23rd ABCM International Congress of Mechanical Engineering
December 6-11, 2015, Rio de Janeiro, RJ, Brazil

# STEEL TOWING TANK DESIGN USING LOW VIBRATION AS A DESIGN CRITERIA

**Andressa Beckert Otto**
**Thiago Pontin Tancredi**
Universidade Federal de Santa Catarina – Centro de Engenharias da Mobilidade – Campus Joinville
andressa.beckert@hotmail.com, thiago.tancredi@ufsc.br

***Abstract.*** *This paper presents the development of the conceptual design of a didactic towing tank to be used as a laboratory for different fields of engineering. The main contribution is the design procedure that determines the tank characteristics based on natural frequencies of vibration that minimize the vibration that would distort the results of experiments. An important feature is the utilization of the steel to build the tank, a non-usual procedure in these designs.*
*The paper presents the procedure to determination of the main dimensions of the tank, based on ITTC Standards and the structural design based on the analysis of the vibration level, exploring also different design configurations.*
*An optimization procedure is used to define the design characteristics that are analyzed with the Element Finite Method. The results describe the influence of each design parameter and the consideration of the water pressure and the dynamic pressure resulted of the experiments.*

***Keywords:*** *Towing Tank; Vibration; Design; Optimization.*

## 1. INTRODUCTION

A towing tank can be define as a place to hydrodynamic test with reduced scale models, in order to simulate the behavior of the sea and ship dynamics. These tests are essential to determine hydrodynamic coefficients that can be extrapolated for corresponding parameters in the full size ship (Muñoz, 2011).

Among the different types of test tanks, the towing tank generally employs a sufficiently large canal for conducting tests with models, a dynamometric car that controls the model movements and measures forces, and a wave generator that replicates the irregular sea wave characteristics.

In this paper it is proposed a design of a towing tank that will be made with steel, using a similar structural arrangement those seen in ships. And in this case, as towing tanks are susceptible to periodic excitations from wave generation and from the dynamometric car's movement, a detailed vibration mode analysis is important in the design phase of the project in order to obtain a control over the vibrations and, with that, avoid resonant frequencies that can eventually influence test results, beside to cause fatigue in the structure.

Correct dimensioning of the tank is important, because smaller tanks require smaller models, which results in greater difficulty to match potential and viscous effects that occur during a reduced scale test. On the other hand, large tanks present higher operational and maintenance costs.

This work is proposed to develop a towing tank design to be built at the Joinville Campus of the Universidade Federal de Santa Catarina, with the minimum dimensions that comply with test requirements.

## 2. CONSIDERATIONS FOR TOWING TANK DESIGN

The dimensioning, construction and operation of a towing tank is a complex subject that is widely studied by the International Towing Tank Conference (ITTC). This work strongly references to ITTC recommendations and works of several authors that discuss said recommendations, as example (Zuleta, 2014) and (Muñoz, 2011).

### 2.1. Froude Number

Froude number relates inertial forces (associated with the acceleration of the fluid) with the gravitational force (weight of the fluid due to gravity). This dimensionless parameter used by the ITTC is calculated through Equation (1).

$$F_r = \frac{V}{\sqrt{gL}} \tag{1}$$

Where "V" is the velocity, "g" the gravitational acceleration, and "L" is the length.

## 2.2. Depth Froude Number

The Froude number of depth is related to the depth of the tank $(H_{tanque})$. According to Molland, Turnock and Hudson (2011 p.26), this parameter is calculated as:

$$F_{r_h} = \frac{V}{\sqrt{gH}} \tag{2}$$

For towing tank design this value must be less than 0.7 for the speeds used in the towing tests, otherwise the measured values will not obey the deep water hypothesis.

## 2.3. Similarity laws

Dimensioning of a towing tank requires knowledge about the similarity criteria used between the model and the real ship. In scale tests, similarity is needed for geometric, kinematic and dynamic characteristics relating to the full-size ship. According to Garcia (2014), geometric similarity is determined through the scale factor, characterized by the ration between the length of the ship and the model, as in equation (3)

$$\alpha = \frac{L_s}{L_m} \tag{3}$$

The scale factor enables the extrapolation of several parameters of the full-size ship to the model, or vice-versa. In Table 1 some of the geometric, kinematic and dynamic similarity cases used in this study are shown.

Table 1. Scale factor due to geometric, kinematic and dynamic similarities between model and full-size ship.

| Similarity | Variable | Unit | Scale factor |
|---|---|---|---|
| Geometric | Length | L | $\alpha$ |
| | Area | $L^2$ | $\alpha^2$ |
| | Volume | $L^3$ | $\alpha^3$ |
| Kinematic | Time | T | $\alpha^{1/2}$ |
| | Velocity | $LT^{-1}$ | $\alpha^{1/2}$ |
| | Acceleration | $LT^{-2}$ | 1 |
| | Angular velocity | $T^{-1}$ | $\alpha^{1/2}$ |
| Dynamic | Binary | $ML^2T^{-2}$ | $\alpha^4$ |
| | Power | $ML^2T^{-3}$ | $\alpha^{7/2}$ |
| | Force | $MLT^{-2}$ | $\alpha^3$ |

Source: Garcia, 2014.

## 2.4. Towing tank depth

Usually, the depth of the tank is determined as half of the tank's breadth, but to be feasible some experiments with submerged bodies, the design considers the addition of 0.5m in the tank depth, as shown in equation (6).

$$H_{tanque} = \frac{B_{tanque}}{2} + 0{,}5 \tag{6}$$

According to the ITTC (1999), the minimum acceptable depth must comply with both cases of equation (7).

$$H_{tanque} > 3.\sqrt{B_m.H_m} \text{ e } H_{tanque} > \frac{2{,}75.V_m^2}{g} \tag{7}$$

## 2.5. Frequency of encounter and wavelength

According to Fujarra (2009), a reduced scale model being towed towards waves generated in a towing tank will have a frequency of encounter with relation to these waves, following the relationship:

$$\omega_{encontro} = \omega_{onda,\ modelo} + \frac{(\omega_{onda,\ modelo})^2 . V_m}{g} \tag{8}$$

Where "$\omega_{encontro}$" is the frequency of encounter of the scale model with waves generated in the tank, "$\omega_{onda,\ modelo}$" is the frequency of wave generation in the tank and "$V_m$" is the velocity of model towing.

The wave frequency of the model can be calculated from equation (9).

$$\omega_{onda,\ modelo} = \frac{2\pi}{T_{onda,\ modelo}} \tag{9}$$

Where "$T_{onda,\ modelo}$" is the period of the wave generated in the tank and depends on the real scale period and the scale factor according to Table 1, as shown in equation (10).

$$T_{onda,\ modelo} = \frac{T_{onda,\ real}}{\sqrt{\alpha}} \tag{10}$$

Where "$T_{onda,\ real}$" is the period of sea waves at the region in question.

Similarly, the towing velocity of the model is determined by:

$$V_m = \frac{V_s}{\sqrt{\alpha}} \tag{11}$$

From the wave generation frequency, the wavelength can be found through equation (12) as cited by Muñoz (2011, p. 84).

$$L_{onda,\ modelo} = \frac{2\pi}{(\omega_{onda,\ modelo})^2} \tag{12}$$

Being "$L_{onda,\ modelo}$" the wavelength generated in the tank.

## 2.6. Blockage effects

Tank dimensions must be great enough to avoid the effects due to restricted waters. One of the pieces of information that must be included in test documentation is the tank's dimensions. (ITTC, 2002).

The correction due to deep water relates the corrections due to vertical blockage and lateral blockage. The first is a consequence of low tank depth, and the second due to insufficient tank width.

The ITTC (2011) supplies the following equations given by Tamura: equation (13) is due to the vertical blockage effect, and equation (14) due to lateral blockage. To avoid the effect of deep water, the result of both equations must be less than 0.02.

$$\frac{\Delta V_v}{V} = 0{,}67 . \left(\frac{A_{x_m}}{A_{x_{tanque}}}\right) . \left(\frac{L_m}{B_{tanque}}\right)^{0{,}75} . \frac{1}{(1 - F_{rh}^2)} \tag{13}$$

Where "$A_{x_m}$" is the maximum cross-sectional area of the model and "$A_{x_{tanque}}$" is the maximum cross-sectional area of the tank.

Formulation for the cross-sectional areas of the tank and the model are given as follows:

$$A_{x_m} = B_m . H_m \tag{14}$$

$$A_{x_{tanque}} = B_{tanque} . H_{tanque} \tag{15}$$

Ueno and Nagamatsu (1971) affirm that the shallow water effect can be ignored when the tank width and depth are in accordance with Eq. (16) and Eq. (17).

$$B_{tanque} > \frac{3}{2} L_m \tag{16}$$

$$H_{tanque} > \frac{3}{4} L_m \tag{17}$$

### 2.8. Critical velocity for avoid reflection of wall

In order to avoid wall interference on experimental results, model velocity must be greater than the critical velocity (ITTC, 2011). Equation (18) shows this case.

$$V_m > V_{crit} = \left(\frac{g}{2.\omega_{onda,modelo}}\right) . \sqrt{1 + 2\frac{L_m}{B_{tanque}} - 1} \tag{18}$$

## 3. TOWING TANK PARAMETER CALCULATION

In this study, the dimensioning of the towing tank to be built at UFSC will be determined. For this tank, several ranges of operation were required, for velocity, length, wave periods and size of the models that can be tested. These requirements are shown in Table 2.

Table 2. Design requirements for UFSC towing tank.

| Parameter | Scale | Range |
|---|---|---|
| $V_s$ | Real | 2 – 16 knots |
| $L_s$ | Real | 30 – 100 m |
| $T_{onda, real}$ | Real | 2 – 12 s |
| $L_m$ | Reduced | 0.5 – 4 m |

The wave period for the model must be between 0.5 and 2.5 seconds. This requirement is due to limitations on the wave generator and must be respected.

### 2.1. Beam and draft of reduced scale model

Weiss et al. (2014) published studies for the design of an offshore platform to support oil exploration (PSV). In these studies, they found dimensionless relations of PSVs through analysis of similar ships. The graphs that relate the L/B and L/D ratios of similar ships were used for this chapter, being “D” the depth. However, the objective of this work is to find the draft of the model. Figure 1 shows the referred graphs.

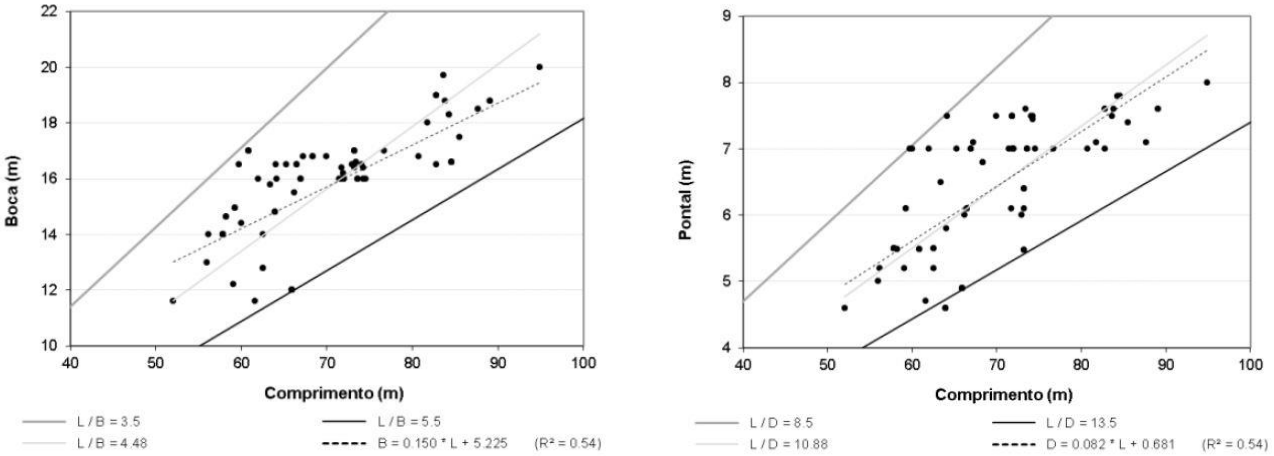


Figure 1. Geometric relations (L/B and D/L) for PSVs (Weiss et al., 2014).

With that, the beam and draft of the model can be found with equations (4) and (5).

$$B_m = \frac{L_m}{3{,}5} \tag{4}$$

$$H_m = \frac{L_m}{8{,}5} \tag{5}$$

Being "$B_m$" the beam of the model and "$H_m$" its draft.

With the equations shown in section 2, tank dimensioning was done using ModeFrontier software. No equation for the tank beam was inputted, having its values generated randomly. The objective of this section is to find the minimum beam size for the tank that still suits all project requirements.

The parameters used as input were the ones listed in Tab. 2 and also the gravitational acceleration. Several combinations were tested for the ranges of each parameter, utilizing the extreme values or in this case the most critical situations. The output data is based on parameters explained in section 2.

The results of the final dimensioning of the tank's structure are shown in Tab. 3.

Table 3. Towing tank dimensioning results.

| **Parameter** | **Description** | **Result** |
|---|---|---|
| $B_{tanque}$ | Tank beam | 6.33 m |
| $H_{tanque}$ | Tank depth[1] | 3.66 m |
| $A_{onda,\ modelo}$ | Tank wave amplitude | 0.642 m |
| $V_m$ | Model velocity | 3 m/s |

## 3.1. Towing tank length

In this study, the tank length was divided in zones of acceleration, constant velocity and deceleration.

In order to obtain precise results in a resistance test, the ITTC recommends that the model meet thirty waves during the resistance measuring zone, according to Eq. (19).

$$L_{constante} = 30.T_{encontro}.V_m \tag{19}$$

Where "$L_{constante}$" is the length of the constant velocity zone of the tank and "$T_{encontro}$" is the wave meeting period. The meeting period is determined by Eq. (20).

$$T_{encontro} = \frac{2\pi}{\omega_{encontro}} \tag{20}$$

Comparing these parameters according to conditions stated in Table 2, a meeting frequency that met the tank's requirements was determined analytically. The length was determined in order to agree with the tank's objectives. With this, the constant velocity zone length was found to be of 100 meters.

For determining the acceleration and deceleration zone lengths, the procedure was based on known towing tanks. In Tab. 4 some accelerations of the dynamometric car of ITTC tanks are shown. This data will be used as reference for the calculation of acceleration and deceleration zones for the UFSC tank.

Table 4. ITTC Dynamometric car characteristics.

| **Organization** | **Country** | **Maximum velocity [m/s]** | **Acceleration [m/s²]** |
|---|---|---|---|
| *MARINTEK* | Norway | 8 | 1 |
| *Indian Institute of Technology Madras Department of Naval Architecture* | India | 5 | 0.3 |
| *University of Ghent Department of Applied Mechanics Naval Architecture* | Belgium | 2 | 0.4 |
| *Brodarski Institute* | Croatia | 14 | 1 |
| *Canal de Experiencias Hidrodinâmicas De El Pardo* | Spain | 10 | 1 |
| *Australian Maritime College* | Australia | 4.6 | - |
| *Osaka University* | Japan | 3.5 | - |

Source: Adapted from Muñoz (2011).

Through tests conducted for the dimensioning of the towing tank, a model velocity value of approximately 3 m/s was found (Table 2), being inside the range of velocities used by towing tanks of the ITTC. The acceleration of the

[1] This does not consider the freeboard of the tank, which will be calculated in section 3.2.

dynamometric car was based on accelerations of the towing tanks in Tab. 4. In that sense, an approximate value of 0.4 $m/s^2$ was used.

The length of the acceleration zone was determined through Torricelli's relation:

$$L_{aceleração} = \frac{V_m^2}{2a} \tag{21}$$

Where "a" is the acceleration of the dynamometric car.

The length of the acceleration zone, for a 3 m/s velocity and 0.4 $m/s^2$ acceleration, is of 22.5 m. The deceleration zone was considered to be of the same length as the acceleration zone.

According to what was commented at the beginning of this section, the total tank length can be found by adding the length of each zone as follows:

$$L_{tanque} = L_{aceleração} + L_{constante} + L_{desaceleração} \tag{22}$$

Substituting the values, the total tank length results in 150 meters.

### 3.2. Freeboard

The free board in the tank is the vertical distance between the waterline and the upper extreme of the tank. To calculate this parameter, the ITTC considers a wave decline measured experimentally in towing tank tests, and modeled by Eq. (23).

$$\text{Borda livre} = 0{,}15.L_{onda,\ modelo} = 0{,}15.\left(\frac{2\pi g}{\omega^2_{onda,\ modelo}}\right) \tag{23}$$

According to what is shown in Tab. 2, one of the requirements of the towing tank project is that the period of waves produced in the tank shall not exceed 2.5 seconds, and this was the extreme value used for the calculation of the freeboard. For a period of 2.5 seconds, the wave frequency is of 2.51 rad/s (Eq. 9). With this, the value of the freeboard was found to be of 0.99 m, or approximately 1 meter.

## 4. RESULTS OF THE DIMENSIONING OF THE TOWING TANK

In Table 5, all the results for the dimensioning of the University's towing tank are shown.

Table 5. Dimensions of the towing tank.

| **Beam** | **Draft** | **Length** | **Freeboard** |
|---|---|---|---|
| 6.33 m | 3.66 m | 150 m | 1 m |

## 4. MODAL ANALYSIS

In this section is presented the modal analyses using computational simulation (Element Finite Method) with *Ansys Workbench 14.5*. The objective is to determine the influence of structural design parameters in the vibration natural frequencies of the structure.

It is known that the large structures (as airplanes and ships) using only thin steel shells require plates with great thickness, which do the design impracticable by economic or manufacture's problems. So the solution is using orthogonal stiffeners. Unlike ships, in the towing tank the stiffeners are outside the plate, to not disturb the fluid flow.

The computational model was done considering the steel structure of the towing tank with transverse stiffeners, as seen in Figure1. It was applied the internal hydrostatic pressure and it was considered two cases: with side shell free in the top and with the top side fixed in longitudinal stiffeners.

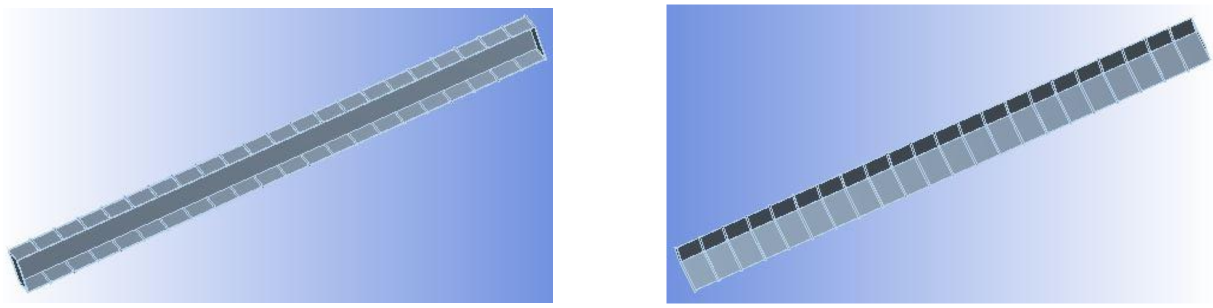

Figure 1. Towing tank with transverse stiffeners.

In Figure 2 are shown the first eight vibration modes of the structure modeled. In this case the analyze was done with solid element with 0.035 m of length, and it was considered a preliminary tank design with 22 m of length, 2 m of breadth, 1 m of draught, 30 mm of shell thickness and transverse stiffeners each 1 m.

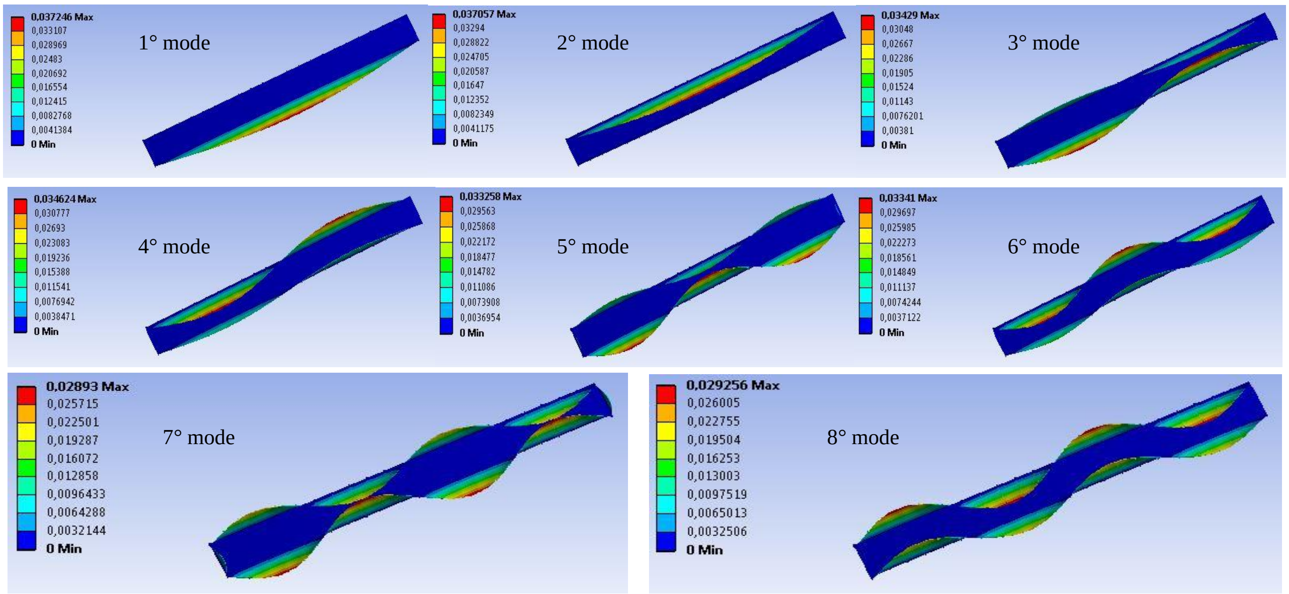


Figure 2. First eight vibration modes of a towing tank with transverse stiffeners.

## 4.1. Parametric analysis of the tank design

In this section are discussed the influence of the parameters of the tank's design in the natural frequency of the structure's vibration. The parametric analysis showed that the main parameters are the shell thickness and tank depth. The Figure 3 show that the frequencies increased with the increase of shell thickness, and decreases with the increase of the tank depth. The analysis was done for the first eight natural frequencies.

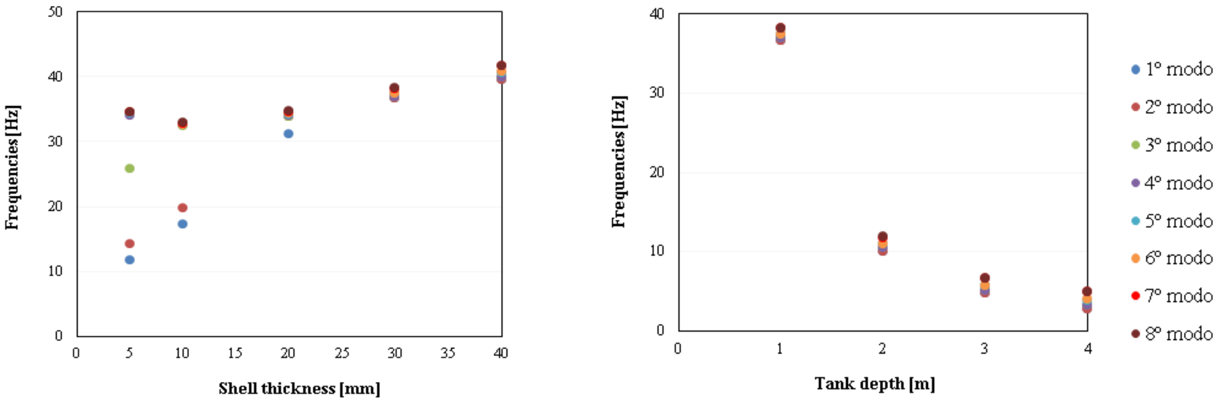


Figure 3. The influence of the parameters of the tank's design in the natural frequency of the structure's vibration.

In the Figures 4 and 5 are shown the comparisons between the structure with and without transverse stiffeners. It is important to observe that, with the increase of bottom shell thickness, closer of fixed is the structure condition and, therefore, higher are the natural frequencies. The influence of the side shell thickness is the same seen in Figure 3.

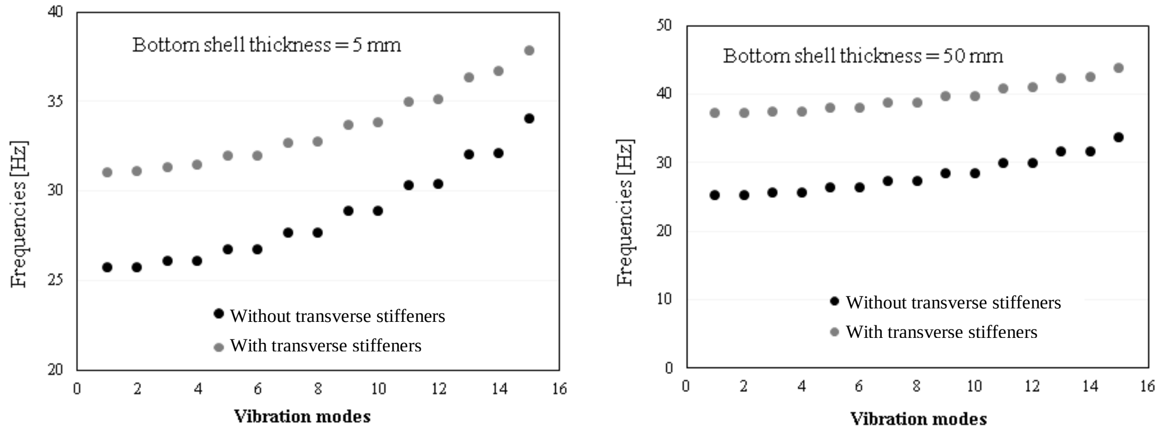


Figure 4. The influence of the bottom shell thickness in the natural frequency of the structure's vibration.

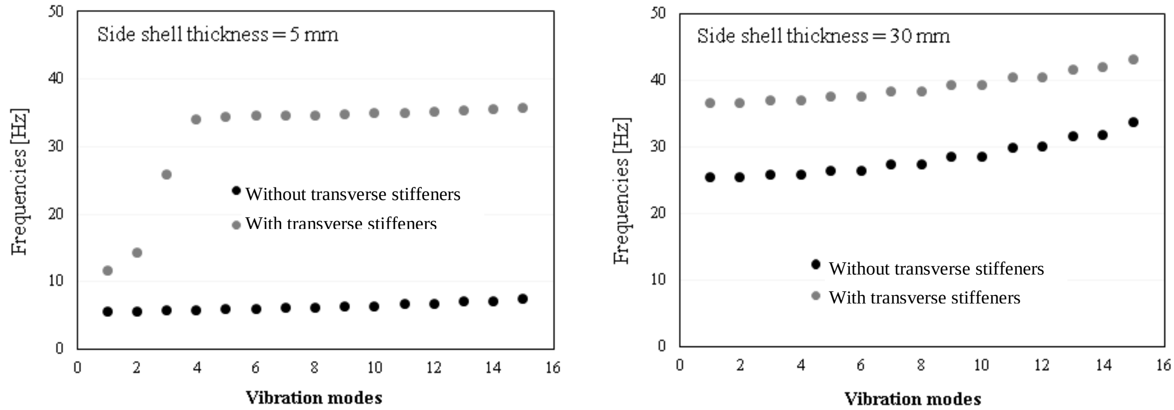


Figure 5. The influence of the side shell thickness in the natural frequency of the structure's vibration.

### 4.2. Parametric analysis of the transverse stiffeners

In this section are discussed the influence of the parameters of the transverse stiffeners in the natural frequency of the structure's vibration. The parametric analysis showed that the main parameters are the transverse web height and the distance between transverse stiffeners. The Figure 6 show that the frequencies increased with the increase of the transverse web height, and decreases with the increase of the distance between transverse stiffeners. The analysis was done for the first eight natural frequencies.

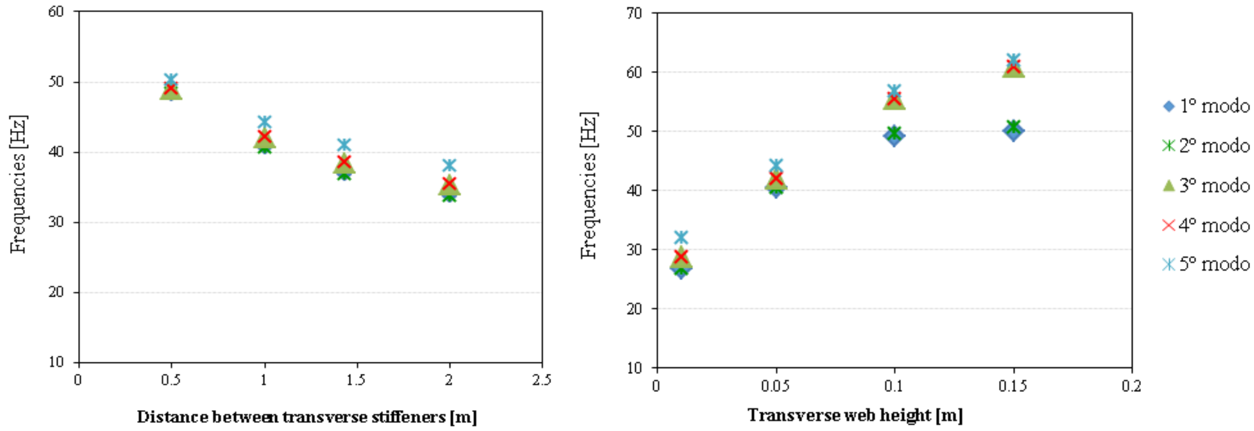


Figure 6. The influence of the transverse web height and the distance between transverse stiffeners.

As expected, the natural frequencies values increased with the increase of the transverse web height and with the decrease of the distance between transverse stiffeners, due to the increased system rigidity and the low weight variation of the structure. In Figures 7 and 8 are shown the local vibration modes, which are the most important in the design.

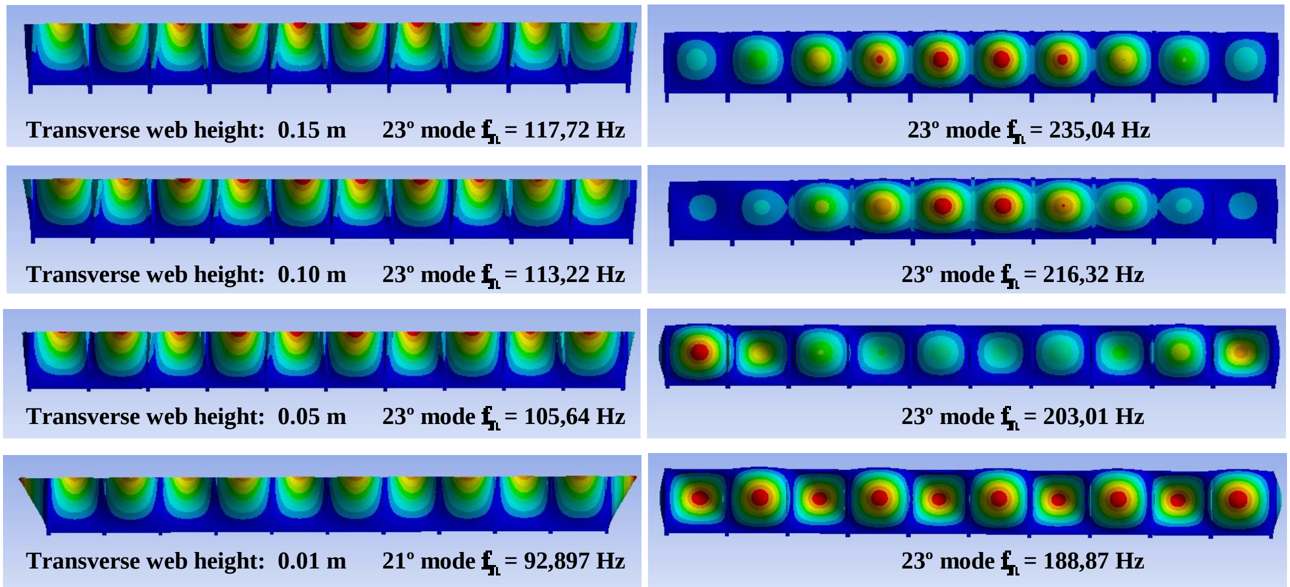


Figure 7. Lateral views of local vibration modes. With (right) and without (left) longitudinal stiffener at top.

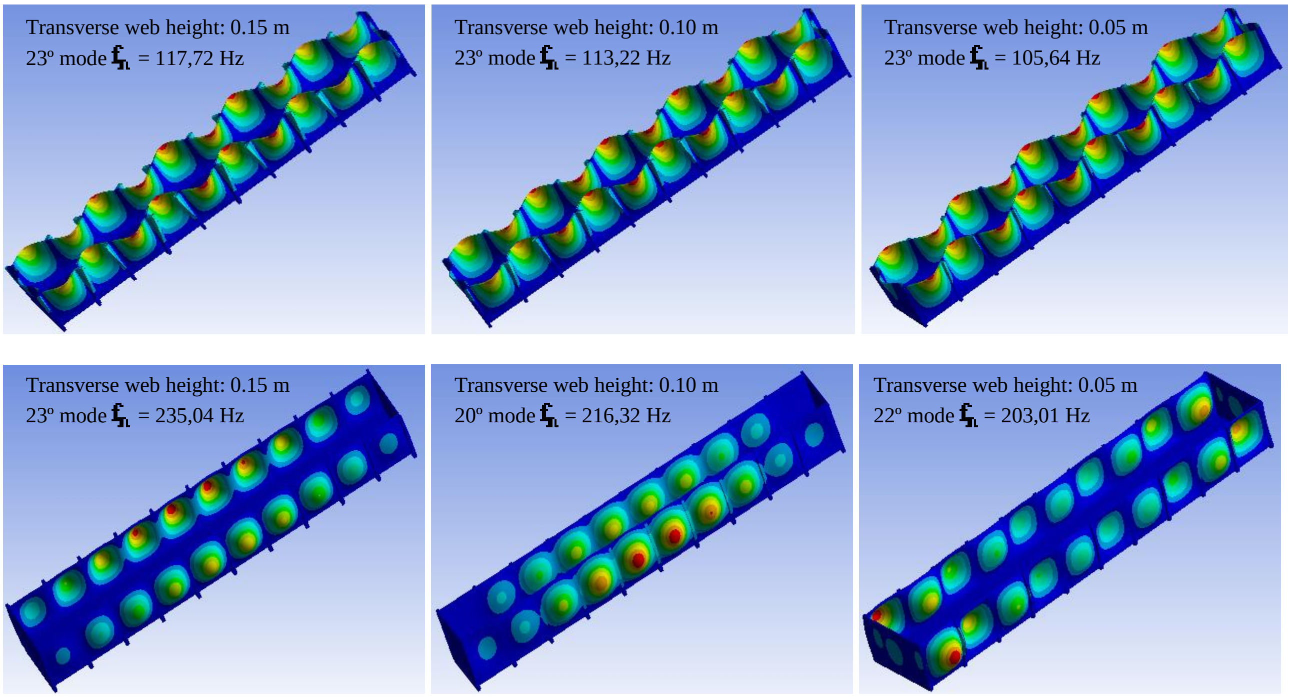


Figure 8. Isometric views of local vibration modes. With (bottom) and without (top) longitudinal stiffener at top.

## 6. CONCLUSIONS

This work has presented a study of the dimensioning of a towing tank made of steel to be built at the Joinville Campus of Universidade Federal de Santa Catarina.

With the tank being a structure made of steel and subject to excitations due to wave generation and model resistances, a modal analysis is necessary, in order to minimize the effect vibrations may inflict on the results and also on structure fatigue.

For the determination of the towing tank's dimensions, the objective was to determine the minimum tank size for conducting the intended tests, as large tanks represent higher operational and maintenance costs. The analysis resulted in the smallest possible tank to fulfill the planned test matrix to have dimensions of a draft of 3.66 m, freeboard of 1.0 m, width of 6.33 m and a length of 150 m.

The structure design was done to keep the von Mises stress below of 180 MPa and also the natural frequencies far away of the wave frequencies that will be used in the planned experiments.

## 7. REFERENCES

FUJARRA, André Luís Condino. **Dinâmica de sistemas II:** Material de apoio. 124 p. Departamento de Engenharia Naval e Oceânica, Escola Politécnica da Universidade de São Paulo. São Paulo, 2009.

GARCIA, Edgard Enrique Mulford. **Procedimento para determinação experimental de carregamentos externos para o projeto estrutural de uma plataforma semi-submersível.** 195 p. Dissertação (Mestrado) Programa de Pós-Graduação em Engenharia Naval e Oceânica, Escola Politécnica da Universidade de São Paulo. São Paulo, 2014.

GONÇALVES, Rogério Sales; CARVALHO, João Carlos Mendes. **Estudo da rigidez de sistemas multicorpos.** In: 17º SIMPÓSIO DE PROGRAMA DE PÓS-GRADUAÇÃO EM ENGENHARIA MECÂNICA. Faculdade de Engenharia Mecânica, Universidade Federal de Uberlândia. Uberlândia, MG.

INMAN, Daniel J. **Engineering Vibration.** 2nd ed. New Jersey, NJ: Pearson Education International, 2001.

ITTC – 22nd International Towing Tank Conference, Recommended Procedure and Guidelines, **The Specialist Committee on Trials and Monitoring**, Seul e Xangai, 1999.

ITTC – 26th International Towing Tank Conference, Recommended Procedure and Guidelines, **Resistance Test**, Venice, 2011.

LEISSA, Arthur W. **Vibration of plates.** Ohio State University. Washington, DC: National Aeronautics and Space Administration, 1969.

LOGAN, Daryl L. **A first course in the finite element method.** 5th ed. United States of America. USA: Cengage Learning, 2012.

MUÑOZ, Jaime Miguel Mariano Saldarriaga. **Estudo da metodologia para o dimensionamento de um tanque de provas do tipo reboque.** 2011. 156 p. Dissertação (Mestrado) – Programa de Pós-Graduação em Engenharia Naval e Oceânica, Escola Politécnica, Universidade de São Paulo. São Paulo, 2011.

NABARRETE, Airton. **Vibrações em sistemas mecânicos.** 4. ed. Notas de aula. Centro Universitário da FEI. 2005.

RAO, Singireseu. **Vibrações mecânicas.** Tradução de Arlete Simille Marques. 4. ed. São Paulo, SP: Pearson Prentice Hall, 2008.

TIMOSHENKO, S.; WOINOWSKY-KRIEGER, S. **Theory of plates and shells.** 2nd ed. New York: McGraw-Hill, 1959.

UENO, Keizo; NAGAMATSU, Tetsuro. **Effect of restricted water on wave-making resistance.** 18 p. In: THE JAPAN SOCIETY OF NAVAL ARCHITECTS AND OCEAN ENGINEERS.

WEISS, James M. G. et al. **Projeto de Platform Supply Boats baseado em otimização multiobjetivo.** In: 25º CONGRESSO NACIONAL DE TRANSPORTE AQUAVIÁRIO, CONSTRUÇÃO NAVAL E OFFSHORE, 2014, Rio de Janeiro, RJ.

ZULETA, Wilson Toncel; CABRERA T., Jairo. **Design model of a hydrodynamic towing tank for Colombia.** Ship Science & Technology, v. 7, n. 14, 2014. Cartagena, Colombia.