SCIREA Journal of Mechanical Engineering

http://www.scirea.org/journal/Mechanical

August 30, 2023

Volume 4, Issue 1, February 2023

http://dx.doi.org/10.54647/mechanical46094

# Research on full balance friction hoist shiplifts

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## Abstract

This paper proposes the conception of full balance friction hoist shiplifts to supply the solution of the problems in layout of the main hoists, locking of the ship chamber under the thorough water leakage accident, and insufficiency of ship chamber pitch stability, existed in traditional full balance winch shiplifts for large tonnage design ships and high lift. The test study for measuring the friction coefficients under lubrication has been conducted and the pivotal issues such as compact layout of main hoists by employing friction hoists, fortification of the thorough water leakage accident, and pitch stability of ship chambers have been elucidated. The scheme comparison of 200m/3000t between full balance winch shiplifts and full balance friction hoist shiplifts has been conducted which shows that full balance friction hoist shiplifts supplies the solution for fortification of the thorough water leakage accident and enhancement of the pitch stability of the ship chambers of shiplifts for large tonnage ships and high lift.

**Keywords:** Shiplifts, friction hoist, compact layout, fortification, thorough water leakage accident, pitch stability.

## 1. Introduction

Vertical shiplifts are navigation facilities with the same function as ship locks. It is widely used in Europe because of its water-saving characteristics as a means to solve the navigation problems caused by water surface elevation difference between two natural rivers connected by an artificial canal [1]. As examples, Germany's Niederfinow 1000t shiplift functions in Oder-Havel Canal [2], Luneburg 1350t shiplift in branch canal of Elbe River, Germany's new Henrichenburg 1350t shiplift in Dortmund-Ems-Kanal and Belgium's Strepy 1350t winch vertical shiplift in branch of the Canal du Centre in Hainaut, etc [3].

The development of vertical shiplifts in China have always been based on the construction of hydro-projects to meet the demands for navigation [4,5], so the influence of change in water level of the upper bays and lower bays should be considered in design. Since 90s in last century, totally 17 shiplifts have been built in the hydro-projects in the hydro-projects of China. Most of them are vertical shiplifts with counterweights, which include full balance rack and pinion shiplifts [6-9], partial balance winch shiplifts[10,11], and the full balance winch shiplifts (FBWSs) [12-15]. For partial balance winch shiplifts, there are great difficulties in layout of main hoist and fabrication of the gearboxes as both the loads and component sizes are very large. Therefore, for applications in high water head dam and large tonnage ships, full balance shiplifts are reasonable and realistic choices. The advantages of full balance rack and pinion shiplifts have nice performance in safety including fortification of the thorough water leakage accident (TWLA) and ship chamber pitch stability [16]. The disadvantage of this type are high construction cost and huge amount of maintenance as the systems are very complicated. The FBWSs are broadly applied in China due to the advantages of the simplicity in systems, low construction cost and convenient maintenance. But the safety issues for large scale FBWSs relating to the fortification of TWLA and the pitch stability of ship chambers exist [17]. It is the consideration of these issues that resulted in the change of the scheme of the Three Gorges shiplift into full balance rack and pinion shiplift from the original scheme of the FBWS [18]. For shiplifts for large ships and of high lifts, it is hard to realize fortification of the TWLA and own the strong pitch stability of the ship chambers, as the longitudinal layout space is limited. This paper proposes the design concept of the FBFHSs which retains the basic features and advantages of FBWSs but with more excellent performance in safety. Special attention has been paid in compact layout of main hoist by employing friction hoists, the fortification of the TWLA and the pitch stability of the ship chambers.

## 2. Methods: test and design

### 2.1 Basic conceptions of FBFHSs

The structure of FBFHSs is similar to that of the FBWSs. The key technique for design of FBFHSs is compact layout of the main hoists by employing the friction hoists to replace the traditional winches in FBWSs, and setting of the accidental brakes by employing the the balance friction drums to replace the traditional pulley assembles. The compact layout of the main hoists spares the longitudinal space, so the length of the ship chambers can be minimized and the balance friction drums can be arranged. Thus, the compact layout brings three benefits: avoiding unnecessary extension of the length of ship chambers, increase of the total break force of the shiplifts so that fortification of he TWLA can be realized, and enhancement of the pitch stability of the ship chambers as the longitudinal central distances increase.

The main hoists are installed on the floor of the top machine rooms, and are mainly composed of friction hoists, balance friction drums, accident brakes and synchronous shaft system. Each friction hoist drives two hoisting friction drums by an electric motor through a gearbox. The hoisting wire ropes are winded on each hoisting friction drum. Each wire rope is winded on a spiral rope groove of the drum for several turns, and the number of turns is determined according to the non-sliding condition under the TWLA of the ship chamber. Figure 1 shows the layout of a typical main hoist of a FBFHS.

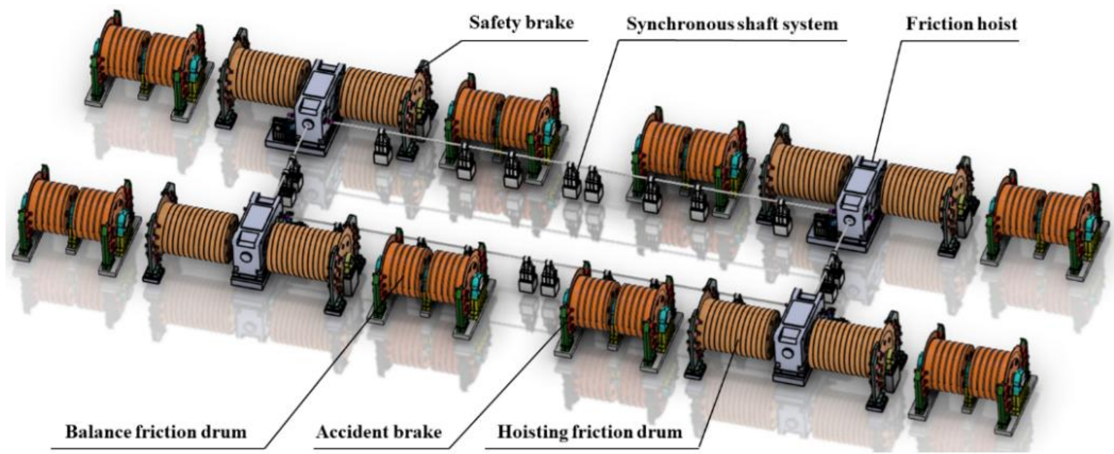


**Fig 1:** Graphic of the main hoist of a FBFHS

The hoisting wire ropes are not fixed on the hoisting friction drum. The torque caused by differences of the tensions by the two sides of the drum is transferred by the friction force between wire ropes and grooves of the drum. A piece of the counterweight of cast steel is freely suspended at one end of each hoisting wire rope, so the tension of the rope section on the side of the counterweight keeps unchanged. The other end of the rope is connected to the lifting eye of the ship chamber structure. The tensions of the wire rope sections on the side of the ship chamber are determined according to the conditions of force balance and deformation coordination of ship chamber structure. The tensions applied to the two ends of a wire rope satisfy the following non-sliding condition:

$$S_1 \geq S_2 e^{-2j\pi\mu} \tag{1}$$

Where $S_1$ and $S_2$ denote respectively the tensions in the sections of same wire ropes on the side of ship chamber and counterweighs, $\mu$ the friction coefficient between the wire ropes and grooves of the friction drums, $j$ the turn number of single wire rope winding on the drum. For TWLA of ship chambers studied in this paper. For k-th wire rope in a friction drum, Formula (1) can also be written in the following form

$$\Delta S_k = S_{k2} - S_{k1} \leq S_{k2}(1 - e^{-2j\pi\mu}) \tag{2}$$

The external torque applied on a hoisting friction drum is as follows

$$M_d = \sum_{k=1}^{n} \Delta S_k r = \sum_{k=1}^{n} (S_{k2} - S_{k1}) R_d \tag{3}$$

Where $n$ is the number of the hoisting wire ropes winding on a hoisting friction drum, and $R_d$ is the nominal radium of the drums.

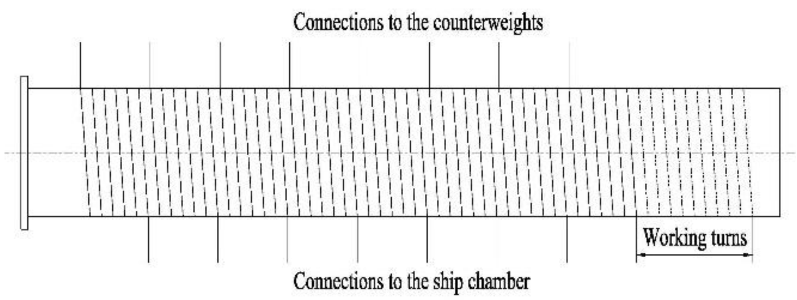


**Fig 2:** Scheme for arrangement of wire ropes winding on the friction drum

The hoisting wire ropes are winded along a spiral groove on the drum as shown in Figure 2. Two drums are assembled asymmetrically in the two sides of a gearbox so that the axial forces from the drums on the gearbox are offset. The length of a drum can be calculated as followings

$$L_{gd} = ((j + 0.5)n + n_h - 0.5)t + L_b + L_g + t_b \tag{4}$$

Where $t$ denotes the rope groove pitch, $L_b$ is the shortest distance between the inner face of brake disc and centre line of the nearest rope groove, $L_g$ is the shortest distance between the drum end without the disc and centre line of the rope groove, $t_b$ is the thickness of brake disc, $n_h$ is the winding turn number for single hoisting wire rope winding on the hoisting friction drum for hoisting the maximum lift $H_{max}$

$$n_h = \frac{H_{\max}}{2\pi R_d} \tag{5}$$

Comparing to ordinary winches in the FBWS, more wire ropes can be placed on each friction hoisting drum and the number of the friction hoists of the main hoist of a FBFHS can be halved for employing same number of hoisting wire ropes. So the longitudinal layout space occupied by the friction hoists of the main hoist of a FBFHS is much smaller than that occupied by the winches of the main hoist of a FBWS. As the longitudinal space is spared, the balance friction drums can be employed to replace the pulley assembles used in the main hoists of the FBWSs. The accidental brakes can be placed on the balance friction drums. In this way, the brake force for fortification of the TWLA of ship chambers is greatly increased.

### 2.2 Test study on the friction coefficient between friction drum and wire ropes

The friction coefficient between grooves of the friction drums of the main hoist and wire ropes is a pivotal factor that affects the safety of FBFHSs. The friction coefficient 0.1 between grooves of drums and wire ropes is recommended considering the existence of grease on the contact surfaces in the calculation of bolt tensions for clamp of tails of wire ropes in cranes [19]. Lijun Liu and Suixin Zhang et al studied the several factors that affect the friction coefficients under lubrication [20]. The research shows that the friction coefficient of wire ropes to the piece of metallic material is between 0.055~0.135 under lubrication, and increases as sliding velocity increases.


**Fig. 3: Pad simulation tester**

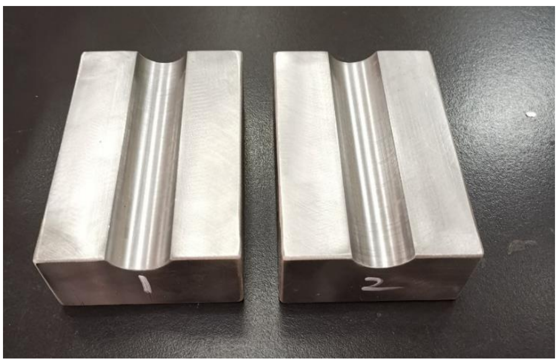

**Fig. 4: Sample block picture**

A test study on the friction coefficient between the steel wire ropes and grooves of steel drums under lubrication has been conducted on the SC-3 pad simulation tester in this program. The SC-3 pad simulation tester is shown in Figure 3. The wire rope sample is a section of galvanized wire rope 6V (21) –1 with diameter of 26mm. The groove sample block is of steel 35 with groove surface roughness 3.2μm as shown in Figure 4. The test surface conditions include the clean surface, the surface with water, the surface with IRIS antirust grease and the surface with No.3 lithium grease. The minimum, maximum and average values of the friction coefficients between the wire rope and the groove under various test conditions are listed in Table 1. It can be seen that the minimum friction coefficient occurs under the surface with IRIS antirust grease condition which is 0.142.

The rope-groove friction coefficient takes the value of 0.05 on the safe side in our work.

**Table 1:** Experimental results of the rope-groove friction coefficient

| Condition | COF Max | COF Min | COF Avg |
|---|---|---|---|
| Clean surface | 0.188 | 0.175 | 0.177 |
| Surface with water | 0.192 | 0.187 | 0.190 |
| Surface with IRIS antirust grease | 0.149 | 0.142 | 0.144 |
| Surface with No. 3 lithium grease | 0.151 | 0.145 | 0.148 |

### 2.3 Fortification of the TWLA

An important feature of the full balance vertical shiplift is that the total gravity force of the ship chamber under the design water depth is equal to the gravity force of the counterweights. As an accident, the leakage of the water of the ship chamber causes unbalance force when the shiplift is in hoisting process. So the fortification of TWLA should be realized. In order to accomplish this object, the essential condition for FBFHSs is that the sum of the critical sliding friction forces of all hoisting wire ropes and balance wire ropes, together with the braking force of the lock mechanisms of the ship chamber, is larger than the gravity force of the water in the ship chamber. If the winding turn numbers for all wire ropes on all the drums are taken same value, the above condition can be formulated as

$$S_{FE} = \frac{W(1-e^{-2j\mu\pi}) + F_L}{W_w} \geq 1.05 \tag{6}$$

Where $S_{FE}$ is the safety factor for fortification of TWLA, $W$ and $W_w$ are respectively the total gravity forces of the ship chamber and the water in the ship chamber, $F_L$ the total braking force of the locking mechanisms of the ship chamber.

The design principal for the safety brakes and the accident brakes is that the safety brakes and the accident brakes of the main hoist are able to brake respectively the maximum torques on the hoisting friction drums and balance friction balance drums under the TWLA. Generally, the main hoist is equipped with 8 hoisting friction drums. The critical sliding tension under which sliding of the hoisting wire ropes occurs is

$$T_{hc} = \frac{W_t}{8n_{hr}} e^{-2j\mu\pi} \tag{7}$$

Where $n_{hr}$ is the number of the hoisting wire ropes winding on a hoisting friction drum. The limit torque applied on a hoisting friction drum is

$$M_{hd} = \frac{W_t R_d}{8}(1 - e^{-2j\mu\pi}) \tag{8}$$

The design of the safety brakes should meet the stipulation [21]

$$S_{sb} = \frac{n_{sb} R_b F_b}{M_{hd}} > S_{b\min} = 1.5 \tag{9}$$

Where $S_{sb}$ is the safety factor for safety brakes, $n_{sb}$ is the number of the hydraulic disc brakes on each hoisting friction drum, $R_b$ is the radius of the circle along which the disc brakes are distributed, $F_b$ is the braking force of the hydraulic disc brake, $S_{bmin}$ is the minimum braking safety factor for safety brakes.

For large scale FBFHSs, there are generally 16 balance friction drums in a main hoist. The critical sliding tension under which sliding of the balance wire ropes occurs is

$$T_{bc} = \frac{W_g}{16n_{br}} e^{-2j\mu\pi} \tag{10}$$

Where $n_{br}$ is the number of the balance wire ropes on a hoisting friction drum. The limit torque applied on a balance friction drum is

$$M_{bd} = \frac{W_g r}{16}(1 - e^{-2j\mu\pi}) \tag{11}$$

The design of the accident brakes should meet the stipulation

$$S_{ab} = \frac{n_{ab} R F_b}{M_{bd}} > S_{b\min} = 1.5 \tag{12}$$

Where $S_{ab}$ is the safety factor of accident brakes, $n_{ab}$ the number of the hydraulic disc brakes on each balance friction drum.

For FBFHSs, all hoisting wire ropes and balance wire ropes are elastic bearing elements of ship chambers in the situation of TWLA, which constitutes a statically indeterminate structural system. When the essential conditions mentioned above are satisfied, a question arises how the safety of the fortification is affected by slide of a few wire ropes occurs, or in other words, how many wire ropes that slides under TWLA are allowed to keep the ship chamber safely locked. To answer this question, further research is to be conducted. To make issue simple and clear here, it is stipulated in this paper that the sliding of any ropes is not allowed in the design of FBFHSs. Considering the number of the wire ropes is colossal and the ship chamber is complicated, the FEA method is applied to compute the tensions of all wire ropes suspending the ship chamber. And then a comparison of the rope tensions obtained by FEA modelling and the critical sliding tensions calculated by (7) and (10) is conducted to judge if non-sliding condition (1) is satisfied for all suspension wire ropes.

In order to more accurately reflect the influence of TWLA process on the wire rope tensions, the finite element modelling adopts the two-step loading method as followings:

First step, to establish the FEA model for the ship chamber in the normal hoisting case of which the water pressure and gravitational force are applied. The hoisting wire ropes are treated as the elastic bearing elements of the ship chamber and the balance wire ropes are treated as forces caused by gravity of the counterweights suspended. According to this model, the information of the initial deformation is obtained.

Second step, to establish the ship chamber FEA model in the case of the TWLA, in which the ship chamber is elastically borne by the all hoisting wire ropes and balance wire ropes. The vertical downward lock forces are applied by the ship chamber locking mechanisms at the location where the mechanisms are assembled. The data about the deformation of the ship chamber structure obtained in the first step is input in the FEA model as the inertial conditions.

### 2.4 Pitch stability of ship chambers

The dynamic model for the pitch stability analysis of ship chambers is shown in Figure 5. It is assumed that the ship chamber is a rigid body, and the water in ship chamber is described as the fluid that satisfying Hausner's assumption [22]. The dynamic equations for slosh of water and the pitch of the ship chamber are described by equations (13) and (14) respectively [18, 23]:

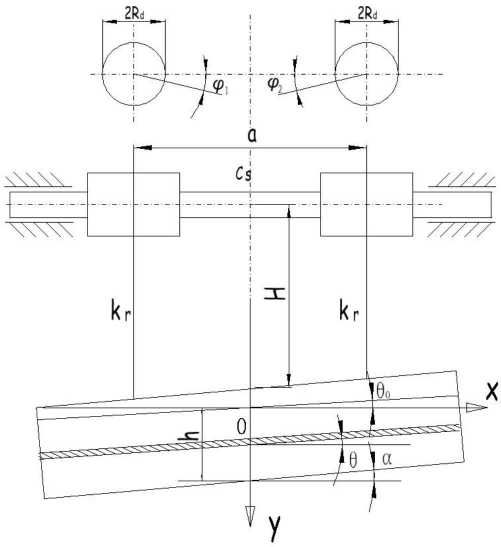


**Fig.5:** Mechanical model for the ship chamber and the main hoist

$$\ddot{\theta}_0 + \omega_w{}^2\theta_0 - \ddot{\alpha} = 0 \tag{13}$$

$$M_w = -J_w(\ddot{\theta}_0 + \ddot{\alpha}) + C_w(\alpha - \theta_0) \tag{14}$$

Where $\theta_0$ is the slosh angle of water surface around the horizontal plane, $\alpha$ is the pitch angle of the ship chamber, $J_w$, $C_w$ and $\omega_w$ are respectively parameters related to the size of ship chamber and the water density:

$$J_w = \frac{1}{24}\rho BhL^3, \quad C_w = \frac{1}{12}g\rho BL^3, \quad \omega_w = \frac{\sqrt{10gh}}{L} \tag{15}$$

Where $h$ is the design water depth of the ship chamber, $B$ and $L$ are respectively the maximum width and the length of the ship chamber, $\rho$ is the water density and $g$ is the gravitational acceleration.

According to the Lagrange's equations, the following coupling differential dynamic equations of torsional vibration of the main hoist, pitch of the ship chamber and slosh of the water can be derived:

$$\begin{aligned}
&J_h\ddot{\varphi}_1 + k_rR_d(R_d\varphi_1 - y - \frac{1}{2}a\alpha) + C_s(\varphi_1 - \varphi_2) = M_{d1} \\
&J_h\ddot{\varphi}_2 + k_rR_d(R_d\varphi_2 - y + \frac{1}{2}a\alpha) + C_s(\varphi_2 - \varphi_1) = M_{d2} \\
&J_c\ddot{\alpha} + \frac{1}{2}ak_r(\frac{1}{2}a\alpha - R_d\varphi_1 + y) + \frac{1}{2}ak_r(\frac{1}{2}a\alpha + R_d\varphi_2 - y) = M_w
\end{aligned} \tag{16}$$

Where $\varphi_1$ and $\varphi_2$ are rotation angles of the drums at upstream and downstream sides of main hoist respectively, $y$ is the vertical displacement of the water, $a$ is the longitudinal central distance of the main hoist, $J_h$ is half of the moment of inertia of the rotating parts of main hoist with respect to the low-speed shaft, $J_c$ is the moment of inertia of rigid body of the ship chamber with respect to the transverse centerline, $C_s$ is the torsional stiffness of one synchronous shaft system, $k_r$ is half of the sum of tensile stiffness of all hoisting wire ropes when the ship chamber is at the lowest position, $M_{d1}$ and $M_{d2}$ are the damping moments acting on the winch at upstream and downstream sides respectively, which are calculated according to the following formula

$$M_{di} = -d_h \varphi_i \ (i = 1, 2) \tag{17}$$

Where $d_h$ is the damping coefficient of torsional vibration of the main hoist.

Substituting (13) and (14) into (16) and let $\varphi=\varphi_1-\varphi_2$, $\phi=\theta_0-\alpha$ , yields

$$\begin{aligned}
&J_h\ddot{\varphi} + d_h\dot{\varphi} + (k_r R_d^{\ 2} + 2C_s)\varphi + k_r a R_d \alpha = 0 \\
&(J_s + 2J_w)\ddot{\alpha} + (\frac{1}{2}k_r a^2 - \omega_w^{\ 2} J_w - C_w)\alpha + \frac{1}{2}k_r a R_d \varphi + (C_w - \omega_w^{\ 2} J_w)\phi = 0 \\
&\ddot{\phi} + \omega_w^{\ 2}(\alpha + \phi) = 0
\end{aligned} \tag{18}$$

The damping is considered to be of Rayleigh's so the damping coefficient of torsional vibration of main hoist is calculated as following

$$d_h = \zeta\sqrt{(k_r r^2 + 2C_s)J_h} \tag{19}$$

Where $\zeta$ is the damping ratio of the main hoist. The corresponding characteristic equations can be derived from the dynamic differential equation (17)

$$\sum_{i=0}^{6} a_i r^i = 0 \tag{20}$$

Where

$$a_0 = (\frac{a}{r} - \frac{4C_w}{k_r a r})(k_r r^2 + 2C_s) - k_r a r \tag{21}$$

$$a_1 = (\frac{a}{r} - \frac{4C_w}{k_r a r})d_h \tag{22}$$

$$a_2 = \frac{2(J_2 + 2J_s)}{k_r a r} + (\frac{a}{r} - \frac{4C_w}{k_r a r})J_h - \frac{k_r a r}{\omega_w^{\ 2}} + \frac{(k_r a^2 - 2\omega_w^{\ 2} J_w - 2C_s)(k_r r^2 + 2C_s)}{\omega_w^{\ 2} k_r a r} \tag{23}$$

$$a_3 = \frac{2(J_s + 2J_w)d_h}{k_r a r} + \frac{(k_r a^2 - \omega_w^{\ 2} J_w - 2C_w)d_h}{\omega_w^{\ 2} k_r a r} \tag{24}$$

$$a_4 = \frac{2(J_s + 2J_w)J_h}{kar} + \frac{(k_r a^2 - 2\omega_w{}^2 J_s - 2C_w)J_h}{\omega_w{}^2 k_r ar} + \frac{2(k_r r^2 + 2C_s)(J_s + 2J_w)}{\omega_w{}^2 k_r ar} \tag{25}$$

$$a_5 = \frac{2(J_2 + 2J_s)d_h}{\omega_w{}^2 k_r ar} \tag{26}$$

$$a_6 = \frac{2(J_s + 2J_w)J_h}{\omega_w{}^2 k_r ar} \tag{27}$$

The eigenvalues can be obtained by solving the characteristic equation (20). The pitch stability can be judged by checking the signs of real parts of the eigenvalues. The dynamic system is stable if the real parts of all eigenvalues are minus. The dynamic system is unstable if the real part of any eigenvalues is positive. The critical longitudinal central distance of the main hoist with which the maximum real part among the eigenvalues is minus and adequately close to zero can be found by numeral computation. We call it the critical longitudinal central distance for pitch stability and denote it by $a_c$. The pitch stability safety factor is defined as following:

$$S_p = \frac{a}{a_c} \tag{28}$$

## 3. Type comparison for the 200m/3000t shiplift

### 3.1 The scheme of FBWS

According to the characteristic dimensions of the design ship, the ship chamber1usable dimensions 95m×18m×4.7m (length×width×design depth) are enough to satisfy the demand for holding the design ship, and the corresponding maximum dimension of the ship chamber is 115m×18.4m×4.7m. However, the longitudinal arrangement of the main hoist cannot be realized for the total length of the ship chamber of 115m, as the lift of 200m is too large which results in the huge dimensions in length and diameter of the drums of the main hoist. In order to realize the layout of the main hoist, the total length of the ship chamber has to be extended to 130m. Thus the weight of the water in chamber is 11600t, and the weight of ship chamber structure and equipment is 5600t, and the total weight of the ship chamber is 17200t. As a full balance shiplift, the total weight of counterweights is also 17200t among which the total weight of torque counterweights is 5000t and the total weight of gravity counterweights is 12200t [24].

There are 192 wire ropes of diameter 88mm suspending the ship chamber, including 128 gravity balance wire ropes, 64 hoisting wire ropes. Besides there are 64 torque balance wire ropes suspending the torque counterweights. There are 16 drums in the main hoist with nominal diameter 7.6m, on which four hoisting ropes and four torque balance ropes wind. The drum length is 6.8m. There are eight double-groove pulleys with diameter of 7.6m in the main hoist. Figure 6 shows the layout of the 200m/3000t FBWS.

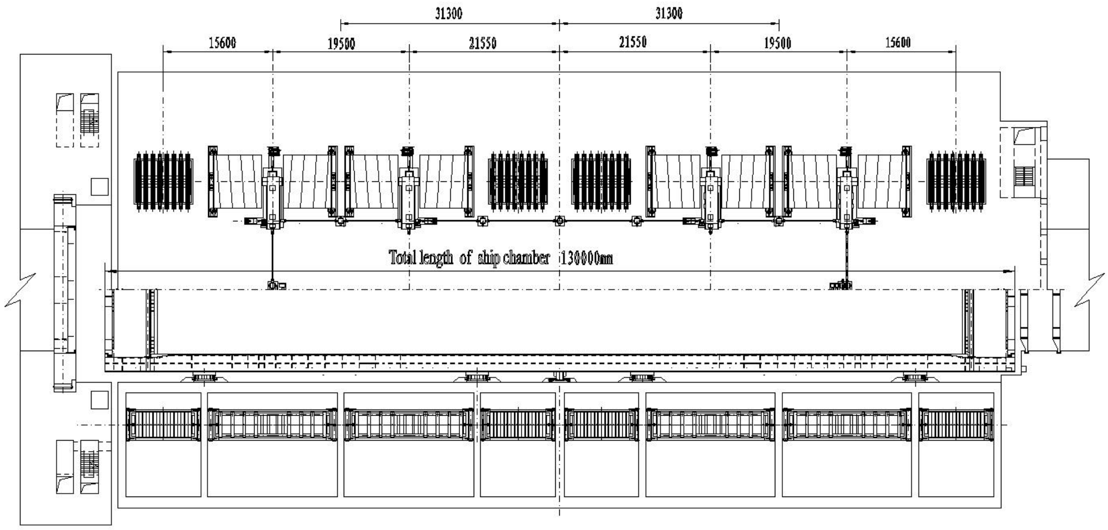


**Fig. 6:** Layout of 200m/3000t FBWS

The braking force for the TWLA is supplied only by the safety brakes and the lock mechanisms of the ship chamber. The usable brake force of the safety brakes that in the TWLA is the gravity force of the torque counterweights. The total braking force of the locking mechanisms of the ship chamber is 19600kN. By combined action of safety brakes and locking mechanism, only 68600kN brake force can be obtained, which accounts for only 60.3% of the total gravity force of the water in the ship chamber. So the fortification of the TWLA is not realized.

The safety factor of the pitch stability of the ship chamber is calculated according to formula (18)~(28) and the parameters for the 200m/3000t FBWS listed in the Table 5. The eigenvalues corresponding to the pitch motion is $-3.231\times10^{-7}\pm 1.39\times10^{-3}i$ ($i$ is imaginary unit) when the longitudinal central distance of main hoist is 44.209m. When the longitudinal central distance of the main hoist is 44.208m, one pair of the eigenvalues are $\pm6.992\times10^{-4}$ which means that the system is unstable. So it can be considered that the critical longitudinal central distance is 44.209m, and the safety factor for the pitch stability is 1.416.

### 3.2 The scheme of FBFHS

The total length of the ship chamber is 115m for the scheme of the FBFHS which is enough to meet the demand for ship holding, with the total width and the design water depth unchanged w.r.t the scheme of the FBWS. The ship chamber is suspended by 160 steel wire ropes with diameter of 97mm. The weight of the water in the ship chamber is 10300t, and the weight of ship chamber structure and equipment is 5100t. The weight of gravity counterweight is 10000t, and the weight of torque counterweight is 5400t.

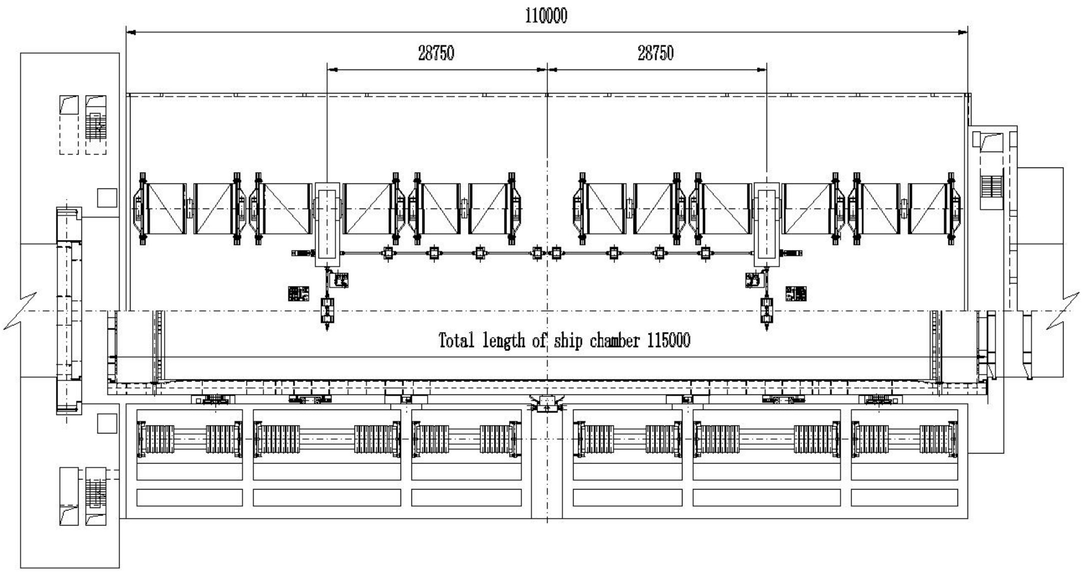


**Fig.7:** Layout of 200m/3000t FBFHS

There are 4 friction hoists, which contain 8 friction hoisting drums of 6.6m in diameter, and 16 balance friction drums of the main hoist. There are 8 hoisting wire ropes winding for 5.5 turns on each friction hoisting drum and 6 balance ropes winding for 5.5 turns on each balance friction drum. A safety brake assembly of 8 VMS-DP70 hydraulic disc brakes is set on the disc in the friction hosting drum, and an accident brake assembly of 6 VMS-DP70 hydraulic disc brakes is set on the disc in each balance friction drum. The braking safety factors of the safety brakes and accident brakes are respectively 1.50 and 1.62. The layout of the 200m/3000t FBFHS is shown in Figure 7. Substituting the values of the parameters ($W$=150920kN, $F_L$=19600 kN, $W_w$=100940 kN, $\mu$=0.05, $j$=5.5) into formula (6), we get that the safety factor for fortification of TWLA is $S_{FE}$=1.424.

According to the FEA modelling shown in Figure 8 and the two-step method discussed above, the tensions of all hoisting wire ropes and the balance wire ropes have been computed to verify the non-sliding condition of the wire ropes. The numbers that identify the hoisting wire ropes and balance wire ropes in the 1/4 ship chamber is shown in Figure 9. The case A (the ship chamber is located in the lowest position) and the case B (the ship chamber is in the highest position) are considered. The tensions of the wire ropes are listed in the Table 2~Table 4. We can see that the lowest tensions of the hoisting wire ropes and the balance wire ropes are respectively 373.4kN and 334.3kN. The minimum tensions of the hoisting wire ropes and balance on the side of the ship chamber determined by the no-sliding condition are respectively 146.9kN and 181.4kN according to formula(7) and (10). Since the computed tensions are all larger than the minimum tensions determined by the non-sliding condition, sliding of the wire ropes will not occur and the fortification of the TWLA is realized.

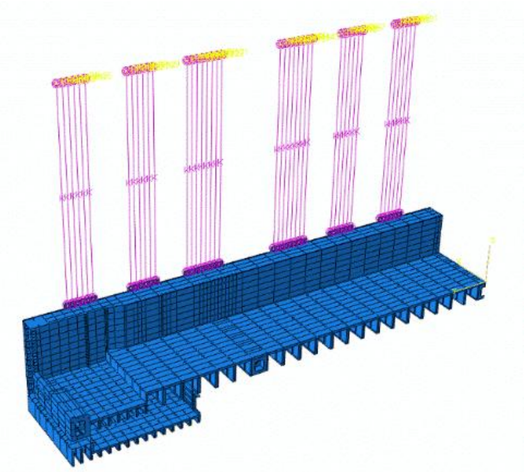

**Fig.8:** Finite element model of 1/4 ship chamber

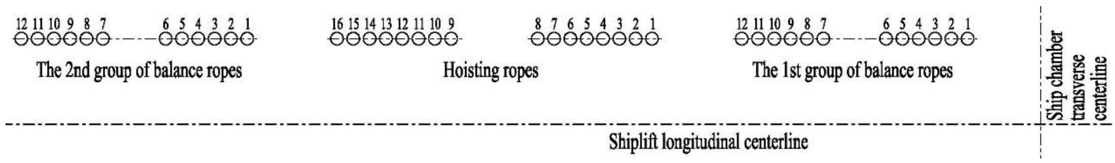


**Fig.9:** No. of the wire ropes on 1/4 ship chamber of the 200m/3000t FBFHS

**Table 2:** Tensions of 1st group of the balance ropes of the 200m/3000t FBFHS

| Condition | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| Case A Tension (kN) | 389.8 | 390.1 | 390.3 | 390.4 | 390.5 | 390.4 |
| Case B Tension (kN) | 334.3 | 336.5 | 338.2 | 339.4 | 340.1 | 340.3 |
| Condition | 7 | 8 | 9 | 10 | 11 | 12 |
| Case A Tension (kN) | 391.6 | 391.9 | 392.1 | 392.3 | 392.4 | 392.4 |
| Case B Tension (kN) | 352.0 | 354.4 | 356.7 | 358.5 | 360.0 | 361.0 |

**Table 3:** Tensions of 2st group of balance ropes of the 200m/3000t FBFHS

| Condition | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| Case A Tension (kN) | 396.0 | 396.0 | 396.0 | 395.9 | 395.8 | 395.6 |
| Case B Tension (kN) | 431.3 | 432.9 | 434.1 | 434.8 | 435.0 | 435.1 |
| Condition | 7 | 8 | 9 | 10 | 11 | 12 |
| Case A Tension (kN) | 393.8 | 393.8 | 393.7 | 393.5 | 393.4 | 393.1 |
| Case B Tension (kN) | 432.8 | 434.0 | 434.6 | 434.8 | 434.6 | 433.9 |

**Table 4:** Tensions of the hoisting ropes of the 200m/3000t FBFHS

| Condition | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| Case A Tension (kN) | 393.4 | 393.7 | 393.9 | 394.0 | 394.2 | 394.2 |
| Case B Tension (kN) | 373.4 | 376.2 | 378.5 | 380.6 | 382.4 | 383.8 |
| Condition | 7 | 8 | 9 | 10 | 11 | 12 |
| Case A Tension (kN) | 394.2 | 394.2 | 395.2 | 395.5 | 395.6 | 395.7 |
| Case B Tension (kN) | 385.0 | 385.8 | 407.9 | 410.6 | 413.0 | 415.2 |
| Condition | 13 | 14 | 15 | 16 | | |
| Case A Tension (kN) | 395.8 | 395.9 | 395.9 | 395.8 | | |
| Case B Tension (kN) | 417.0 | 418.6 | 420.0 | 420.8 | | |

The safety factor of the pitch stability of the ship chamber is calculated according to formula (18)~(28) and the parameters for the 200m/3000t FBFHS listed in the Table 5. The eigenvalues corresponding to the pitch vibration is -2.266×10-7±1.662×10-4*i* with the longitudinal central distance of the main hoist being 33.28m, but one pair of real eigenvalues ±1.155×10-3 appears with the longitudinal central distance of the main hoist being 33.278m, which means that the dynamic system is unstable. It can be considered that the critical longitudinal central distance is 33.279m, and the corresponding check safety factor is 1.73.

**Table 5:** Parameters for check of the ship chamber pitch stability of the 200m/3000t shiplift schemes

| **Type** | $J_h$(tm$^2$) | $d_h$(kN.m.s) | $r_d$(m) | $i$ | $k_r$(kN/m) |
|---|---|---|---|---|---|
| Winch type | $1.625\times10^5$ | $7.434\times10^4$ | 3.8 | 238.76 | $6.860\times10^4$ |
| Friction hoist type | $1.764\times10^5$ | $5.273\times10^4$ | 3.3 | 207.25 | $8.334\times10^4$ |
| **Type** | $J_s$(t.m2) | $C_s$(kN.m) | $J_w$(t.m$^2$) | $C_w$(kN.m) | $\omega_w$(1/s) |
| Winch type | $7.887\times10^6$ | $3.272\times10^7$ | $7.917\times10^6$ | $3.301\times10^7$ | 0.1651 |
| Friction hoist type | $5.621\times10^6$ | $4.720\times10^7$ | $5.480\times10^6$ | $2.285\times10^7$ | 0.1866 |

### 3.3 Comparison

(1) The essential safety factor $S_{FE}$ for fortification of TWLA for the scheme of FBWS is only 0.607 which shows that the fortification of TWLA is not realized. $S_{FE}$ for the scheme of the FBFHS is 1.424 and the FEA computation verifies that sliding of the wire ropes does not occur, which means that fortification for TWLA for the scheme is realized.

(2) The pitch stability safety factor $S_p$ of the scheme of the FBFHS is 1.73 which is 22.2% higher than that of the scheme of the FBWS.

(3) The length of the ship chamber of the FBFHS is evidently less than that of the FBWS, which means the ship chamber basement and the tower columns of the FBFHS occupy smaller space than that of the FBWS, so it owns more economic rationality.

## 4. Conclusion

This paper proposes the conception of FBFHSs in order to solve the problems of the failure in fortification of the TWLA, the deficiency in the pitch stability and the difficulty in layout of the main hoist existing in design of FBWSs with large tonnage of design ships and high lifts. The authors' main work includes:

(1) Propose compact layout of the friction hoist to supply the possibility of increasing the fortification ability and pitch stability of ship chambers.

(2) Carry out a test research which measures the friction factors under different lubricants.

(3) Put forward the design method for fortification of FBFHSs.

(4) Put forward the method for calculation of the pitch stability of ship chambers by building a coupling dynamic model for shiplifts.

(5) Conduct a comparison between the schemes of FBWS and FBFHS of the 200m/3000t shiplift, demonstrating the design method of FBFHS, and demonstrate the superiority of FBFHS over FBWS in the safety performances.

We conclude from the research that the FBFHS schemes are superior to the FBWS schemes in the safety performances such as the fortification of TWLA and the pitch stability of ship chambers for shiplifts with large tonnage of design ship and high lift.

## Acknowledgement

This research has been funded by the National Key Research and Development Program of China (2016YFC0402002).

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