# Simulation of Loading/Discharging Procedure of Tankers

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# Simulation of Loading/Discharging Procedure of Tankers

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ABSTRACT: Traditionally the onboard systems of the ship are designed on past experience and standardization. The current requirement for energy efficiency improvements requires a fresh look in their design process, based on first principles and prescribed goals, i.e. Goal Based Design (GBD). In this paper this approach is applied on the cargo and ballasting systems of an oil tanker, using a newly developed simulation tool. The tool will be integrated in a multi-objective optimization procedure. All the involved design and operational parameters and constraints have been taken into account in order to produce realistic results. These can be useful for both for the designer and the operator, supporting the decision making process onboard the ship.

## 1 INTRODUCTION

The design and operation of energy efficient ships is of paramount importance nowadays for the shipping industry. In order to achieve this, the efficiency of all onboard ship systems needs to be studied carefully. For oil tankers one of the most important systems in terms of energy consumption is the cargo handling system. IMO's regulations regarding the ship's overall efficiency, and, in general, its impact on the environment are very demanding and will become even more demanding in the following years. A result of this environmental awareness is the recently enforced Ship Energy Efficiency Management Plan (SEEMP), which is mandatory for ships over 400 GT from the 1st of January 2013(MARPOL, Annex VI). A simulation tool for the optimization of loading and discharging procedures of tankers can be valuable for the operator. A vast number of parameters must be taken into account in order to achieve an optimum energy efficient operation. At the same time, the operator must make decisions during the procedure in order to ensure that all the constraints, e.g. the maximum allowable trim and draft, the bending moments, the net positive suction head of the pump etc. are not violated. The aim of the herein developed tool is to include all the above considerations in a simulation procedure that enables the operator to optimize the discharging performance of the vessel with the minimum energy footprint.

## 2 MATHEMATICAL MODELING

### 2.1 *Basic Principles*

#### 2.1.1 *Bernouli Equation*

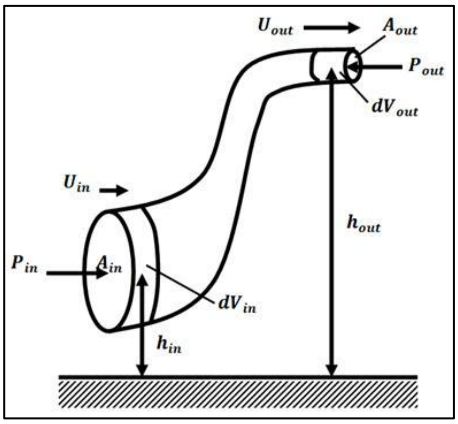


Figure 1. Flow through pipe

Let us assume the flow in the pipe shown in the Figure 1. The liquid is uncompressible, there are no hydraulic losses and the mass continuity equation yields to the volume continuity equation, e.g. $dV_{in}=dV_{out}=dV$

The amount of energy entering the pipe must be equal to the amount of energy exiting, for a small period of time *dt*.

$E_{in} = E_{out}$ , or

$$W_{in} + PE_{in} + KE_{in} = W_{out} + PE_{out} + KE_{out} \quad (1)$$

Where:

- $W_{in}$ is the energy given to the system due to some force $F_{in}$ at the section in the entrance of the pipe (e.g. due to the atmospheric pressure, or pressure due to some liquid if the inlet of the pipe is at the bottom of a tank etc.)

$W_{in} = F_{in} \cdot dx_{in}$ , or

$$W_{in} = \left(\frac{F_{in}}{A_{in}}\right) \cdot A_{in} \cdot U_{in} \cdot dt = P_{in} \cdot A_{in} \cdot U_{in} \cdot dt \quad (2)$$

Where $U_{in}$ is the speed of the mass of the liquid at the entrance of the pipe and $P_{in}$ is the pressure at the inlet section of the pipe and $A_{in}$ is the area of the section. The volume $dV$ of the liquid is given by the expression.

$$dV = A_{in} \cdot U_{in} \cdot dt \quad (3)$$

So the equation can be rewritten as

$$W_{in} = P_{in} \cdot dV \quad (4)$$

By adding the density of the liquid into the equation, $\rho = dm/dV$ , it is rewritten as

$$W_{in} = \frac{P_{in} \cdot dm}{\rho} \quad (5)$$

- $PE_{in}$ is the potential energy of the mass of the liquid that enters the pipe for a small period of time $dt$.

$$PE_{in} = dm \cdot g \cdot h_{in} \quad (6)$$

Where $dm$ is the mass of the liquid, $g$ is the gravitational acceleration equal to 9.81 and $h_{in}$ is the height of the center of the mass of the liquid from a reference level.

- $KE_{in}$ is the kinetic energy of the mass of the liquid that enters the pipe for a small period of time, $dt$.

$$KE_{in} = \frac{1}{2} \cdot dm \cdot {U_{in}}^2 \quad (7)$$

- The same applies to the outlet of the pipe ($W_{out}$, $PE_{out}$ and $KE_{out}$).

Taking all the above into consideration, the Bernoulli equation derives:

$$P_{in} + \rho \cdot g \cdot h_{in} + \frac{\rho \cdot {U_{in}}^2}{2} = P_{out} + \rho \cdot g \cdot h_{out} + \frac{\rho \cdot {U_{out}}^2}{2} \quad (8)$$

### 2.1.2 *Pressure losses due to friction in the pipe's wall*

In order to include the pressure loss due to friction in the pipes walls, the Darcy-Weisbach formula is used.

$$dP_i = f \cdot \frac{L_i \cdot \rho \cdot U^2}{2 \cdot d} \quad (9)$$

Where $L_i$ is the length of the pipe, $d$ is the diameter and $f$ is the friction loss coefficient which depends on Reynold's number and the relative roughness ($\varepsilon/d$) of the pipe wall, where $\varepsilon$ is the roughness of the pipe. The Reynolds number is calculated with the following formula:

$$Re = \frac{U \cdot d}{\nu} \quad (10)$$

Where $\nu$ is the kinematic viscosity of the liquid.
With those two factors defined, the solution of the Colebrook equation gives the friction coefficient $f$.

$$\frac{1}{\sqrt{f}} = -2 \cdot \log\left(\frac{\varepsilon/d}{3.7} + \frac{2.51}{Re \cdot \sqrt{f}}\right) \quad (11)$$

If $Re<2300$ then $f=64/Re$

Also, in order to include pressure losses due to other components (e.g. valves) the equation below is used

$$dP_j = n_j \cdot \frac{\rho \cdot U^2}{2} \quad (12)$$

Where $n_j$ is the added resistance coefficient and it is different for every component (Papantonis 1998).

### 2.1.3 *Application in a simple hydraulic system*

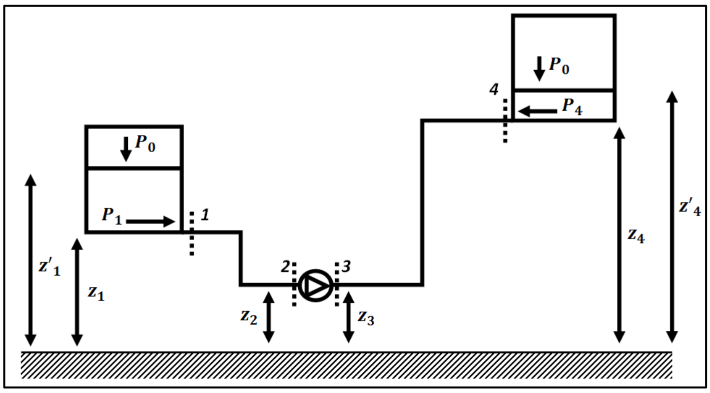


Figure 2. Simple hydraulic system

In the above system, a pump is used to transfer liquid from one tank to another via pipes. By dividing the system into three parts (from 1 to 2, from 2 to 3 and from 3 to 4), the Bernoulli equation can be applied in those three parts (Theotokatos 2007). The Bernoulli equation is divided by (ρ·g) in order to express the pressure in meters of liquid column.

$$\frac{P_1}{\rho \cdot g} + z_1 + \frac{{U_1}^2}{2 \cdot g} = \frac{P_2}{\rho \cdot g} + z_2 + \frac{{U_2}^2}{2 \cdot g} + \frac{dP_{12}}{\rho \cdot g} \quad (13)$$

$$\frac{P_2}{\rho \cdot g} + z_2 + \frac{{U_2}^2}{2 \cdot g} + \frac{P_{23}}{\rho \cdot g} = \frac{P_3}{\rho \cdot g} + z_3 + \frac{{U_3}^2}{2 \cdot g} \quad (14)$$

$$\frac{P_3}{\rho \cdot g} + z_3 + \frac{{U_3}^2}{2 \cdot g} = \frac{P_4}{\rho \cdot g} + z_4 + \frac{{U_4}^2}{2 \cdot g} + \frac{dP_{34}}{\rho \cdot g} \quad (15)$$

Where $P_i$ is the pressure in the point $i$ and is calculated:

$$P_i = P_0 + \rho \cdot g \cdot (z_i' - z_i) \quad (16)$$

By adding all the right-hand sides of the equations and all the left-hand sides, the equation below derives:

$$\frac{P_1}{\rho \cdot g} + z_1 + \frac{{U_1}^2}{2 \cdot g} + \frac{P_{23}}{\rho \cdot g} = \frac{P_4}{\rho \cdot g} + z_4 + \frac{{U_4}^2}{2 \cdot g} + \frac{dP_{12} + dP_{34}}{\rho \cdot g} \quad (17)$$

$$\frac{P_{23}}{\rho \cdot g} = \frac{P_4 - P_1}{\rho \cdot g} + z_4 - z_1 + \frac{{U_4}^2 - {U_1}^2}{2 \cdot g} + \frac{dP_{12} + dP_{34}}{\rho \cdot g} \quad (18)$$

So, for a specific flow rate, the velocities $U_i$ are calculated:

$$U_i = \frac{Q}{A_i} = \frac{Q}{\left(\pi \cdot \frac{{d_i}^2}{4}\right)} = \frac{4 \cdot Q}{\pi \cdot {d_i}^2} \quad (19)$$

Also, the friction loss coefficient derives as shown before and the pressure drop $dP_{12}$ and $dP_{34}$ are calculated. The quantity $H_{system}=P_{23}/\rho \cdot g$ is the required manometric head of the system in order to achieve the requested flow rate Q.
By plotting $H_{system}$ for different flow rates, a curve derives which expresses the characteristic curve of the system. If there is a pump in the system, the characteristic curve of the pump is provided by the manufacturer. The pump curve describes the relation between flow rate and head for the pump (Ioannidis 1996). The flow rate at which the system operates derives from the intersection of the two characteristic curves as shown below.

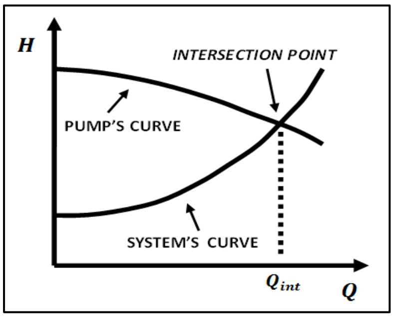


Figure 3. Operation point

## 2.2 *Dealing with complex hydraulic networks*

When dealing with complex hydraulic networks, the definition of the flow rate at each branch can be calculated by using the Linear Method (Jeppson 1974).

### 2.2.1 *Linear Method*

By rewriting the equation for pressure loss due to friction in the pipes walls and the equation for pressure losses due to other components, the following equations derive.

$$dP_i = \frac{f \cdot L_i \cdot \rho \cdot U^2}{2 \cdot d} = \frac{f \cdot L_i \cdot \rho \cdot \left(\frac{4 \cdot Q}{\pi \cdot d^2}\right)^2}{2 \cdot d} = \left(\frac{8 \cdot f \cdot L_i}{g \cdot \pi^2 \cdot d^5}\right) \cdot Q^2 = R_i \cdot Q^2 \quad (20)$$

$$dP_j = \frac{n_j \cdot \rho \cdot U^2}{2} = \frac{n_j \cdot \rho \cdot \left(\frac{4 \cdot Q}{\pi \cdot d^2}\right)^2}{2} = \left(\frac{8 \cdot n_j}{g \cdot \pi^2 \cdot d^4}\right) \cdot Q^2 = R_j \cdot Q^2 \quad (21)$$

Where

$$R_i = \frac{8 \cdot f \cdot L_i}{g \cdot \pi^2 \cdot d^5} \quad (22)$$

$$R_j = \frac{8 \cdot n_j}{g \cdot \pi^2 \cdot d^4} \quad (23)$$

The system is subdivided into different branches that are connected via junction nodes. The flow rate of each branch is unknown. The procedure for calculating the flow rates is as follows:

- Arbitrary directions of the flow are defined for every branch. Also, an arbitrary flow rate is assumed at every branch. Usually, the flow rate is taken Q=$1m^3/sec$ for every branch.
- For every branch in the network the value $R_i$ is calculated for the assumed flow rate.
- For every junction node (e.g. connection of different branches) a mass conservation equation is formed.

$$\sum Q_{in} = \sum Q_{out} \quad (24)$$

- For Every closed loop of pipes in the network, an energy equation is formed which ensures that from a single point of the loop, the energy losses when 'travelling' around the loop, back to this particular point, must be zero.

$$\sum R_i \cdot {Q_i}^2 = 0 \quad (25)$$

- For each branch, a linearization factor $a_i$ is calculated.

$$a_i = R_i \cdot |Q_i| \quad (26)$$

  So, the energy equations are rewritten as follows:

$$\sum R_i \cdot |Q_i| \cdot Q_i' = 0 \quad (27)$$

  Where $Q_i'$ is the new flow rates that will derive after solving the formed linear system.
- In the next iterations, $Q_{i(n)}=Q_{i(n-1)}$ for the second iteration and $Q_{i(n)}=(Q_{i(n-1)}+Q_{i(n-2)})/2$ for the rest iterations.

The method presented above is used for solving hydraulic networks, but can't be applied if there are tanks or pumps in the network. Therefore, a modification of the method is necessary in order to take into account those components.

### 2.2.2 *Inclusion of tanks*

In order for tanks to be included in the system, an assumption is made (Jeppson 1974) that there is a pseudo-loop which consists of a route between two tanks, and a 'no flow' pipe between those two tanks. The energy equation in this case takes into account the difference in the pressure between the two tanks.

$$\sum R_i \cdot {Q_i}^2 = H_1 - H_2 \quad (28)$$

### 2.2.3 *Inclusion of pumps*

Every pump's characteristic curve must be expressed by a quadratic equation.

$$h_p = A \cdot Q^2 + B \cdot Q + H_0 \quad (29)$$

In order to insert the pump into the created linear system o equations, the following transformation must be made:

$$G = Q + \frac{B}{2 \cdot A} \tag{30}$$

So, the pump's characteristic curve is rewritten as follows.

$$h_p = A \cdot G^2 + h_0 \tag{31}$$

Where

$$h_0 = H_0 - \frac{B^2}{4 \cdot A} \tag{32}$$

In summary, if pumps and tanks exist in a pipe network, and there N pipe branches, whose flow rate must be calculated, the linear theory is applied as follows (Jeppson 1974):

- J linear junction continuity equations are formed.
- L nonlinear energy equations are formed around real loops in the piping network
- K additional pseudo-loops are defined by 'no flow' pipes between tanks and energy equations are written around these pseudo-loops. The number of these pseudo-loops must equal the difference between the number of unknown flow rates. e.g. K=N-(J+L)
- M additional pump transformation equations are formed, where M is the number of the pumps.
- The nonlinear equation are linearized with the use of the linearization factor *a*. For the pumps, factor *a* is calculated as follows:

$$a = A_i \cdot |G_i| \tag{33}$$

The linear system is solved iteratively until convergence occurs. Below, a simple example is presented that will demonstrate how the above methodology is applied.

### 2.2.4 *Example*

Text In figure 4, there are two pumps connected in parallel, and are used to transfer liquid from the tank

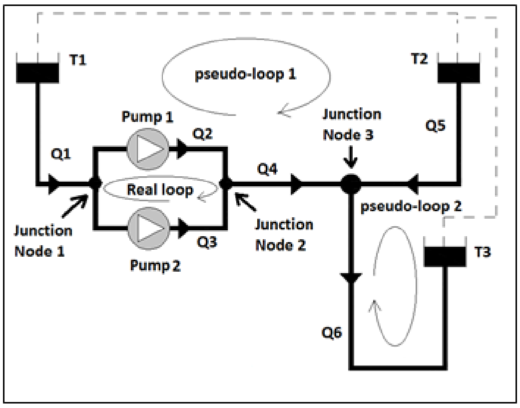


Figure 4. Example

T1, to tanks T2 and T3. The characteristic curves of the pumps are known. The problem is to define the flow rates Q1 to Q6.

The first step is to define arbitrary directions of the flow at every branch. Then, the real loops and the pseudo-loops are defined, with the clockwise direction considered arbitrarily as 'positive'. The equations are formed as follows:

- Number of junction nodes: 3

$$Q_1 - Q_2 - Q_3 = 0 \tag{34}$$

$$Q_2 + Q_3 - Q_4 = 0 \tag{35}$$

$$Q_4 + Q_5 - Q_6 = 0 \tag{36}$$

- Number of loops: 1

$$R_2 \cdot {Q_2}^2 - A_1 \cdot {G_1}^2 - R_3 \cdot {Q_3}^2 + A_2 \cdot {G_2}^2 = h_{01} - h_{02} \tag{37}$$

Where

$$h_{01} = H_1 - \frac{{B_1}^2}{4 \cdot A_1} \tag{38}$$

$$h_{02} = H_2 - \frac{{B_2}^2}{4 \cdot A_2} \tag{39}$$

- Number of pseudo-loops: 2

$$R_5 \cdot {Q_5}^2 - R_4 \cdot {Q_4}^2 - R_2 \cdot {Q_2}^2 + A_1 \cdot {G_1}^2 - R_1 \cdot {Q_1}^2 =$$

$$= H_{T2} - H_{T1} - h_{01} \tag{40}$$

$$-R_6 \cdot {Q_6}^2 - R_5 \cdot {Q_5}^2 = H_{T3} - H_{T2} \tag{41}$$

Where $H_{Ti}$ is the height of the level of tank *i* from a reference level.

- Transformation equations: 2

$$G_1 = Q_2 + \frac{B_1}{2 \cdot A_1} \tag{42}$$

$$G_2 = Q_3 + \frac{B_2}{2 \cdot A_2} \tag{43}$$

- Calculation of the linearization factor $a_i$

$a_i = R_i \cdot |Q_i|$ , *i*=1 to 6

$a_7 = A_1 \cdot |G_1|$

$$a_8 = A_2 \cdot |G_2| \tag{44}$$

- The linear system is formed as follows:

$$\begin{pmatrix} 1 & -1 & -1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 1 & -1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 1 & -1 & 0 & 0 \\ 0 & a_2 & -a_3 & 0 & 0 & 0 & -a_7 & a_8 \\ -a_1 & -a_2 & 0 & -a_4 & a_5 & 0 & a_7 & 0 \\ 0 & 0 & 0 & 0 & -a_5 & -a_6 & 0 & 0 \\ 0 & -1 & 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & -1 & 0 & 0 & 0 & 0 & 1 \end{pmatrix} \begin{pmatrix} Q_1 \\ Q_2 \\ Q_3 \\ Q_4 \\ Q_5 \\ Q_6 \\ G_1 \\ G_2 \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \\ 0 \\ h_{01} - h_{02} \\ H_{T2} - H_{T1} - h_{01} \\ H_{T3} - H_{T2} \\ B_1/(2 \cdot A_1) \\ B_2/(2 \cdot A_2) \end{pmatrix} \tag{45}$$

- For the next iterations: $Q_{i(n)}=Q_{i(n-1)}$ for the second iteration and $Q_{i(n)}=(Q_{i(n-1)}+Q_{i(n-2)})/2$ for the rest iterations.

- The procedure is repeated until convergence occurs.
- The derived flow rates are considered constant for a small period of time (time-step), and the new levels of the tanks are calculated according to those flow rates.

The procedure is repeated with the new levels of the tanks. New flow rates derive and new levels of tanks are calculated at every time-step.

## 3 APPLICATION IN A DISCHARGING PROCEDURE OF A TANKER

As a verification of the developed simulation tool, the homogenous discharging procedure of a tanker will be simulated and result will be compared with available operational data. Two systems of the ship will be working simultaneously; the cargo system and the ballast system.

### 3.1 *Vessel information*

The ship examined is a 105000 DWT Aframax, with 6x2 cargo tanks (six port and six starboard). The hydrostatic table of the ship is imported in the code in order to calculate its response during the procedure.

### 3.2 *Cargo System*

In the examined scenario, three cargo pumps will be used to discharge oil from all the cargo tanks (Cargo tanks 1 to 6, port and starboard).

#### 3.2.1 *Pumps*

The vessels is equipped with steam driven cargo pumps. Their characteristic curve is given by the manufacturer, but it refers to water. Therefore, it needs to be adapted to the different viscosity of cargo oil (ANSI HI 2004). Also, the characteristic curves for different RPM are calculated according to pump affinity laws (Lobanoff & Ross 1985):

$$Q_2 = Q_1 \left(\frac{RPM_2}{RPM_1}\right) \tag{46}$$

$$H_2 = H_1 \left(\frac{RPM_2}{RPM_1}\right)^2 \tag{47}$$

The pump's characteristic curves are expressed in quadratic form.

#### 3.2.2 *Piping*

A simplified diagram of the cargo piping network is presented in figure 5. The description of the symbols for the both the cargo and ballast system are presented in table 1. The state of the valves (open/close) is defined by the user. For the visualization of the arrangements, as necessary in the frame of CAD, numerous software packages were explored and practically examined (Pytharoulis 2013). Below, the piping diagram has been visualized with a visualization routine of RHINOCEROS® software, developed for the REFRESH project.

Table 1. Symbol Description

| SYMBOL | DESCRIPTION |
|---|---|
| | Pump (P) |
| | Valve |
| | Separator Tank (S) |
| | Local Added Resistance (RA,RB) |
| | Box Type Strainer (ST) |
| SC | Sea Chest |
| M | Manifold |
| SLOP T. | Slop Tank |
| C.O.T. | Cargo Oil Tank |
| W.B.T. | Water Ballast Tank |
| P&S | Port & Starboard |

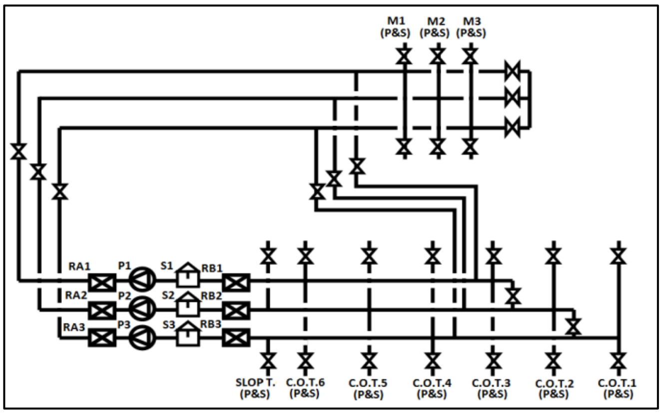


Figure 5. Cargo Piping Diagram

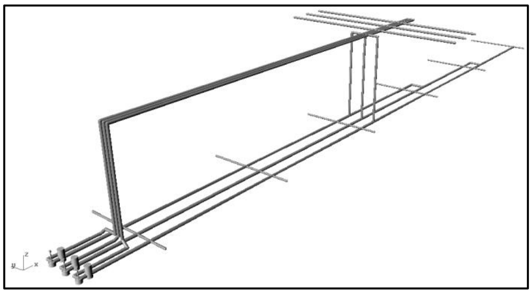

Figure 6. 3D Cargo piping network

#### 3.2.3 *Tanks*

The level, the corresponding volume and the longitudinal center of gravity for each tank is calculated using the tank's calibration tables.

#### 3.2.4 *Fluid properties*

The examined liquid has density 875 $kg/m^3$ and kinematic viscosity 60 cst.

#### 3.2.5 *Initial conditions and shore tank information*

The initial volume of each ship tank is shown in table 2. The pressure at every tank is 700 mmHg.

table 2. Cargo tanks initial volume

| Tank | Ullage (m) | C. Mtrs |
|---|---|---|
| C.O.T. No.1 (p) | 7.7 | 5114.7 |
| C.O.T. No.1 (s) | 7.7 | 5109.4 |
| C.O.T. No.2 (p) | 5.7 | 7806.3 |
| C.O.T. No.2 (s) | 5.8 | 7726.4 |
| C.O.T. No.3 (p) | 6.1 | 7577.8 |
| C.O.T. No.3 (s) | 6.2 | 7564.2 |
| C.O.T. No.4 (p) | 12.9 | 3863.1 |
| C.O.T. No.4 (s) | 12.9 | 3852.1 |
| C.O.T. No.5 (p) | 8.0 | 6553.7 |
| C.O.T. No.5 (s) | 8.0 | 6556.2 |
| C.O.T. No.6 (p) | 10.9 | 3507.2 |
| C.O.T. No.6 (s) | 10.5 | 3681.3 |
| Slop T. (p) | 9.8 | 564.4 |
| Slop T. (s) | 9.9 | 560.4 |

For the shore tanks, the information provided by the shore officer is limited. The distance of the tanks from the vessel's manifold is 1.5 miles and the height above the sea level is 50 ft. The horizontal plane area of each shore tank is approximately 4000 $m^2$. Because of the limited information for the shore piping, the network outside of the ship (pipes, tanks etc.) is simplified. It is assumed that the piping consists of one pipe with length equal to the one given by the shore officer and a tank with its level at the given height of the tanks. That applies to all the manifolds, therefore, in our case, there are two pipes connected to the two manifolds of the vessel, with two shore tanks, one for each manifold. The diameter of those pipes is defined with the use of existing pumping logs. Using a diameter 0.7 meters for the shore pipes, the pressure at the manifold is equal to the one that was measured during the procedure (approximately 5 kg/cm$^2$) when the three cargo pumps where working on 1050 RPM. Other constraints that are imposed by the shore officer are the maximum allowable discharging rate which is 2.2 m$^3$/sec and the maximum allowable pressure which is 10.5 bar = 10.71 kg/cm$^2$.

## 3.3 *Ballast System*

In the examined scenario, the ballasting at the beginning will be accomplished without the use of any pump. This will be accomplished by taking advantage of the difference in the height of the ballast tanks level and the sea level. After a period of time, two ballast pumps will be activated to continue the ballasting procedure.

### 3.3.1 *Pumps*

There are two ballast pumps driven by electric motors. Their characteristic curves are imported in quadratic form. The procedure is the same that was applied in the cargo pumps.

### 3.3.2 *Piping*

A simplified ballast piping diagram is presented in figure 7.

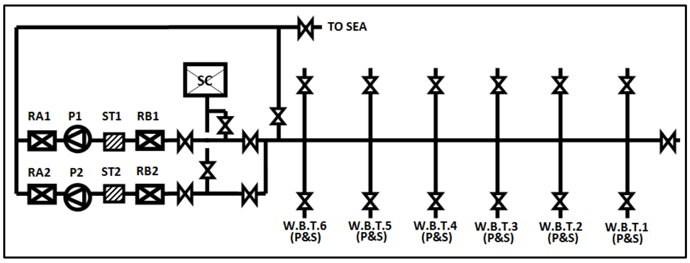


Figure7. Ballast piping diagram

The state of the valves (open/close) is defined by the user according to the examined scenario. Again, the above piping diagram has been visualized by using a visualization routine developed in RHINOCEROS Software, for the REFRESH project.

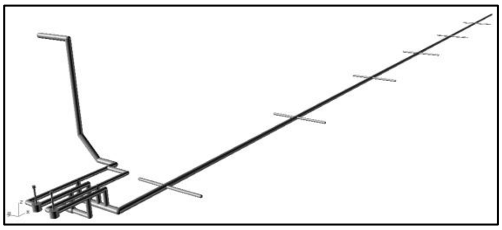

Figure 8. 3D Ballast piping network

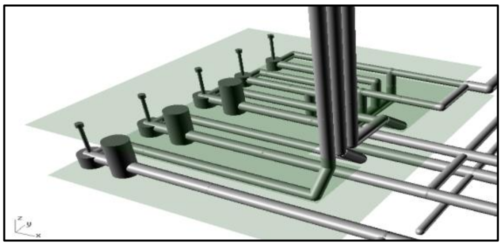

Figure 9. Simplified pump room arrangement

### 3.3.3 *Tanks*

The ballast tanks calibration tables are imported and linear interpolations are performed in order to calculate the necessary data at every time-step.

### 3.3.4 *Fluid properties*

The liquid of the system is sea water with density 1025 $kg/m^3$ and kinematic viscosity 1.17 cst.

### 3.3.5 *Initial conditions*

All the ballast tanks are empty at the beginning of the discharging procedure.

## 3.4 *Special considerations*

### 3.4.1 *Draft and trim*

During the procedure, the ship's draught and trim change constantly, thus causing a decrease to the overall manometric height, since the ship's mean draught is constantly decreasing during the procedure (Adamopoulos 2012). The trim by stern of the ship facilitates the procedure (Gunner 2001). Therefore, the trim should be kept near its maximum allowable value. The maximum allowable trim of the vessel is 3.7 meters by stern (or 1.6%L). The added head of the No.1 Cargo tank, $\Delta h$, due to change in trim and draft can be observed in figure 10.

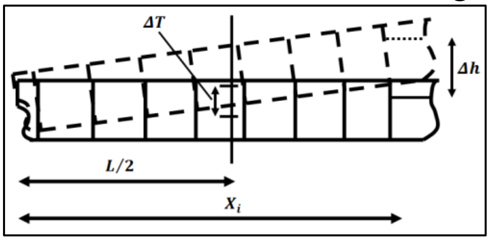


Figure 7. Added head due to draft and trim

$$\Delta h = \Delta T + \left(X_i - \frac{L}{2}\right) \cdot \frac{-Trim}{L} \qquad (48)$$

Where $\Delta T$ is the change in the mean draft of the vessel, $X_i$ is the longitudinal center of gravity of each tank, $L$ is the length of the vessel and the trim is the difference between the drafts at fore peak and after peak e.g. $Trim = T_{fore} - T_{aft}$

### 3.4.2 *NPSH*

An important parameter that should always be checked is the Net Positive Suction Head (NPSH) of each pump in order to ensure that it is over the lower limit which is given by the manufacturer of the pump. The actual NPSH is calculated and it can be compared with the minimum required value if it is available.

### 3.4.3 *Bending moments and shear forces*

The bending moments and the shear forces of the ship must never exceed the maximum values that are defined in the stability booklet of the vessel. In the selected scenario the discharging and ballasting procedure are performed uniformly from all cargo and ballast tanks in order to ensure that we are within the maximum allowable shear forces and bending moments.

## 4 RESULTS OF THE SIMULATION

table 3. Cargo pumping log

| | | C.P. No. 1 | C.P. No. 2 | C.P. No. 3 | man. no.1 | man. no.2 | R.O.B. | DISCHARGED (bbls) | |
|---|---|---|---|---|---|---|---|---|---|
| hours | RPM | $Kgs/cm^2$ | $Kgs/cm^2$ | $Kgs/cm^2$ | $Kgs/cm^2$ | $Kgs/cm^2$ | bbls | STEP | TOTAL |
| 0 | 1050 | 7.7 | 7.6 | 7.5 | 5.1 | 5.1 | 440520 | 0 | 0 |
| 1 | 1050 | 7.7 | 7.7 | 7.7 | 5.2 | 5.2 | 390064 | 50455 | 50455 |
| 2 | 1050 | 7.7 | 7.7 | 7.7 | 5.2 | 5.2 | 340085 | 49978 | 100434 |
| 3 | 1050 | 7.7 | 7.6 | 7.7 | 5.2 | 5.2 | 291071 | 49014 | 149449 |
| 4 | 1050 | 7.7 | 7.6 | 7.6 | 5.2 | 5.2 | 242561 | 48509 | 197958 |
| 5 | 1050 | 7.7 | 7.6 | 7.6 | 5.2 | 5.2 | 194870 | 47691 | 245650 |
| 6 | 1050 | 7.7 | 7.6 | 7.6 | 5.2 | 5.2 | 148139 | 46801 | 292451 |
| 7 | 1050 | 7.6 | 7.6 | 7.6 | 5.2 | 5.2 | 102081 | 46058 | 338509 |
| 8 | 1050 | 7.6 | 7.6 | 7.6 | 5.2 | 5.2 | 56827 | 45253 | 383763 |

table 4.Tanks handling plan

| hours -> | 0 | | 2 | | 4 | | 6 | | 8 | |
|---|---|---|---|---|---|---|---|---|---|---|
| Tank | Ullage (m) | C. Mtrs | Ullage (m) | C. Mtrs | Ullage (m) | C. Mtrs | Ullage (m) | C. Mtrs | Ullage (m) | C. Mtrs |
| C.O.T. No.1 (p) | 7.7 | 5114 | 11.6 | 3409 | 14.0 | 2366 | 16.3 | 1392 | 19.3 | 265 |
| C.O.T. No.1 (s) | 7.7 | 5109 | 11.7 | 3357 | 14.0 | 2345 | 16.4 | 1362 | 19.4 | 242 |
| C.O.T. No.2 (p) | 5.7 | 7806 | 11.1 | 4802 | 13.8 | 3359 | 16.1 | 2069 | 18.9 | 581 |
| C.O.T. No.2 (s) | 5.8 | 7726 | 11.3 | 4738 | 13.8 | 3337 | 16.2 | 2043 | 19.0 | 555 |
| C.O.T. No.3 (p) | 6.1 | 7577 | 11.0 | 4926 | 13.6 | 3460 | 16.1 | 2099 | 18.7 | 685 |
| C.O.T. No.3 (s) | 6.2 | 7564 | 10.9 | 4944 | 13.6 | 3469 | 16.1 | 2108 | 18.7 | 693 |
| C.O.T. No.4 (p) | 12.9 | 3863 | 11.5 | 4644 | 13.5 | 3519 | 15.9 | 2178 | 18.3 | 910 |
| C.O.T. No.4 (s) | 12.9 | 3852 | 11.5 | 4635 | 13.6 | 3463 | 16.0 | 2128 | 18.4 | 861 |
| C.O.T. No.5 (p) | 8.0 | 6553 | 10.4 | 5241 | 13.3 | 3655 | 15.9 | 2196 | 18.1 | 984 |
| C.O.T. No.5 (s) | 8.0 | 6556 | 10.4 | 5213 | 13.3 | 3622 | 15.9 | 2173 | 18.2 | 964 |
| C.O.T. No.6 (p) | 10.9 | 3507 | 10.9 | 3514 | 13.5 | 2405 | 16.2 | 1312 | 18.4 | 558 |
| C.O.T. No.6 (s) | 10.5 | 3681 | 10.9 | 3517 | 13.5 | 2410 | 16.2 | 1315 | 18.4 | 560 |
| Slop T. (p) | 9.8 | 564 | 9.8 | 564 | 9.7 | 576 | 9.5 | 587 | 9.5 | 587 |
| Slop T. (s) | 9.9 | 560 | 9.9 | 560 | 9.7 | 572 | 9.6 | 584 | 9.6 | 584 |
| **NOTES** | BALLASTING BY GRAVITY TO BALLAST T. No.1-6 (P&S) | | | | BAL/ING WITH 2 PUMPS TO B.T. No.1-5 | | NO BALLASTING | | | |
| Tank | Sounding (m) | C. Mtrs | Sounding (m) | C. Mtrs | Sounding (m) | C. Mtrs | Sounding (m) | C. Mtrs | Sounding (m) | C. Mtrs |
| Fore Peak | 0.2 | 0 | 0.2 | 0 | 0.4 | 12 | 0.5 | 23 | 0.5 | 23 |
| W.B.T. No.1 (p) | 0.2 | 35 | 1.5 | 563 | 13.2 | 1990 | 18.6 | 2458 | 18.6 | 2458 |
| W.B.T. No.1 (s) | 0.2 | 36 | 1.5 | 560 | 13.2 | 1990 | 18.6 | 2457 | 18.6 | 2457 |
| W.B.T. No.2 (p) | 0.1 | 25 | 1.8 | 1063 | 13.5 | 2302 | 19.4 | 2731 | 19.4 | 2731 |
| W.B.T. No.2 (s) | 0.1 | 24 | 1.8 | 1056 | 13.5 | 2301 | 19.4 | 2731 | 19.4 | 2731 |
| W.B.T. No.3 (p) | 0.1 | 22 | 2.8 | 1379 | 13.9 | 2398 | 19.7 | 2823 | 19.7 | 2823 |
| W.B.T. No.3 (s) | 0.1 | 26 | 2.8 | 1374 | 13.9 | 2397 | 19.9 | 2838 | 19.9 | 2838 |
| W.B.T. No.4 (p) | 0.1 | 24 | 3.7 | 1575 | 14.4 | 2435 | 20.1 | 2852 | 20.1 | 2852 |
| W.B.T. No.4 (s) | 0.1 | 30 | 3.7 | 1574 | 14.4 | 2435 | 20.1 | 2852 | 20.1 | 2852 |
| W.B.T. No.5 (p) | 0.1 | 57 | 4.9 | 1631 | 15.1 | 2399 | 20.5 | 2794 | 20.5 | 2794 |
| W.B.T. No.5 (s) | 0.1 | 61 | 4.9 | 1631 | 15.1 | 2399 | 20.5 | 2794 | 20.5 | 2794 |
| W.B.T. No.6 (p) | 0.2 | 43 | 6.1 | 1387 | 7.0 | 1510 | 7.1 | 1525 | 7.1 | 1525 |
| W.B.T. No.6 (s) | 0.1 | 26 | 6.1 | 1387 | 6.9 | 1502 | 7.0 | 1510 | 7.0 | 1510 |
| Displacement (ton) | 83111.9 | | 84278.6 | | 81875.2 | | 73165.2 | | 60461.9 | |
| Trim (m) | 0.8 | | -3.6 | | -2.0 | | -2.1 | | -3.7 | |
| Mean Draft (m) | 10.4 | | 10.6 | | 10.4 | | 9.4 | | 7.9 | |

In the examined procedure, cargo is discharged from C.O. Tanks 1 to 6. All the cargo pumps are working constantly at 1050 RPM. The ballasting is performed by gravity at the beginning of the procedure (no use of pumps), in W.B. Tanks 1 to 6. And then after 2.5 hours, all the ballast pumps start working at 1180 RPM, and the ballasting occurs in W.B. Tanks 1 to 5. At 5.5 hours, the ballasting procedure stops, but the discharging procedure of the cargo continues. The procedure is performed manually e.g. the discharging is performed with a specified ballast scenario and when a constraint is violated (e.g. violating trim constraint) the procedure is terminated, a different ballast scenario is imported, and then procedure is initialized again with the condition that it was terminated as an input. The pumping log and the handling plan of the simulation are presented above and include the following information:

- RPM of each pump
- The discharge pressure of each pump (*$kg/cm^2$*)
- The pressure at each manifold (*$kg/cm^2$*)
- The volume of the cargo remaining on board (R.O.B.), the volume that has been discharged between the different hours in the log (STEP) and the overall discharged volume of cargo (TOTAL) in *bbls*.

The estimated hydraulic energy is:

- 3596.9 kWh for C.O. Pump No.1
- 3634.4 kWh for C.O. Pump No.2
- 3633.8 kWh for C.O. Pump No.3

– 234.3 kWh for W.B. Pump No.1
– 232.8 kWh for W.B. Pump No.2

Note that the term hydraulic energy is used to express the energy, that the hydraulic system demands from the pump and not the overall consumed energy. The energy demand for each subsystem is presented in table 5.

table 5.Energy demand for each subsystem

| SUBSYSTEM | ENERGY DEMAND (kWh) |
|---|---|
| Cargo | 10865.2 |
| Ballast | 467.1 |
| Overall | 11332.2 |

## 5 OPTIMIZATION PROSPECTS

With the developed tool, two basic types of optimization can be performed:

– Discrete Optimization Problem: Optimization of system's topology/arrangement (adding or removing pumps, change of capacities etc.)
– Continuous Optimization Problem: Optimization of system's performance (e.g. pump RPM etc.)

In figure 11, a discharging scenario, similar to the one examined in this paper, was tested for two different arrangements (use of two or three pumps for the discharging) and for different RPM. The required hydraulic energy at each case has been plotted against the RPM of the pumps.

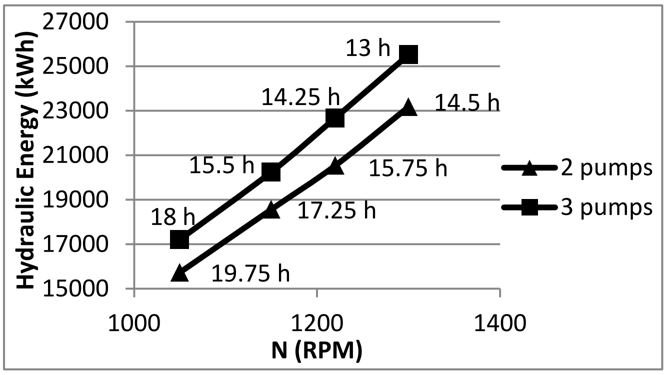


Figure 8. Hydraulic Energy – Pumps RPM

The time for the procedure to be completed is shown next to each point on the graph. In general, the minimization of the required hydraulic energy in respect to the time spent at the port for discharging is a more complex optimization problem; it can be integrated within a multi-objective optimization procedure that deals with the overall, life-cycle energy optimization of the tanker; in more simplified cases the optimization of specific loading/discharging scenarios for least energy consumption and minimum time at port will be targeted.

## 6 SUMMARY-CONCLUSIONS

The mathematical modeling of a developed simulation tool for loading and discharging procedures of tankers has been presented. The simulation tool was developed in MATLAB and the simulation of one hour of the procedure takes approximately two and a half minutes in a conventional PC. The tool was used in a typical specified discharging scenario has been examined and the results are presented in the form of pumping log and tanks handling plan. The hydraulic energy of each subsystem (cargo and ballast) is also calculated. Presented simulation results agree well with available operational data confirming the validity of the developed tool and enabling the optimization of ship's operation with respect to the energy efficiency of affected onboard hydraulic systems.

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