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Dynamics of lashed trailers on board ships under the combined effect of roll and pitch motions

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ABSTRACT: In this paper we are extending our earlier work on the mathematical modelling of lashed trailer dynamics. The trailer can be placed anywhere on the vehicle deck(s) of a Ro/Ro ship. The paper begins with a discussion on the cargo-securing manual as applied to an existing ship. Thereafter we present the developed mathematical model where the excitation of the lashed trailer is provided by the combined roll and pitch motions, incited by steep waves encountering the ship from the side. The characteristics of the lashings, as well as those of the trailer support (trestle) and suspension systems are taken into account. Ship motions are simulated with a panel code and they are imported to the mathematical model of the lashed trailer, thus providing an integrated environment for the investigation of the cargo securing system on board a Ro/Ro ship.

1 INTRODUCTION

The stowage and securing of cargo units on board ships has not been left to chance by the international community. According to SOLAS Chapters VI and VII, a Cargo Securing Manual (CSM) is required for all types of ships engaged in the carriage of cargoes other than solid and liquids in bulk. Supplemented by IMO resolutions, ISO standards, classification society and national agency guidelines, a specific framework has been established for the effective implementation of this requirement (IMO 1981 & 1996; ISO 1989 & 1996; UK Department of Transport 1991; DNV 2002).

The preparation of CSM underlies a prescriptive calculation method by means of which a specific lashing arrangement may be judged as sufficient for the securing of the examined cargo unit(s) (e.g. a vehicle or a tier of containers). The method examines the adequacy of lashings towards cargo's transverse sliding and tipping and longitudinal sliding. The calculation of cargo forces at various positions on board is based on the expected accelerations, while corrections on these values are introduced in terms of the ship's speed, length and ratio of beam to metacentric height. By checking the static balance of external and stabilizing forces, assuming that the cargo is rigidly attached to the deck, is verified the adequacy of the lashing arrangement.

However, the assumption that the cargo is rigidly attached to the deck may not be sufficient especially when large ship motions occur. In earlier research a more realistic modeling of the dynamics of a lashed trailer on the deck was targeted. Key studies were those by Andersson (1983), Turnbull & Dawson (1994, 1998 & 1999) and Turnbull (2000). In the mathematical model of Turnbull & Dawson the trailer – lashing system is represented by a number of distributed lump masses (representing trailer mass) which are connected to the spine of the trailer. Trailer's chassis could be treated either as rigid or as flexible. Vehicle' suspension is modeled by a pair of springs and dampers while lashings are assumed to act as linear springs that can receive tension axially but only in the one direction. Turnbull & Dawson had assumed a sinusoidal-type motion of the ship with prescribed maximum amplitudes for rolling and pitching. On this basis they could calculate the loads on the lashings by a first principle approach i.e. by solving trailer motions equations. They have included also the securing of cargo inside the trailer and its influence on lashings forces. Rolling period and trailer's position on board proved to be the main parameters determining the maximum values of lashing loads. Taking a step forward, Themelis and Spyrou (2003) used the modeling approach of Turnbull & Dawson and connected it to the roll motion dynamics of the ship in beam waves. At that time a relatively simple rolling equation had been incorporated for the prediction of ship motion response. Another advance was that finite elements strength analysis was performed to critical parts of the lashing arrangement, such as the link point of lashings at the spine of trailer, including also the case of impulsive loading of slack lashings. The basic motive of that study was to correlate the wave environment with lashing arrangement loads and strength in a unified assessment.

In a recent study by Jiu et al (2007) car transport without lashings on board Ro/Ro ships was studied. A mechanical system, based on springs, masses and dumpers, for car – deck interactions has been developed where deck vibrations were also considered in a simplified manner. It was supported that in moderate seas cars can sustain transverse sliding with a strong dependence on the friction coefficient between car's tyres and deck as well as car position on the ship. The dynamics of unlashed cargo shift in irregular beam waves was studied also by Matusiak (2000). Nevertheless lashings are necessary when the wave induced loads are producing sliding forces that exceed the available friction force.

In the current study, further improvements on the approach of Themelis & Spyrou (2003) are presented. Firstly, the longitudinal motion of the trailer that is caused by the pitch motion is taken into account. This means that the lashing arrangement is deployed in 3 dimensions as it should be able to handle forces in all three main directions. Additionally, a panel seakeeping code is integrated with the analysis of lashings, in order to obtain a more realistic calculation of the rolling and pitching motion of the ship and also a more detailed representation of the hull.

2 PRESCRIPTIVE CALCULATION

According to IMO's Cargo Securing Manual (CSM) the key external forces that act on the trailer occur due to the longitudinal, transverse and vertical accelerations of the adjacent vehicle deck (DNV 2002). The force to be taken by the lashing arrangement should thus be calculated with the following formula:

$$F_{(x,y,z)} = ma_{(x,y,z)} + F_{w(x,y)} + F_{s(x,y)} \tag{1}$$

All symbols are defined in the Nomenclature near the end of the paper. Also, in a number of Tables in the Appendix are collected the recommended acceleration values according to the so called "LRS method". These should be considered as valid under the following conditions: operation in unrestricted area and during the whole year, 25 days duration of voyage, ship length of 100 m, service speed of 15 knots and $B/GM \geq 13$. For values different than the above, specific corrections have to be applied (the Tables that provide the corrections can be found also in the Appendix). The effectiveness of the securing device is examined in a quasi-static sense, by considering the balance of forces and moments against transverse sliding, transverse tipping and longitudinal sliding. More specifically, the following conditions need to be verified:

For transverse sliding:

$$F_y \leq \mu mg + CS_1 f_1 + CS_2 f_2 + ...CS_n f_n \tag{2}$$

Transverse tipping:

$$F_y h \leq bmg + CS_1 c_1 + CS_2 c_2 + ... + CS_n c_n \tag{3}$$

Longitudinal sliding:

$$F_x \leq \mu(mg - F_z) + CS_1 f_1 + ... + CS_n f_n \tag{4}$$

It is thus obvious that CSM, through the prescribed calculation process, does not "permit" any interference of system dynamics. Consequently, characteristics of trailer parts, such as the trestle and the suspension system, could not be taken into account. Moreover, this calculation procedure results in assuming equal loads for all lashings, that is not necessary to be happening. It is worth noting also that, the prescribed procedure leaves no space for extracting the safety factor for given weather conditions since no such connection is found in the calculation process.

Application of the above for a typical trailer placed on an existing ferry follows. In Table 1 are supplied the key data of the trailer and of the ship. The lashing arrangement that was adopted in order to apply the CSM calculation is shown schematically in Figure 1.One should also note that the CSM method does not discriminate between a truck-with-trailer and a trailer-trestle arrangement. In either case it treats the object as a single heavy body.

The calculated values of forces and moments corresponding to two different positions of the trailer are collected in Tables 2 to 4. The trailer has been placed longitudinally at 0.9 L (measured from AP), initially on the main and later also on the upper deck. It should be noted here that the placement on the upper deck is only of academic interest, because trailers are not supposed to be placed on that deck according to the GA of the examined ferry. The main intention is to show how the anticipated accelerations and thus the external forces change regarding the vertical position of the trailer.

According to the obtained results, in the upper deck scenario the balance between forces and moments cannot be achieved for longitudinal sliding with the specific lashing arrangement shown in Figure 1.

3 MODELING OF TRAILER AND LASHINGS

A typical trailer with its lashing arrangement is shown in Figure 2. The spine of the trailer is assumed to be torsionally flexible in the longitudinal direction but rigid

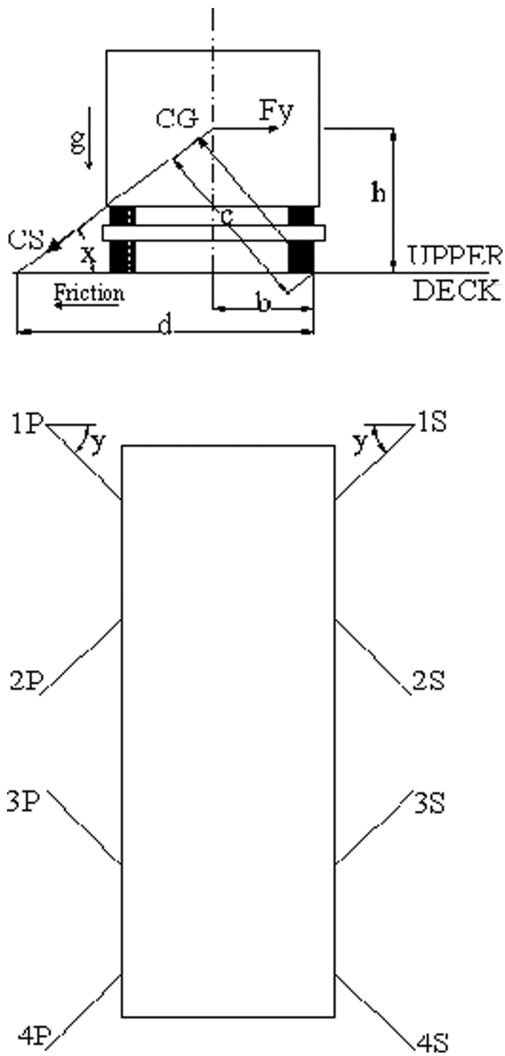

Figure 1. Transverse section and upper view of trailer.

in bending (Themelis & Spyrou 2003). The mass of the trailer is represented by a number of (equal) masses, in this case six, distributed as shown in Figure 3. We have assigned one mass above the trestle, one above the suspension and one at each lashing spar. The spars are those points where the lashings are connected to the spine. They are assumed to be rigid and capable of independent rotation about the center of the spine. The trestle supports the trailer when it is not connected to a truck. The suspension is assumed to lie exactly above the wheels.

As it is intended to study the motions of the trailer due to rolling and pitching, lashings should be capable to take loads not only on a vertical plane perpendicular to the spine as in the case of pure roll, but also on a plane parallel to the spine. As a result the motion of each mass can be described by its vertical, z transverse, y and longitudinal x, displacements relatively to the deck, as well as by the relative angles of the trailer θ, α respectively in roll and pitch, that cause rotation of the spars and thus possible elongation of lashings. The motion of each mass is thus described in three degrees of freedom, plus two degrees per mass for the rotation

Table 1. Ship and trailer/lashing system characteristics.

Trailer specification		
Mass	m	30 tons
Distance of CG from deck	h	1.85 m
Lever arm of gravitational force	b	1.25 m
Horizontal distance of securing point from axis of tipping	d	2.75 m
Friction coefficient (steel – rubber)	μ	0.3
Lashings' specification		
Arrangement	8 chains, 4 in each side symmetrically	
Maximum Securing Load (M.S.L.)	M.S.L.	98.1 kN
Calculated Strength (C.S.=M.S.L./1.5)	C.S.	65.4 kN
Vertical securing angle	x	45^0
Horizontal securing angle	y	45^0
Ship data		
Length between perpendiculars	L_{BP}	111.80 m
Breadth	B	18.9 m
Draught	T	4.9 m
Depth	D	7.25 m
Speed	V_S	24 kn
Metacentric height	GM	2.019 m
Breadth/Metacentric height ratio	B/GM	9.361
Displacement	Δ	6045 ton

Table 2. Balance check for transverse sliding [formula (2)].

	Transverse sliding (left side of trailer) ($0.9L_{BP}$)				
Position on ship	F_y (kN)	$\mu * m*g$ (kN)	Lashings (kN)	Total (kN)	Balance
Main deck	237.2	88.3	170.1	258.3	Yes
Upper deck	275.3	88.3	170.1	258.3	No

Table 3. Balance check for transverse tipping [formula (3)].

	Transverse tipping (left side of trailer) ($0.9L_{BP}$)				
Position on ship	F_y*h (kN*m)	$b*m*g$ (kN*m)	Lashings (kN*m)	Total (kN*m)	Balance
Main deck	438.8	367.9	359.7	727.6	Yes
Upper deck	509.3	367.9	359.7	727.6	Yes

Table 4. Balance check for longitudinal sliding [formula (4)].

	Longitudinal sliding ($0.9L_{BP}$)				
Position on ship	F_x (kN)	$\mu^*(m^*g - F_z)$ (kN)	Lashings (kN)	Total (kN)	Balance
Main deck	108	−9.4	170	160.6	Yes
Upper deck	141	−14.4	170	155.6	Yes

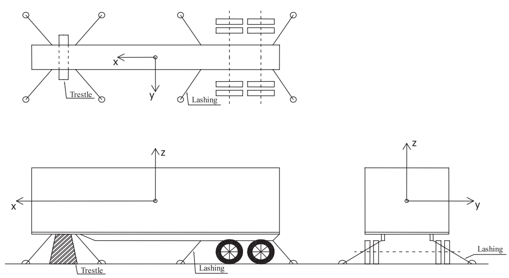

Figure 2. A typical trailer with its lashing arrangement.

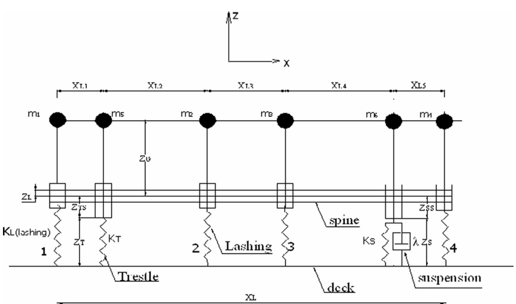

Figure 3. Simplified model of trailer.

of the spar. With six masses representing the trailer, it should thus have 30 degrees of freedom and solution of an equal number of differential equations will be required.

The trestle is usually made of steel and it could be modeled in the y-z plane by two hard linear springs connected in parallel mode as depicted in Figure 4. In the x-z plane it is modeled by two linear springs in perpendicular mode as shown in Figure 5.

Angles ϕ and ξ correspond respectively to the roll and pitch angles of the ship (and thus of the deck where the trailer is located) with respect to an earth fixed system (absolute angle). The vertical forces of the two

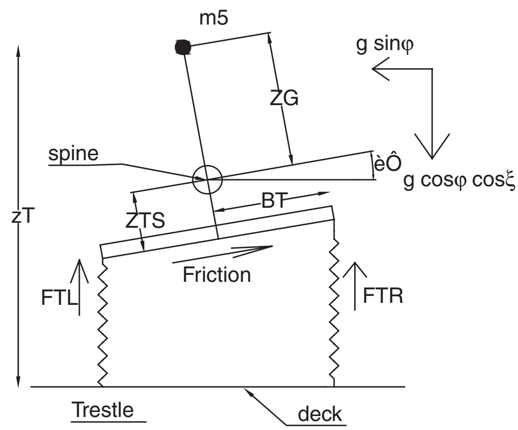

Figure 4. Modeling of the trestle in the y-z plane.

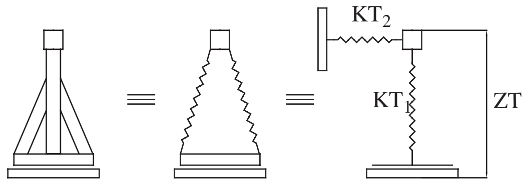

Figure 5. Model of the trestle and its analysis to springs.

springs of the trestle are expressed according to Hook's law for linear springs:

$$F_{TL,R} = K_{T1}[z_T - (Z_G + Z_{TS})\cos\theta_T \cos\alpha \mp B_T \sin\theta_T - X_T \sin\alpha - Z_T] \tag{5}$$

Whereas the (-) is for the force acting on the left side spring. The total vertical force acting on the trestle is

$$F_T = F_{TL} + F_{TR} \tag{6}$$

The longitudinal forces of the horizontal spring are expressed likewise (see also Figure 6):

$$F_{LT} = K_{T2} X_T (1 - \cos\alpha) \tag{7}$$

The suspension is modelled in a similar way to the trestle, adding dampers in parallel to the springs, (Figure 7). The suspension is not supposed to receive forces at the x-z plane. The force at each side of the suspension due to connection of the spring and the damper is expressed by the following equation:

$$F = Kx + \lambda_I \dot{x} \tag{8}$$

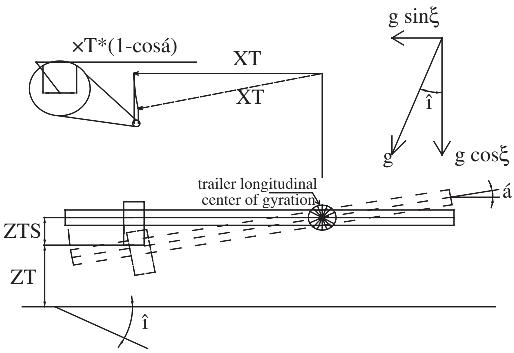

Figure 6. Modeling of the trestle in the x-z plane.

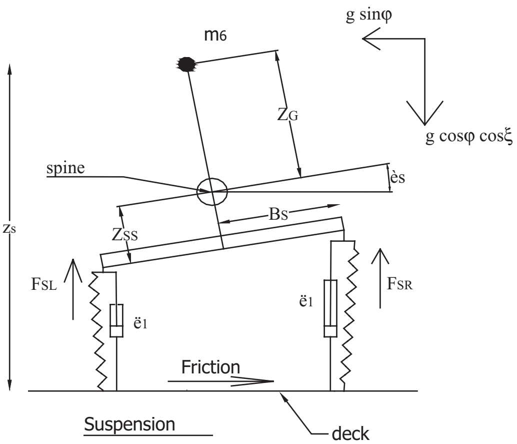

Figure 7. Modeling of the suspension at the y-z plane.

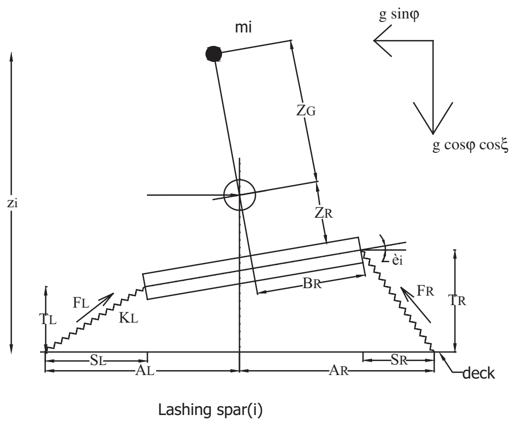

Figure 8. Modeling of a lashing spar at the y-z plane.

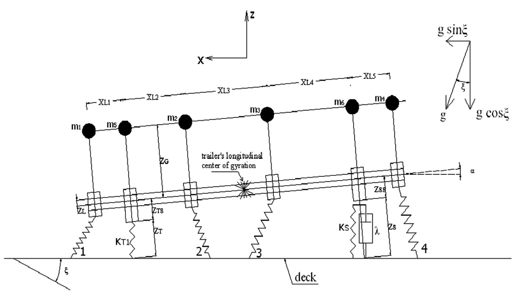

Figure 9. Modeling of a lashing spar at the x-z plane.

where λ_1 is the damping coefficient. According to Figure 7, the analytical expression of the forces acting from left and right on the suspension can be derived:

Right and Left Force:

$$
\begin{aligned}
&F_{SR} = K_S[z_S - (Z_G + Z_{SS})\cos\theta_S \cos\alpha \pm \\
&B_S \sin\theta_S \cos\alpha - X_S \sin\alpha - Z_S] + \\
&\lambda_l[\dot{z}_S + (Z_G + Z_{SS})\dot{\theta}_S \sin\theta_S \cos\alpha + \\
&(Z_G + Z_{SS})\dot{\alpha}\cos\theta_S \sin\alpha \mp B_S\dot{\theta}_S \cos\theta_S \cos\alpha \pm \\
&B_S\dot{\alpha}\sin\dot{\theta}_S \sin\alpha - X_S\dot{\alpha}\cos\alpha]
\end{aligned}
\tag{9}
$$

In the above equation the upper signs (whenever they appear) correspond to the right lashing force and the lower signs to the left lashing force.

The total force acting on the suspension is:

$$F_S = F_{SL} + F_{SR} \tag{10}$$

In the current model we take into account the frictional force which develops between the deck and the tires and the frictional force between the trestle and the trailer.

The lashings are modelled as linear springs connected to the trailer spine at the spars. The load that a lashing is expected to take is expressed by the equation:

$$F = K \cdot (L - L_0) \tag{11}$$

In the above expression L, L_0 and K are respectively, the instantaneous length of the spring, the initial length and the stiffness of the lashing. The length L_{Li} of the left lashing i is :

$$L_{Li} = \sqrt{S_{Li}^{\ 2} + T_{Li}^{\ 2} + C_{Li}^{\ 2}} \tag{12}$$

S_{Li}, T_{Li}, C_{Li} are respectively, the transverse, vertical and longitudinal components of L_{Li}. Similarly, S_{Ri}, T_{Ri}, C_{Ri} refer to the right lashing of length L_{Ri}.

It is necessary to mention that the angle α is the same for all the masses due to our assumption that the trailer is of 'flat-bed' type, which means that it is very stiff in bending. Thus no bending arrow of the spine is taken into account, since it is expected to be insignificant compared to the systems response. The analytical expressions for the constants S_{Li}, T_{Li}, C_{Li} are provided (see also Figures 8 and 9):

$$S_{Li} = A_{L(i)} - y_{(i)} + Z_{L(i)} \sin\theta_i - B_{L(i)} \cos\theta_i \tag{13}$$

$$T_{Li} = z_{(i)} - Z_G \cos\theta_i - Z_{L(i)} \cos\theta_i - B_{L(i)} \cos\theta_i - X_{L(i)} \sin\alpha \tag{14}$$

$$C_{Li} = x_{(i)} + (-1)^{i+1}(X_{DL(i)} - X_{L(i)}) - Z_{L(i)} \sin\alpha - \mathrm{X}_{L(i)}(1 - \cos\alpha) \tag{15}$$

The total load on a left lashing at position i should be:

$$F_{Li} = (L_{Li} - L_{0i})K_{Li} \tag{16}$$

Furthermore the total length of a lashing is expressed as follows:

$$L_{Li} = \sqrt{S_{Li}^{\ 2} + T_{Li}^{\ 2} + C_{Li}^{\ 2}} \tag{17}$$

At this point we should mention that the lashings are either chains or steel ropes which cannot handle compressional stresses (negative elongation is not possible). Thus:

$$L_{Li} \geq L_{0i} \tag{18}$$

We insert the following extra assumptions, since neither of the length components can receive negative values:

$$S_{Li} \geq S_{L0} \tag{19}$$

$$T_{Li} \geq T_{L0} \tag{20}$$

$$C_{Li} \geq C_{L0} \tag{21}$$

Now we are in a position to express the components $F_{LZi}, F_{LYi}, F_{LXi}$ on the basis of the total loads:

$$F_{LZi} = F_{Li} \frac{T_{Li}}{L_{Li}} \tag{22}$$

$$F_{LYi} = F_{Li} \frac{S_{Li}}{L_{Li}} \tag{23}$$

$$F_{LXi} = F_{Li} \frac{C_{Li}}{L_{Li}} \tag{24}$$

In similar terms, the equations for the lashings on the right side of the trailer can be derived.

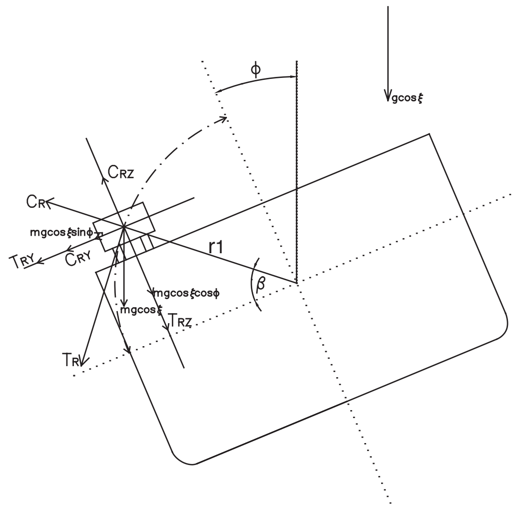

Figure 10. Forces on the trailer due to roll motion of the ship.

4 EQUATIONS OF MOTION OF LASHED TRAILER

The roll and pitch motions of the ship incur inertial forces on the trailer. We have made the assumption that the motion of the trailer does not affect the motion of the ship. In addition, we have assumed that the deck is completely rigid. Ship motions in specific wave conditions have been predicted by running the well known panel code SWAN2002 (Boston Marine Consulting 2002). Further information about the use of this software will be presented in a later section. According to Figure 10, the total transverse and vertical forces due to the roll motion of the ship contain components of the tangential force T, the centrifugal C and the gravity force mg. In more detail, the resultant longitudinal and vertical (to the deck) forces due to roll motion are expressed as (Bhattacharyya 1978):

$$R_{Ry} = -mg \sin\phi \cos\xi - T_{Ry} - C_{Ry} \tag{25}$$

$$R_{Rz} = mg \cos\phi \cos\xi - T_{Rz} - C_{Rz} \tag{26}$$

$$T_{Ry} = mr_1 \ddot{\phi} \sin\beta \tag{27}$$

$$T_{Rz} = mr_1 (\dot{\phi})^2 \cos\beta \tag{28}$$

$$C_{Ry} = mr_1 (\dot{\phi})^2 \cos\beta \tag{29}$$

$$C_{Rz} = mr_1 (\dot{\phi})^2 \sin\beta \tag{30}$$

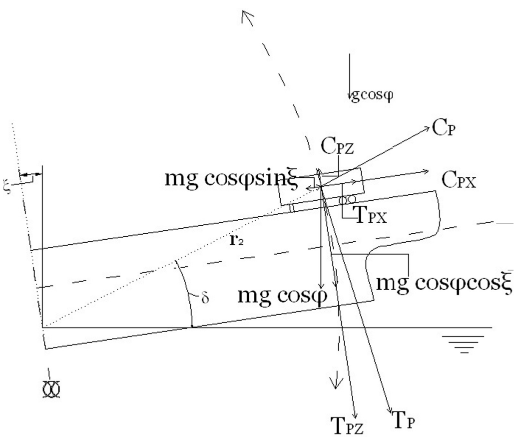

Figure 11. Forces on the trailer due to pitch motion of the ship.

Similarly, for the total longitudinal and vertical forces due to the pitch motion of the ship, the following expressions are derived (Figure 11):

$$R_{Pz} = mg\cos\phi\cos\xi + T_{Pz} - C_{Pz} \quad (31)$$

$$R_{Px} = mg\cos\phi\cos\xi - C_{Px} - T_{Px} \quad (32)$$

$$T_{Pz} = mr_2\ddot{\xi}\cos\delta \quad (33)$$

$$T_{Px} = mr_2\ddot{\xi}\sin\delta \quad (34)$$

$$C_{Px} = mr_2(\dot{\xi})^2\cos\delta \quad (35)$$

$$C_{Pz} = mr_2(\dot{\xi})^2 sin\delta \quad (36)$$

Knowledge of the time histories of ϕ, ξ let us analytically calculate the forces on the trailer due to the roll and pitch motions of the ship which could reveal a trailer's tendency to move against the deck upon which it is secured.

Motion of the mass above trestle
According to Figures 10 and 11, the equations of motion of the mass m_5 above the trestle in the longitudinal, lateral and vertical direction of the ship should be respectively:

$$m_5\ddot{x}_T = R_{PX5} - F_{LT} \quad (37)$$

$$m_5\ddot{y}_T = R_{RY5} - F_T\mu_T sign(\dot{y}_T) \quad (38)$$

$$m_5\ddot{x}_T = R_{PZ5} + R_{RZ5} - F_{LT} \quad (39)$$

The equation of angular momentum of mass m_5 is:

$$\begin{aligned} I_5\ddot{\theta}_T = &-F_{TR}[(Z_G + Z_{TS})\sin\theta_T\cos\alpha + B_T\cos\theta_T] + \\ &F_{TL}[-(Z_G + Z_{TS})\sin\theta_T\cos\alpha + B_T\cos\theta_T] - \\ &F_T\mu_T(Z_G + Z_{TS})sign(\dot{y}_T) - S_{K1}(\theta_T - \theta_1) - \\ &S_{K2}(\theta_T - \theta_2) \end{aligned} \quad (40)$$

where S_{K1}, S_{K2} are respectively the torsional stiffness of the spine between the trestle – lashing 1 and trestle – lashing 2.

Motion of the mass above suspension

$$m_6\ddot{x}_S = R_{PX6} + F_S\mu_S sign(\dot{x}_S) \quad (41)$$

$$m_6\ddot{y}_S = R_{RY6} - F_S\mu_S sign(\dot{y}_S) \quad (42)$$

$$m_6\ddot{z}_S = R_{PZ6} + R_{RZ6} - F_S \quad (43)$$

$$\begin{aligned} I_6\ddot{\theta}_S = &-F_{SR}[(Z_G + Z_{SS})\sin\theta_S\cos\alpha + B_S\cos\theta_S] + \\ &F_{SL}[-(Z_G + Z_{SS})\sin\theta_S\cos\alpha + B_S\cos\theta_S] - \\ &F_S\mu_S(Z_G + Z_{SS})sign(\dot{y}_S) - S_{K4}(\theta_S - \theta_3) - \\ &S_{K5}(\theta_S - \theta_4) \end{aligned} \quad (44)$$

where S_{K4}, S_{K5} are respectively, the torsional stiffness of the spine between the suspension – lashing 3 and suspension – lashing 4.

Motion of the masses at lashing spars

$$m_i\ddot{x}_i = R_{PXi} + F_{Xi} \quad (45)$$

$$m_i\ddot{y}_i = R_{RYi} + F_{Yi} \quad (46)$$

$$m_i\ddot{z}_i = R_{PZi} + R_{RZi} - F_{Zi} \quad (47)$$

$$\begin{aligned} I_i\ddot{\theta}_i = &-F_{RYi}[(Z_{Ri}\cos\theta i\cos\alpha - B_{Ri}\sin\theta i)] + \\ &F_{LZi}[-Z_{Li}\sin\theta_i\cos\alpha + B_{Li}\cos\theta_i] - \\ &F_{RZi}[Z_{Ri}\sin\theta i\cos\alpha + B_{Ri}\cos\theta i)] - \\ &F_{LYi}[Z_{Li}\cos\theta i\cos\alpha + B_{Li}\sin\theta i] - \\ &S_{Ki,i-1}(\theta_i - \theta_{i-1}) - S_{Ki,i+1}(\theta_i - \theta_{i+1}) \end{aligned} \quad (48)$$

where $S_{Ki,i-1}$ is the torsional stiffness of the spine between the mass i and the mass i-1.

As presented above the trailer has 30 degrees of freedom but the presented equations are only 24. The six remaining equations express, per mass (i.e. we should use six of these) the moments about the longitudinal center of the trailer:

$$
\begin{aligned}
& I_{total} \cdot \ddot{\alpha} = F_{LX(i)} \cdot [Z_{L(i)} \cdot \cos\theta_i + X_{L(i)} \cdot \sin\alpha] + \\
& F_{RX(i)} \cdot [Z_{R(i)} \cdot \cos\theta_i + X_{R(i)} \cdot \sin\alpha] - \\
& -F_{LZ(i)} \cdot [X_{L(i)} \cdot (1-\cos\alpha) - Z_{L(i)} \cdot \cos\theta_i \cdot \sin\alpha] - \\
& F_{RZ(i)} \cdot [X_{R(i)} \cdot (1-\cos\alpha) - Z_{R(i)} \cdot \cos\theta_i \cdot \sin\alpha] \\
& +m_i \cdot g \cdot \cos\phi \cdot \cos\xi \cdot [X_{L(i)} \cdot (1-\sin\alpha) - Z_G \cdot \cos\theta_i \cdot \sin\alpha] + \\
& +m_i \cdot g \cdot \cos\phi \cdot \sin\xi \cdot [X_{L(i)} \cdot \sin\alpha + Z_G \cdot \cos\theta_i \cdot \cos\alpha] + \\
& -F_{LT} \cdot [Z_{TS} \cdot \cos\theta_5 \cdot \cos\alpha + X_T \cdot \sin\alpha] - \\
& F_T \cdot [X_T \cdot \cos\alpha + Z_{TS} \cdot \cos\theta_5 \cdot (\cos\alpha - 1)] + \\
& +F_S \cdot [X_S \cdot \cos\alpha + Z_{SS} \cdot \cos\theta_6 \cdot (\cos\alpha - 1)]
\end{aligned}
\tag{49}
$$

The above system of differential equations has been solved by using Mathematica 5 (Wolfram Media Inc. 2001).

5 PREDICTION OF SHIP MOTION

As already said, the necessary input-excitation to the above presented mathematical model is produced by using a ship motions program, namely 'SWAN2 2002'. This allows for parameters defining hull geometry, the wave environment, speed etc. to be taken into to account in a detailed way and hopefully to produce realistic input to the mathematical model of lashings/trailer dynamics. SWAN2 2002 is a computer program for the analysis of the steady and unsteady zero-speed and forward-speed free surface flows past ships which are stationary or cruising in water of infinite or finite depth in a channel. SWAN2 2002 solves the steady and unsteady free-surface potential flow problems around ships using a three-dimensional 'Rankine-source' panel method in the time domain, employing distribution of quadrilateral panels over the ship hull and the free surface.

Characteristic numerical free roll decay tests for roll and pitch of the examined ferry are presented respectively in Figures 12 and 13.

The roll and pitch natural periods of the ship were found to be respectively 10 and 8.5 s. Periods of wave encounter spanning the range between these two values were examined. In Figures 14 and 15 are shown examples of ship motion predictions in waves, that refer to an encounter period in between the roll and pitch natural periods $T_e = \frac{T_{ROLL}+T_{PITCH}}{2}$, speed fixed at the service value and angle of encounter 135^0 (stern quartering, head seas correspond to 0^0). These require a wave period 13.18 s which corresponds to $\lambda/L = 2.43$. The assumed wave height was 4.725 m thus the steepness was $H/\lambda = 1/59$.

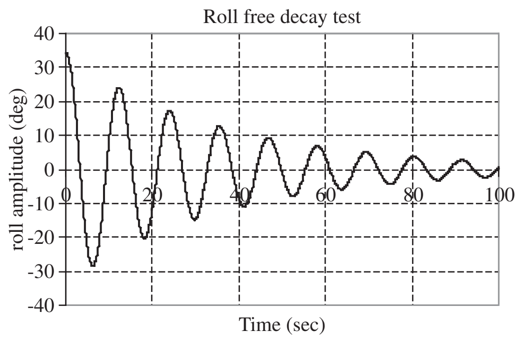

Figure 12. Execution of free roll decay test.

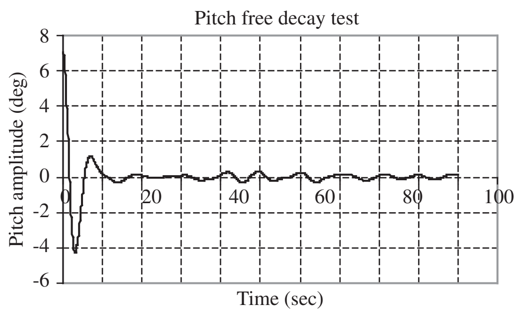

Figure 13. Execution of a free pitch decay test.

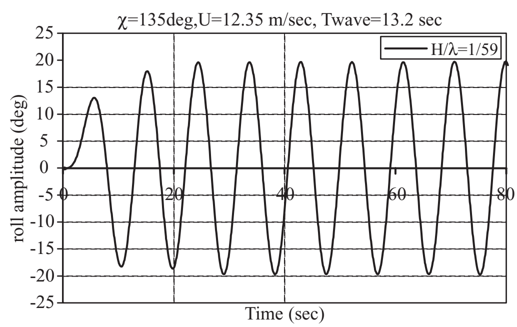

Figure 14. Ship roll motion results.

6 IMPLEMENTATION OF MATHEMATICAL MODEL FOR TRAILER AND LASHINGS

The input data of the mathematical model are presented below in Table 5. The lashing arrangement is similar to that of Figure 1. Firstly, we calculated the forces of all the lashings for encounter angle 135^0, $H_{wave} = 4.725$ m, $T_{wave} = 10$s (which is the resonance roll period of the ship). Therefore a more severe condition was targeted. In Figure 16 is shown the development of force on lashing 1 through time. The

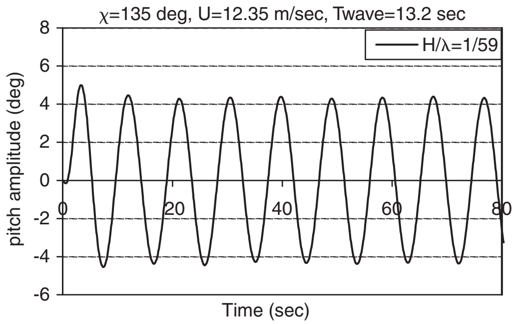

Figure 15. Ship pitch motion results.

Table 5. Trailer and lashings characteristics.

Trailer-lashings characteristics			
A_L, A_R(m)	1.5	$C_{Li} = C_{Ri}$(m)	0.5
B_S, B_T(m)	0.9	$R_{Li} = R_{Ri}$(m)	1.154
B_L, B_R(m)	0.9	λ(kN*s/m)	15
m_5, m_6(tons)	10	X_S(m)	10.5
m_i (tons), i=1,..4	2.5	$X_{L(i)}, X_{R(i)}$(m)	1, 2, 4.5, 2,1
I_5, I_6(tons*m^2)	5	Z_G(m)	0.85
I_i (tons*m2), i=1,..4	1.25	Z_S(m)	0.85
K_T(MN*m)	40	Z_T(m)	1
K_S(MN*m)	1	Z_{SS}(m)	0.15
K_{Li}(MN/m/m)	8	Z_{TS}(m)	0
S_K(MN*m/rad)	1.5	Z_{Li}, Z_{Ri}(m)	0.15
$S_{Li} = S_{Ri}$(m)	0.6	μ_T	0.3
$T_{Li} = T_{Ri}$(m)	0.85	μ_S	0.5

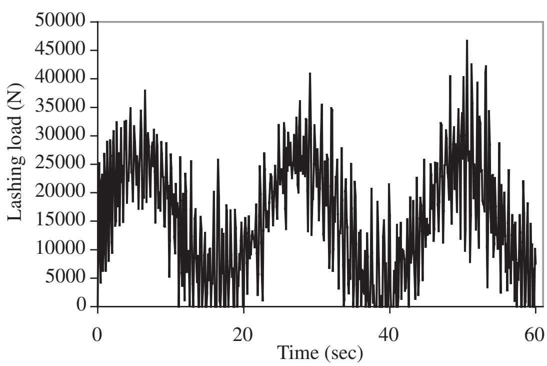

Figure 16. Force on left lashing 1.

suspension force for the same condition is provided in Figure 17. Thereafter we determined the lashing loads for all lashings, for encounter scenarios such as 10, 45, 60, 90, 120, 135 and 170 deg. In Figures 18 to 20 are shown plots of the lashing load as function of the angle of encounter and the wave period.

The difference between the values of the left and right lashing No 1 is attributed to the fact that the wave meets the ship from the port side.

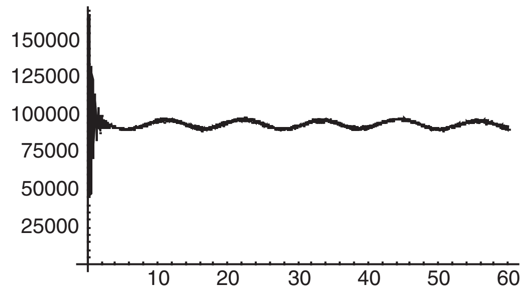

Figure 17. Force on the suspension.

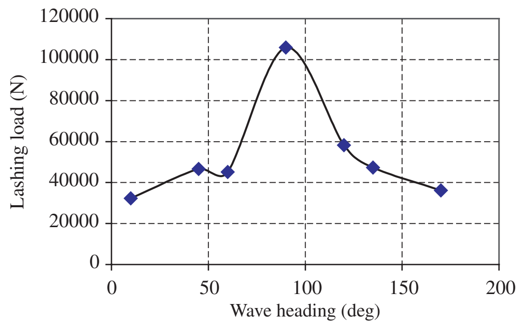

Figure 18. Force comparison on left lashing 1 for different encounter angles.

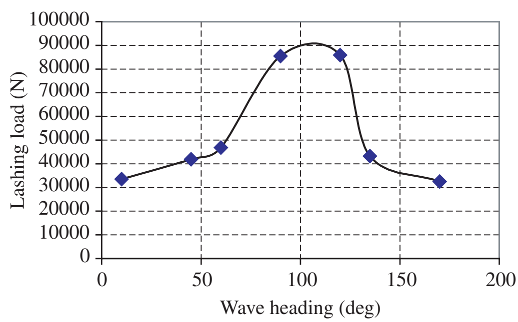

Figure 19. Force comparison on right lashing 1 for different encounter angles.

Secondly, the wave period was varied assuming beam seas and $H_{wave} = 4.725$ m.

At this point it is verified that the biggest forces on the lashings are developed for the wave period that is equal to the resonance roll period of the ship.

As a final step, the position of the trailer on the ship regarding the lashing loads was examined (see Figure 21). The trailer is located consecutively on the main and upper deck and in a longitudinal position at 0.9 L from aft perpendicular. Four different positions

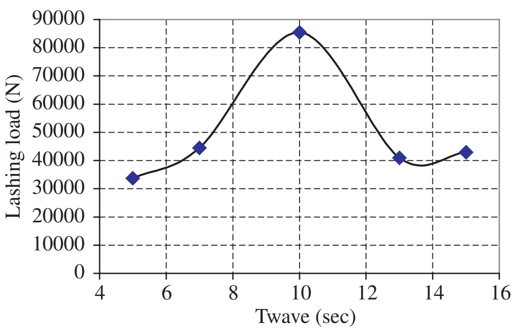

Figure 20. Force on right lashing 1 for different wave periods.

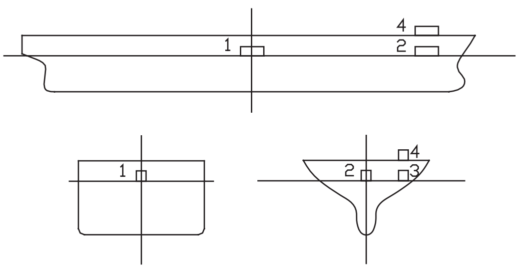

Figure 21. Schematics of the trailer's 4 different positions on the ship.

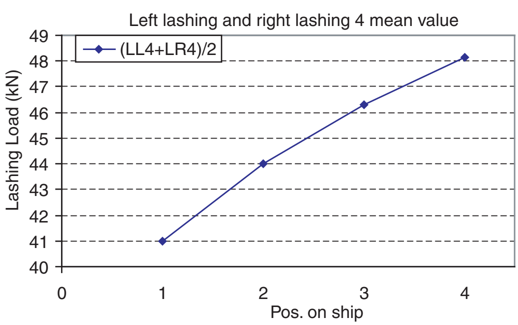

Figure 22. Force mean value on left and right lashing 4 according to the trailer's position.

of the trailer are shown in Figure 21. Figure 22 shows the mean value of lashing 4 (left and right) for the four different cases-positions. For this case the encounter angle is $\chi = 135^0$, $H_{wave} = 4.725$m and $T_{wave} = 13.2$s.

1st position: on main deck and about the gyration center of roll and pitch.
2nd position: on main deck, moved at the x-axis direction for 45 m.
3rd position: on main deck, moved at the x-axis direction for 45 m and y-axis direction for 8.5 m.
4th position: as in '3rd position' but the trailer has moved at the z-axis direction (upwards) for 8 m.

From Figure 22 it is apparent that the load on the lashings is increasing as the trailer moves away from the ships transverse and longitudinal center of gyration as expected.

7 CONCLUDING REMARKS

We have developed an integrated 3-dimensional approach for estimating the sufficiency of lashing systems on board Ro/Ro ships, taking into account ship motion prediction-simulation and trailer dynamics.

Whereas the calculation method described in the Cargo Securing Manual assumes an equal loading of lashings, according to the presented modeling each lashing could contribute differently to the total stabilizing force. This results from the fact that the modeling the trailer takes into account the existence of the trestle and suspension, as well as the torsional flexibility of its spine allowing for a most realistic approach.

Furthermore, the factors that significantly affect the lashing loading are briefly described next:

By changing the wave period there is an increase to the lashing forces up to the resonant roll period and a decrease after that.

The greater lashing loads are caused by 'beam seas', which once again confirms that roll motion is the dominant factor for the lashing load.

The farthest the lashing position of the trailer on the ship compared to the gyration centers of pitch and roll, the biggest the forces on the lashing systems.

It is apparent that the developed model has considerable complexity despite the several simplifying assumptions that were made at it was set up. Since the safe transportation of cargo is of major importance, we hope that this paper will provide incentive for further research on this topic (which we think that it could soon be definitively solved), especially since research in this area is somewhat limited.

NOMENCLATURE

$a_{(x,y,z)}$:	longitudinal, transverse and vertical accelerations according to CSM.
A_L, A_R :	left and right distances between the lashings points on the deck and the spine.
B_L, B_R :	left and right width of the chassis at the lashing spars.
B_S, B_T :	half width of the chassis at the suspension and trestle.
c_i :	lever-arm of securing force.
CS_i :	calculated strength of securing devices.
$F_{(x,y,z)}$:	longitudinal, transverse and vertical forces.
f_i :	a function of μ and the vertical securing angle h.

$F_{LX,LY,LZ}$: left horizontal, lateral, vertical force component acting on the i-lashing respectively.
F_{LT} : total horizontal force acting on the trestle.
$F_{RX,RY,RZ}$: right horizontal, lateral, vertical force component acting on the i-lashing respectively.
F_S : total vertical force acting on the suspension.
$F_{s(x,y)}$: longitudinal and transverse forces by sea sloshing.
$F_{SL,SR}$: vertical force at the left, right side of the suspension respectively.
F_T : total vertical force acting on the trestle.
$F_{TL,TR}$: vertical force at the left, right side of the trestle respectively.
$F_{w(x,y)}$: longitudinal and transverse forces by wind pressure.
$F_{x,y,z}$: longitudinal, transverse, vertical force from load assumption respectively.
H : wave height.
I_i : moments of inertia of each mass about its centre.
I_{total} : moments of inertia of all the masses about its longitudinal centre.
$K_{Li,Ri}$: stiffness of left and right hand lashing i.
K_S : stiffness of the springs of the suspension at z-axis.
K_{T1} : stiffness of the springs of the trestle at z-axis.
K_{T2} : stiffness of the spring of the trestle at x-axis.
L_0 : initial lashing length.
m : mass of unit.
m_i : lumped mass at position i.
$S_{L0,R0}$: initial lateral length component of left, right lashing respectively.
$T_{L0,R0}$: vertical initial length component of left, right lashing respectively.
$X_{DL(i),DR(i)}$: distance at the left, right side of the trailer between its i-lashing point on deck and its longitudinal centre of gyration respectively.
$X_{Li,Ri}$: distances between the masses.
X_S : horizontal distance of suspension from its longitudinal centre of gyration.
X_T : horizontal distance of trestle from its longitudinal centre of gyration.
Z_G : height of the masses above the spine.
Z_{Li}, Z_{Ri} : vertical distance between the $\underline{i}$th spar and the spine.
Z_S : vertical distance of mass $m_S = m_6$ from deck.
z_S : height of the suspension after loading.
Z_S : vertical distance of mass $m_S = m_6$ from deck.
Z_{SS} : vertical distance between the top of suspension and the spine.
Z_T : vertical distance of mass $m_T = m_5$ from deck.
z_T : height of the trestle after loading.
Z_{TS} : vertical distance between the top of the trestle and the spine.

Greek symbols

α : trailer response angle to pitch motion.
θ_i : response angle of mass m_i to roll motion.
θ_S : response angle of mass $m_S = m_6$ to roll motion.
θ_T : response angle of mass $m_T = m_5$ to roll motion.
λ : wave length.
λ_1 : damping constant of the suspension.
μ_S : friction coefficient between the suspension and the deck.
μ_T : friction coefficient between the trestle and the trailer.
ω_w : wave frequency.

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APPENDIX

The prescribed acceleration values according to trailer's position can be obtained from Tables 6 to 8 given below.

Table 6. Basic accelerations according to the L.R.S. method.

Transverse acceleration a_y (m/sec^2)												Longit. accel. a_x (m/sec^2)
Upper deck	7.1	7.1	6.9	6.8	6.7	6.7	6.8	6.9	7.1	7.4	7.4	3.8
Main deck	6.5	6.5	6.3	6.1	6.1	6.1	6.1	6.3	6.5	6.7	6.7	2.9
Lower deck	5.9	5.9	5.6	5.5	5.4	5.4	5.5	5.6	5.9	6.2	6.2	2.0
Tank top	5.5	5.5	5.3	5.1	5.0	5.0	5.1	5.3	5.5	5.9	5.9	1.5
L(m)	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1	L(m)
	Vertical acceleration a_z (m/sec^2)											
	7.6	7.6	6.2	5.0	4.3	4.3	5.0	6.2	7.6	9.2	9.2	

Table 7. Correction factor for ship speed and length.

Ship length (m)											
Speed (kn)	50	60	70	80	90	100	120	140	160	180	200
9	1.20	1.09	1.00	0.92	0.85	0.79	0.7	0.63	0.57	0.53	0.49
12	1.34	1.22	1.12	1.03	0.95	0.90	0,79	0.72	0.65	0.60	0.56
15	1.49	1.36	1.24	1.15	1.07	1.00	0.89	0.80	0.73	0.68	0.63
18	1.64	1.49	1.37	1.27	1.18	1.10	0.98	0.89	0.82	0.76	0.71
21	1.78	1.62	1.49	1.38	1.29	1.21	1.08	0.98	0.90	0.83	0.78
24	1.93	1.76	1.62	1.50	1.40	1.31	1.17	1.07	0.98	0.91	0.85
27	2.08	1.90	1.75	1.62	1.51	1.41	1.27	1.16	1.06	0.99	0.92
30	2.23	2.04	1.88	1.74	1.62	1.51	1.37	1.25	1.14	1.07	0.99

Table 8. Correction factor for B/GM<13.

	B/GM						
	7	8	9	10	11	12	13 or above
Upper deck	1.56	1.40	1.27	1.19	1.11	1.05	1.0
Main deck	1.42	1.30	1.21	1.14	1.09	1.04	1.0
Twin deck	1.26	1.19	1.14	1.09	1.06	1.0	1.0
Lower deck	1.15	1.12	1.09	1.06	1.04	1.02	1.0