BPO}	nominal flow of ballast pump	m^3/h
H_{BPO}	nominal load pressure of ballast pump	M
Δp_{SO}	The nominal value of hydraulic oil pressure drop in the hydraulic motor	bar

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