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High speed marine craft motion mitigation using flexible hull design

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ABSTRACT

High speed marine craft motions can be severe and concerns regarding human performance and safety are widespread. The motions can reduce crew abilities to perform tasks and can result in injury. This is especially of concern for military and/or rescue operations and means to reduce or mitigate against motion exposures are required to protect the occupants. In this paper the motion mitigation provided by various 'flexible' hull systems during a slam event is examined numerically, following an extended introduction describing high speed marine craft motion effects and whole body vibration and repeated shock. Introducing hull flexibility to isolate the occupants from the externally induced disturbances can either exacerbate or alleviate the problem depending on the design.

1. Introduction

The motions of marine craft can be uncomfortable, damaging and detrimental to successful and safe operation(s) on-board. Not only can physiological, biomechanical and psychological motion responses reduce crew performance and impair ship functionality (Stevens and Parsons, 2002) but the motions can cause undesirable phenomena to the craft including loss of stability, loss of steering, shipping water, slamming, cargo damage and decreased propulsion efficiency (Lewis, 1986).

In particular, occupants of high speed marine craft, which are typically 6–15 m in length and capable of speeds in excess of 30 knots, are exposed to uncomfortable, non-linear motions that can cause physical and mental fatigue (Lemmer, 1998; Myers et al., 2008) and chronic and acute injuries (Troesch and Falzarano, 1993; Peterson et al., 1997; Ensign et al., 2000; Bass et al., 2008; Coats and Stark, 2008). The motion exposures have been reported by Ensign et al. (2000) to cause annoyance, fatigue, sleepiness, discomfort, anxiety, nausea, loss of visual acuity and hand eye coordination, abdominal pain, sprains, torn ligaments, broken ankles and legs, damaged vertebrae and damage to internal organs. The most commonly cited injuries including damage to the lower back, kidneys, neck and bruises on the buttocks and inner thighs (Niekerk and Barnard, 2006). The motions of high speed marine craft have also been reported to reduce cognitive (McMorris et al., 2009) and physical ability (Myers et al., 2011).

From the literature it is clear that the motion exposures experienced by the occupants of high speed marine craft are potentially dangerous and detrimental to successful and safe operations. This is especially of concern for military and/or rescue operations and means to reduce or mitigate against motion exposures are required to protect the occupants. In this paper a detailed background describing high speed marine craft motion effects and the hazards of whole body vibration and repeated shock is presented. This is followed by a numerical analysis of the motion mitigation provided by various 'flexible' hull designs, including a suspended hull design, an elastomer coated hull and a reduced stiffness aluminium hull, during a slam event.

2. Background

2.1. High speed marine craft motion effects

2.1.1. Speed

The motions of high speed marine craft, travelling at speed in waves, are characterised by non-linear motion responses with numerous slam events and shock motions (peak magnitudes $\gg$ r.m.s) (Coats and Stark, 2008; Townsend et al., 2008). With an increase in speed from stationary to planing speeds the motions are generally found to increase in magnitude and principal frequency. At greater speeds the motions become non-linear as the hydrodynamic forces outweigh the hydrostatic forces (Rosen, 2004). At lower speeds vertical oscillatory motion within the frequency range 0.1–0.5 Hz are likely and sea sickness incidences can be expected (Lewis, 1986; British Standards Institution, 1987). Although the predominant motion responses occur in the vertical Z-axis direction, X-axis and Y-axis accelerations can also be relatively large and may contribute to the undesirable motion effects. Fig. 1 shows the magnitude and principal vibration frequencies recorded in a high speed RIB craft.

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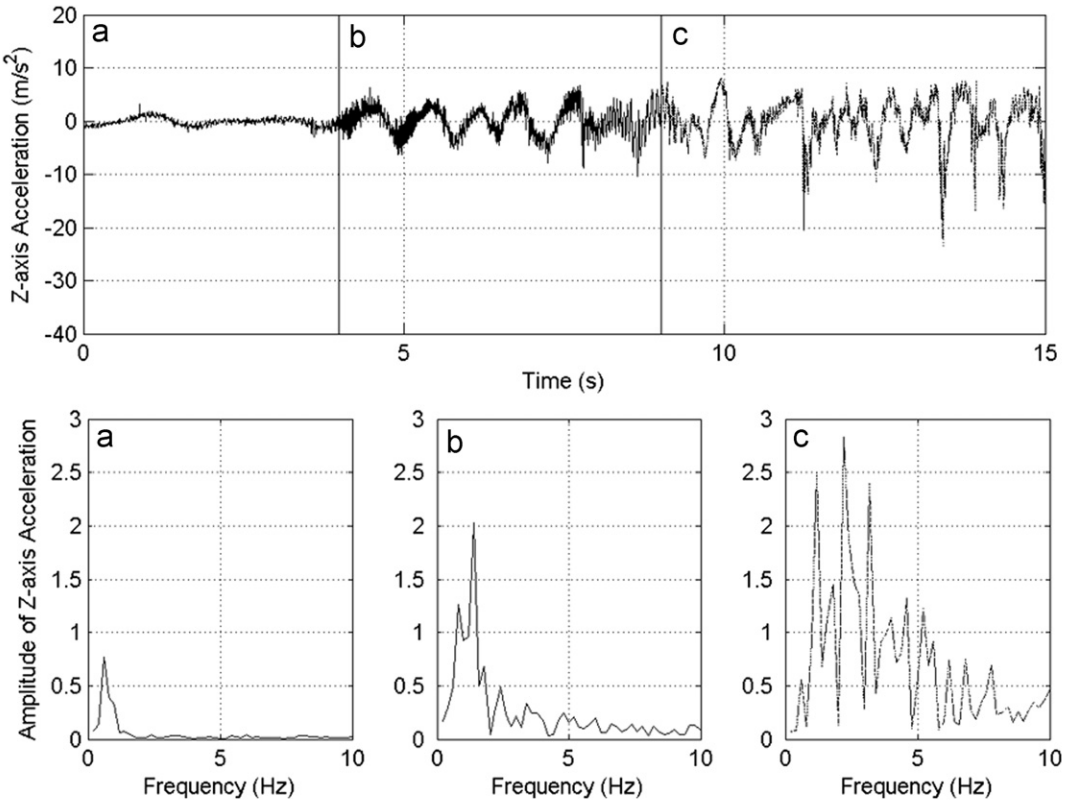

Fig. 1. Typical *Z*-axis bow acceleration time and frequency motion responses of an accelerating 7.5 m RIB from rest onto the plane ((a) at rest, (b) accelerating and (c) at approximately 30 knots). (Data sampled at 250 Hz and second order Butterworth bandpass filtered with lower and upper cutoffs of 0.1592 Hz and 100 Hz, positive *Z*-axis motion represents a downward motion) (Townsend et al., 2008).

2.1.2. Wave height

The motion responses of high speed marine craft to waves of increasing height, for given conditions, principally results in an exaggeration of the motion responses (Townsend et al., 2009). In general, with increasing wave height motion responses become non-linear and exhibit additional frequency responses to those of the wave encounter (Townsend et al., 2008). Although Grant and Wilson (2004) and Townsend et al. (2008) comment that the acceleration responses may attenuate with increasing wave height and that the relative wave profile (the relative wave height/slope compared to the craft length) may be of greater importance than the absolute wave height when discussing the motion responses of high speed marine craft.

2.1.3. Wave encounter frequency

The motion responses of high speed marine craft vary with wave encounter frequency. However, with high speed marine craft occasionally jumping over waves and missing wave encounters, the practical use of encounter frequency to describe the motions may be limited. Although, cumulative and therefore potentially dangerous effects may occur at certain encounter frequencies. For example Townsend et al. (2008) found that a model RIB towed into head waves can result in cumulative increase and decrease in motion magnitudes, due to a phasing relationship between the RIB motion and the wave excitation/encounter.

2.1.4. Deadrise

Calm water performance of high speed marine craft smaller deadrise angles are considered favourable, reducing the wetted area and frictional resistance improving planning efficiency (Savitsky and Koelbel, 1979). However, larger deadrise angles are favourable in rough water, reducing rough water pounding and improving directional stability (Savitsky and Koelbel, 1979).

Table 1
Effect of section shape on ride quality of high speed marine craft (Savitsky and Koelbel, 1979).

Section type	Shape	Commented ride quality
Vee-bottom	Concave	Hard ride and impacts
	Convex	Pounds less compared to other hulls of equal deadrise
	Straight	Creates a wet, pounding boat
Inverted vee-bottom	W-bottom	Hard pounder in significant seas, tendency to lean outboards on turns
Round Bottom	Round	Pounds badly when driven hard in rough water, poor directional stability

2.1.5. Section shape

The main section types and their commented effects on ride quality of high speed marine craft are summarised in Table 1.

2.1.6. Longitudinal centre of gravity

With a forward longitudinal centre of gravity (LCG) trim angle is reduced which at low speeds usually adversely affects sea keeping, making a craft directionally unstable, wet with a greater tendency to broach in following seas and can reduce transverse stability (Savitsky and Koelbel, 1979). However, at high speeds a forward LCG usually reduces impact accelerations (Savitsky and Koelbel, 1979).

Table 2
Summary of WBV studies (updated from Waters et al., 2007).

Source	Study group	Conclusions
Experimental studies/injury reports from shock exposures		
Burton and Sandover (1987)	Formula 1 Grand Prix drivers	Mechanical shock and vibration are strongly related to back pain
Village et al. (1989)	Miners	39% of injury claims, over a 2 year period, were for back and neck injuries
Anttonen and Niskanen (1994)	Snowmobile operators	Most operators suffered from musculoskeletal symptoms of the back that increased with exposure time
Cross and Walters (1994)	Miners	11% of all head, back and neck injury claims, over a 4 year period were attributed to vehicle jarring motions
Ensign et al. (2000)	Special boat operators	Majority of injuries reported were sprains and strains, primarily involving the lower back, knee and shoulder and are consistent with performance of heavy physical activity. The risk of injury increased with exposure
Mansfield and Marshall (2001)	Race car drivers	Reported prevalence of pain, aching or discomfort in the lumbar spine of those who rally for more than 10 days per year
Stayner (2001)	Operators of: agricultural tractors; earthmoving and construction machines, industrial (fork-lift) trucks, helicopters, overhead cranes, rail and subway vehicles, military vehicles, road vehicles and standing operators of concrete plant	Insufficient data to suggest a possible relationship between reductions in vibration magnitude and reductions in incidences of lower back disorders. Spinal injuries are probably related to the peak or magnitude of accelerations rather than the average vibration exposures
Carvalhais (2004)	Surf boat operators	High reported injury rates (over 52 %) during operator careers with lost or restricted workdays 10 times greater than that of the general Coast Guard population, most common injuries reported in the lower back, shoulders, neck, knees and ankle/foot regions
Grech et al. (2008)	Military personnel abroad a research ship	Ship motions interfere with crew sleeping patterns. Vomiting incidences was not a significant predictor of fatigue
Myers et al. (2008)	RIB crew	Suspension seats reduced the severity of impact shocks in high speed craft
McMorris et al. (2009)	Military RIB crew	Suspension seats in high speed craft may be advantageous for cognitive performance
Epidemiological studies with relevance to shock exposure		
Rosegger and Rosegger (1960)	Tractor drivers	Shaking and jolting motions may result in premature degenerative deformations of the thoracic and lumbar spinal vertebrae
Dupuis and Christ (1996)	Tractor drivers	Increased risk of spinal abnormalities after 5 years of exposure
Boshuizen et al. (1990)	Tractor drivers	Increased prevalence of back pain in workers exposed to WBV above a threshold value
Goldstein et al. (1991)	Gymnasts	Training hours and age correlated with spinal abnormalities
Bovenzi and Betta (1994)	Tractor drivers	Low back disorders are associated with vibration dose and perceived postural load (i.e., magnitude, duration, frequency)
Wilstrom et al. (1994)	Reviewed studies on WBV effects on organ systems	Lower back should be more sensitive to single shocks (than to continuous vibrations of less magnitude)
Bovenzi and Hulshof (1998)	Reviewed epidemiological studies (from 1986 to 1997)	Long term WBV exposure increases the risk of low back disorders
Brinckmann et al. (1998)	WBV in steel, mining and coal industries	Exposure to shock loading transmitted through unsprung vehicle seats results in a decrease in lumbar disc height
Tsirikos et al. (2001)	Professional jockeys	Progressive spine degeneration (due to repetitive trauma and physical stress on the spine)

2.1.7. Helm control

Operator skill (Helmsman's throttle and steering control) has been reported to have a significant effect on high speed marine craft motions (Nieuwenhuis, 2005; Coats and Stark, 2008; Townsend, 2008). Helmsman's control is therefore anticipated to be an influential factor in determining the motion exposures experienced by the crew of high speed marine craft.

2.2. Whole body vibration and repeated shock

Human tolerance to vibration primarily depends on the complex interactions of motion duration, direction, frequency, magnitude and biodynamical, psychological, physiological, pathological and intra- and inter-subject variabilities. The complex interactions and their effects on humans are not fully understood (Griffin, 1990). However, whole body vibration (WBV), especially those associated with rough vehicle rides, can damage the human body (Griffin, 1998; Waters et al., 2007). Table 2 shows a summary of WBV experimental studies, injury reports and epidemiological studies.

2.2.1. Frequency

The physical responses of the human body to vibration are commonly represented as a complex system of masses, elasticities, damping and coupling in the low frequency range defined to be below 50 Hz (NASA, 1995). The responses over specific frequency ranges are found to exhibit resonance motions which, with sufficient magnitude are anticipated to cause significant biological effects. The resonance frequency ranges associated with various body parts and the specific symptoms and their reported motion occurrences are summarised in Tables 3 and 4, respectively and Table 5 summarises the motion frequencies that are known to affect human performance. Exposure to these frequency ranges are probable during high speed marine craft transits.

2.2.2. Fatigue mechanisms

Fatigue during high speed marine craft transits reduce the physical and cognitive performance of the occupants (Myers et al., 2008, 2011; McMorris et al., 2009). This fatigue is often attributed to occupants preferring to support a proportion of their weight through their legs (Gardner et al., 2002; Cripps et al., 2003; Ullman, 2006; Myers, 2008), repeatedly absorbing shock motions which results in muscle fatigue over time. Similar to all activities requiring physical exertion. This mechanism seems to be supported by tests carried out by Myers et al. (2008) who found raised heart rates and oxygen usage during transits on board high speed marine craft.

Table 3
Resonance frequency ranges of various body parts (NASA, 1995).

Body part	Resonance motion frequency ranges (Hz)
Whole body	
Standing erect	6 and 11–12
Standing relaxed	4–5
Sitting	5–6
Transverse motion	2
Head	
Head	20–30
Head sitting	2–8
Eye ball	40–60
Eardrum	1000
Head/shoulder standing	5 and 12
Head/shoulder seated	4–5
Shoulder/head transverse	2–3
Body	
Main torso	3–5
Hip standing	4
Hip seating	2–8
Pelvic area, semi supine	8
Spinal column	8
Chest wall	60
Anterior chest	7–11
Abdominal mass	4–8
Abdominal wall	5–8
Abdominal viscera	3–3.5
Shoulder standing	4–6
Shoulder seated	4
Limb motion	3–4
Hand	1–3
Thorax	3.5
Foot seated man	>10

Table 4
Vibration frequency ranges for discomfortable symptoms (NASA, 1995).

Symptom	Vibration frequency range (Hz)
General discomfort	1–50
Motion sickness	0.1–0.63
Skeleto-muscular discomfort	3–8
Chest pain	3–9
Abdominal pains	3–10
Influence on breathing	4–8
Muscle contractions	4–9
Testicular pain	10
Urge to urinate	10–18
Influence on speech	13–20

Table 5
Vibration frequency ranges that affect human performance (NASA, 1995).

Activity	Vibration frequency range (Hz)
Speech	1–20
Tracking	1–30
Reading (texts)	1–50
Manual tracking	3–8
Head movement	6–8
Reading errors (instruments)	5.6–11.2
Depth perception	25–60
Equilibrium	30–300
Tactile sense	30–300
Hand grasping handle	200–240

2.2.3. Injury mechanisms

Various injuries and injury mechanisms are associated with WBV and repeated shock. With very few studies into the effects of repeated impacts associated with high speed marine craft motions, in spite of the reported significant risk of injury, limited data is available to identify the injury mechanisms. This is further compounded by the ethical difficulties in reproducing the dangerous motions in a laboratory.

Table 6
Typical r.m.s. acceleration magnitudes (Griffin, 1990; Sarioz and Narli, 2005).

Motion	Acceleration r.m.s (m/s^2)
Typically building vibration	0.1
Smooth rail or road vehicle motion	0.2
Typical motion limit for intellectual work	0.981
Rough rail or road vehicle motion	1.0
Off-road vehicle motion	2.0
Fast small craft acceleration limit (at forward perpendicular)	6.3765
Hazardous motion	10

Indicative scales of vibration magnitudes and typical acceleration limiting criteria have been developed as shown in Table 6. However, measures based on individual motion magnitudes, ignoring vibration frequency, duration, direction, posture and transfer points, cannot adequately describe motion severity. Frequency weighting can improve their representation of motion severity, however the results then become highly dependent on the manner in which the weightings are calculated (Griffin, 1990).

Although lower back pain, diagnosable as damage to vertebrae or intervertebral discs, is one of the most commonly reported effects of whole body vibration, no specific dose–effect relationship, relating injury to vibration exposure has been identified (Stayner, 2001). Although Bovenzi and Betta (1994) report that there is a linear relationship between posture and the prevalence of lower back pain. Typically lower back pain is associated with vibration magnitudes between 1.0 m/s^2 and 10 m/s^2, rather than exposure durations (Griffin, 1990; Stayner, 2001; Myers et al., 2008) and posture is considered a compounding factor in almost all epidemiological studies (Stayner, 2001). Posture has also been suggested to decrease the spine's ability to resist loads by a factor of up to 100 (Seidel et al., 1998) and that sitting can place additional stress on the musculature and intervertebral discs of the lumbar spine (Stayner, 2001). Mathematical modelling, replicating the mechanisms of vibration within the human body have been attempted by Pankoke et al. (1998) amongst others. However, conclusive results are difficult to obtain due to the invasive nature of any attempt to validate the results.

2.3. Motion mitigation

Performance and safety concerns regarding high speed marine craft motion exposures are widespread and with the increasing legislation, including the EU directive (European Union, 2002) and operators cost concerns, including the possibilities of insurance payout, sick pay and operational failure, there is a need to either isolate the occupants from the motion exposure or reduce the motion exposure. Current possible motion mitigation strategies include developing ride control systems (Coats and Stark, 2008), imposing speed restrictions (as vibration dose exposures are lower at lower speed Townsend et al., 2008), providing additional helmsman training (as Helmsman's throttle and steering control has a significant effect on the motion Nieuwenhuis, 2005; Coats and Stark, 2008; Townsend et al., 2008) and/or fitting suspensions seats.

Anecdotal evidence suggests that suspension seats restrict crew movement, add weight and reduce craft feedback (which could lead to boat mistreatment, damage or dangerous driving). To the authors' knowledge no ride control systems for small high speed craft are commercially available. Furthermore, speed restrictions are not considered a realistic option for military and some rescue applications. While Helmsmen's throttle/steering control is not infallible, particularly during night-time transits.

In an attempt to address the issues of WBV and repeated shock for high speed craft operations, this paper examines the motion mitigation provided by various 'flexible' hull systems during a slam event. Such systems by reducing the impact on the entire craft could reduce the structural strength requirements and therefore vessel mass and cost. In addition to reducing the need for isolation mountings for sensitive, e.g., electronic, equipment.

3. Flexible hull design

3.1. Initial appraisal

As an initial appraisal of flexible hull design, the interaction between a high speed craft hull, seat and human occupant was modelled as a forced, multiple-spring–mass–damper-system as depicted in Fig. 2.

The equations of motion describing Fig. 2 were modelled as

$$\begin{bmatrix} F_1 \\ F_2 \\ F_3 \end{bmatrix} = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_3 \end{bmatrix} + \begin{bmatrix} c_1+c_2 & -c_2 & 0 \\ -c_2 & c_2+c_3 & -c_3 \\ 0 & -c_3 & c_3 \end{bmatrix} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix} + \begin{bmatrix} k_1+k_2 & -k_2 & 0 \\ -k_2 & k_2+k_3 & -k_3 \\ 0 & -k_3 & k_3 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} \quad (1)$$

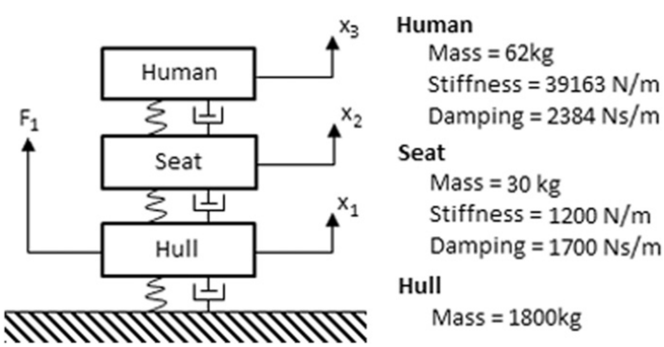

Fig. 2. Simplified hull–seat–human model.

where the subscripts 1, 2, 3 refer to the hull, seat and human components respectively and m, c and k represent the system mass, damping and restoring coefficients respectively. The human body and seat model were based on the mass, damping and stiffness coefficients presented in Coe (2011) and Coe et al. (2009) respectively. The seat representing a typical suspension seat, e.g., a STIDD suspension seat. In this study, the hull mass (m_1) was assumed constant and the damping (c_1) and stiffness (k_1) coefficients were varied. F_2 and F_3 were assumed zero.

To represent a slam event F_1 was modelled as a symmetrical, smooth impulse force;

$$F = -F_a e^{-(t-t_p)^2/2\sigma^2} \quad (2)$$

where F_a, the forcing amplitude was calculated as the hull mass multiplied by 50 m/s^2, a typical slam acceleration (Townsend, 2008). t and t_p represent the time and the time at which the peak force occurs and σ^2, a constant proportional to the impulse force duration, was assumed to be 0.0001.

The motion responses, modelled in MATLAB based on the fourth order Runge–Kutta integration scheme, are presented in Figs. 3 and 4.

For the given parameters, the simplified model shows that hull stiffness has a negligible effect on the motion response of a seated human. However, the model shows that hull damping variations can influence the motion response of a seated human, suggesting that there is scope for motion mitigation through flexible hull design.

3.2. A finite element model

A finite element model was developed to identify the motion mitigation provided by a suspended hull design, an elastomer coated hull and a reduced stiffness aluminium hull, to a freefalling drop (0.75 m) into water. The model, based on the human–seat two degree of freedom mass–spring–damper model developed by Coe et al. (2009) and a finite element model of a high speed craft hull cross section, i.e., a wedge, is shown in Fig. 5. The model was implemented in ANSYS, a commercial finite element package. The human–seat components were modelled as mass, spring and damper elements represented by MASS21 and COMBIN14 elements and the wedge was modelled using ANSYS geometric primitives and meshed with quadrilateral SHELL63 elements, assuming linear isotropic material properties. The modelled material and physical properties are summarised in Table 7.

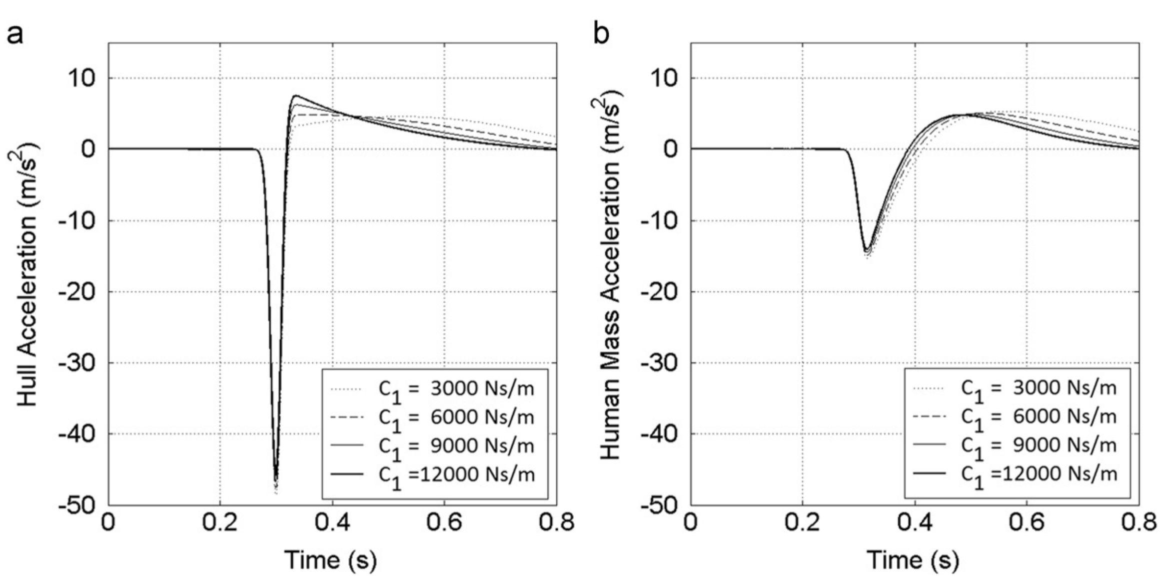

Fig. 3. Effect of hull damping variation on forced motion responses ($k_1 = 30{,}000$ N/m) ((a) hull response and (b) seated human response).

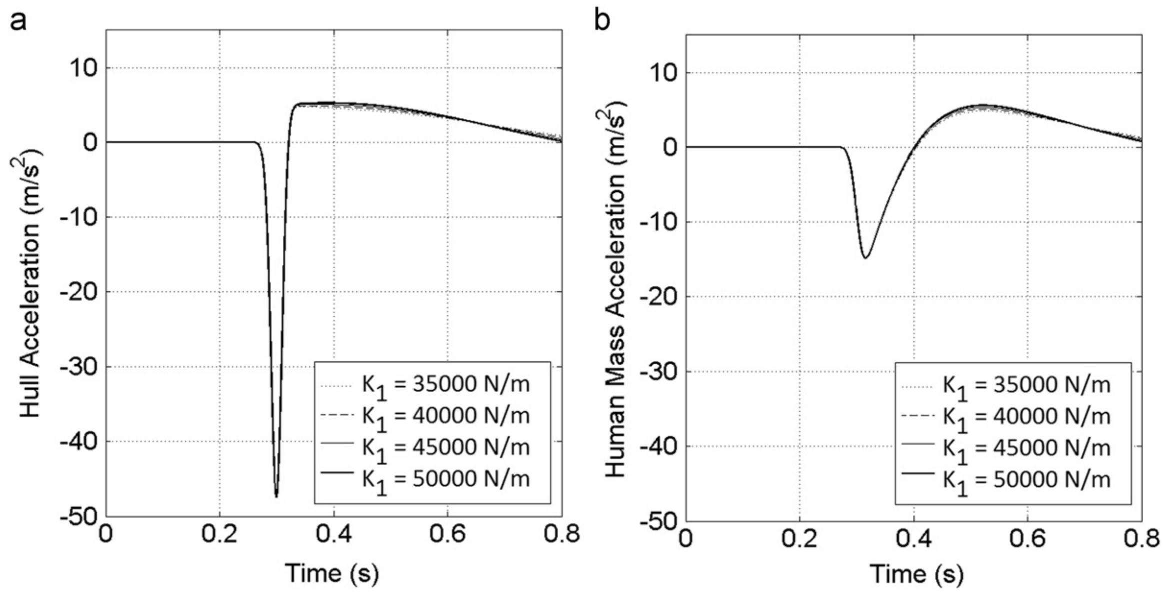

Fig. 4. Effect of hull stiffness variation on forced motion responses ($c_1 = 6000$ N s/m) ((a) hull response and (b) seated human response).

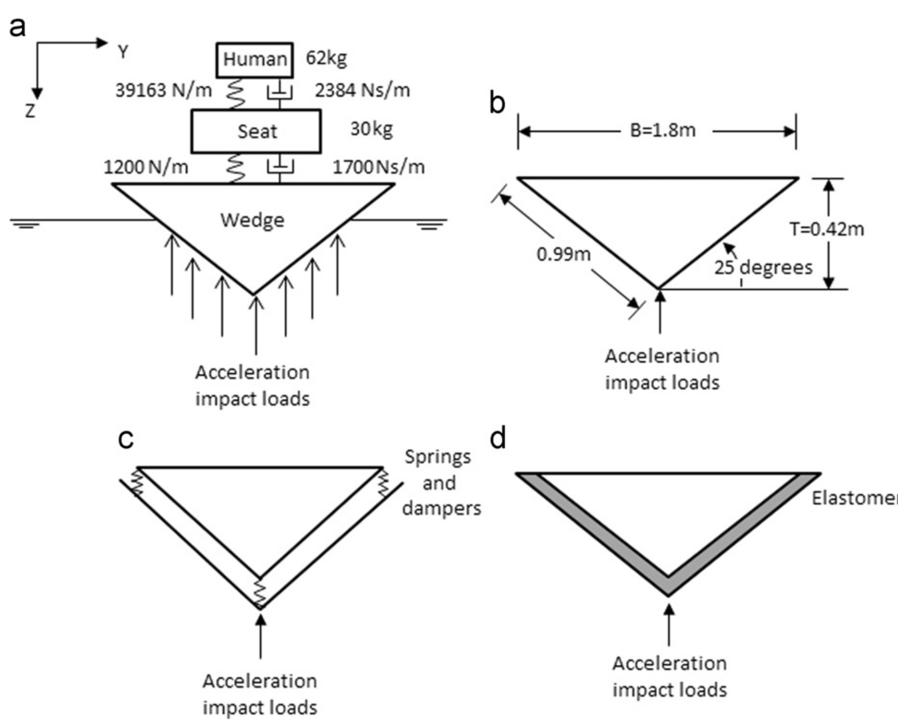

Fig. 5. A diagrammatic representation of the finite element model ((a) the human–seat–wedge model, (b) the modelled wedge dimensions ($L=B$), (c) the suspended hull design (similar in concept to that proposed by Vorus, 1999) and (d) the elastomer coated hull design).

3.2.1. Applied dynamic loads

A theoretical model was used to predict the acceleration of the wedge entering the water, based on Zarnick (1978) methods and the experimentally measured pressures for a freefalling wedge presented by Lewis et al. (2010). The initial conditions at the point of wedge entry were calculated from classical mechanics, ignoring air resistance, to provide the velocity of the wedge at the moment of water entry. From which the force on the wedge was calculated by

$$F_w = \left(V_w \times \frac{Dm_a}{Dt}\right) + \ddot{z} \times m_a + (\cos\beta \times \rho V_w^2 y_{wetted}) + (gmy_{total}l) \qquad (3)$$

where V_w represents the wedge velocity, Dm_a/Dt the rate of change of added mass with time, $\ddot{z}$ the acceleration in the vertical direction, β the wedge deadrise angle, ρ the water velocity, y_{wetted} the wetted half beam, g acceleration due to gravity, m the wedge mass, y_{total} the wedge total half beam and l the wedge length. The added mass was assumed to be

$$m_a = C_{am}\rho\tfrac{1}{2}\pi y_{wetted}^2 \qquad (4)$$

where C_{am} represents the coefficient of added mass. The wetted half beam, taking into account the deformation of water up the side of the wedge, was calculated by

$$y_{wetted} = \frac{\pi}{2} - \left(\frac{\pi}{2} - \left(\frac{\pi\beta}{180}\right) 1 - \frac{2}{\pi}y\right) \qquad (5)$$

Table 7
Modelled parameters.

Regular aluminium hull	
Stiffness (GPa)	69
Poisson ratio (v)	0.33
Density (ρ) (kg/m^3)	2700
Reduced stiffness aluminium hull	
Stiffness (GPa)	6.9
Poisson ratio (v)	0.33
Density (ρ) (kg/m^3)	2700
Suspended hull design (modelled using COMBIN14 viscous damping elements)	
Stiffness (kN/m)	32.7
Damping ratio	0.5
Elastomer coated hull (elastomer modelled as Blatz-ko foam in ANSYS)	
Youngs modulus (E) (kPa)	15.525
Density (ρ) (kg/m^3)	50

y represents the geometrically wetted half beam, calculated from the depth of immersion and the deadrise angle. The coefficient of added mass was calculated as

$$C_{am} = \frac{\pi}{4}\left(\frac{1-\frac{\pi}{2}-\left(\frac{\pi\beta}{180}\right)}{\frac{\pi}{2}}\right) \tag{6}$$

This provided a time history of the wedge motion during impact.

3.2.2. Verification

Verification of the human–seat two degree of freedom mass–spring–damper model can be found in Coe et al. (2009). To verify the finite element model of the wedge section a cantilever beam deflection comparison and a modal analysis were performed.

Cantilever beam deflection comparison: Assuming the wedge section to be an Euler–Bernoulli cantilever beam with an applied load in the vertical direction, the deflection z of the cantilever beam can be expressed as

$$z = \frac{FL^3}{3EI} \tag{7}$$

where F is the applied load at the free end, L is the length of the wedge, E is Young's modulus of the structure and I is the cross sectional second moment of area. For the modelled wedge, the second moment of area was calculated as 0.00035 m^4, yielding a tip deflection of 0.798×10^{-4} m for an applied load of 1×10^3 N.

In comparison the deflection provided by the finite element wedge model, which was constrained in all degrees of freedom at one end (i.e., $x=0$) with a point load (1×10^3 N) applied in the $-z$ direction to the other end, was found to converge to 0.668×10^{-4} m (when increasing the mesh density from 9 to 2624 elements). Although similar, providing confidence in the finite element model, there is a slight difference. The difference was attributed to the Euler–Bernoulli assumption that the beam is long and slender. Repeating the analysis for longer, equivalent, wedge models the deflection differences were found to reduce, providing further confidence in the model.

Modal analysis: As verification of the model wedge behaviour, a modal analysis was performed to identify the free vibrations of the undamped system (based on the block-Lanczos algorithm). To capture the rigid body modes, as well as higher resonant frequencies, no constraints were applied. The first 10 natural frequencies of the modelled wedge are shown in Table 8.

The presence of six modes at a nominal 0 Hz, which represent the six rigid body modes, confirmed that all parts of the model were physically connected. Furthermore, the higher modes did not display any unexpected behaviours.

Table 8
First 10 natural frequencies (modes) of the modelled wedge.

Mode no.	Frequency (Hz)
1	0
2	6.35E−05
3	3.36E−04
4	4.71E−02
5	5.60E−02
6	0.12165
7	40.666
8	42.144
9	58.131
10	58.975

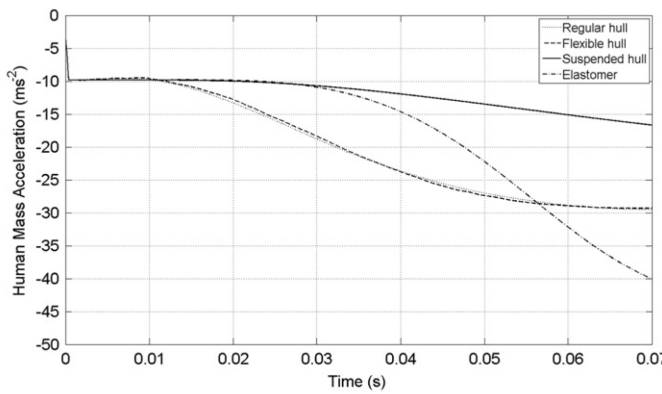

Fig. 6. Simulated human mass responses.

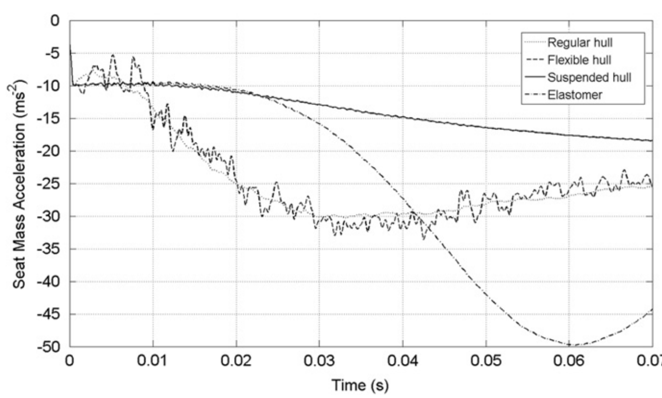

Fig. 7. Simulated seat mass responses.

4. Results and discussion

The simulated motion responses of a suspended hull design, an elastomer coated hull and a reduced stiffness aluminium hull, compared to a regular aluminium hull, to a freefalling drop of 0.75 m into water are presented in Figs. 6–8.

Considering the regular aluminium hull as the baseline against which comparisons can be drawn, it can be concluded that a reduction in hull stiffness has little effect on the response of the system. However, hull damping was found to influence the motion response. The suspended hull and the elastomer coated hull designs both demonstrated a change in the acceleration

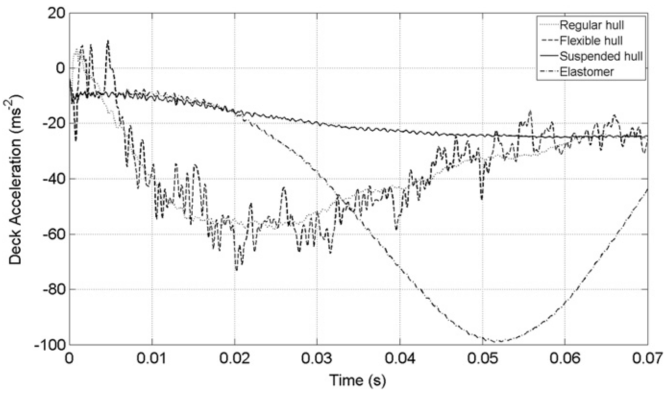

Fig. 8. Simulated deck responses.

magnitude transmitted to the human body, to the modelled slam event when compared to the regular aluminium hull response.

The elastomer design was found to initially delay the onset of the shock, followed by an amplification of the shock magnitude, yielding a peak acceleration of approximately 100 m s^{-2} at the deck, compared to approximately 60 m s^{-2} at the deck for a regular aluminium hull. That is, the modelled elastomer hull design was found to be detrimental to performance, exposing the occupants to a greater acceleration magnitude than that of a regular aluminium hull.

The motion mitigation provided by the suspended hull design was found to reduce the magnitude and onset rate of the shock. Such a system has the potential to provide vibration isolation, however in this study the practical considerations of the system were ignored. The model did not consider the limit of travel of the springs within the system and the risk of severe end stop impact. Furthermore, the hydrodynamic implications were not considered. However, the results show that theoretically there is scope for intentionally allowing hull flexibility in order to provide vibration isolation.

Interestingly, Fig. 8 also shows that high frequency deck oscillations immediately following a slam event can be expected. Furthermore, it appears that the magnitude of these oscillations increases with a reduction in hull stiffness.

With the need for employers to demonstrate that the risk to their employees from vibration is ‘as low as reasonably practicable’ (UK Statutory Instruments, 2005), the increasing legislation, including the EU directive (European Union, 2002) and operators cost concerns, including the possibilities of insurance pay-out, sick pay and operational failure, developing a ‘suspended hull design’ could alleviate some of the issues with WBV and repeated shock associated with high speed marine craft operations. However, further research is needed, for example detailed comparison studies of the competing technology, transient analysis accounting for the repeated shock motion effects and the system design.

5. Conclusions

At the outset, this paper presented an extended introduction describing high speed marine craft motion effects and whole body vibration and repeated shock. From the literature it is clear that concerns regarding human performance and safety are widespread. In an attempt to address the issues of WBV and repeated shock associated with high speed marine craft operations, this paper examined the motion mitigation provided by various ‘flexible’ hull systems during a slam event. The systems investigated including a suspended hull design, an elastomer coated hull and a reduced stiffness aluminium hull.

The results showed that a reduction in hull stiffness has little effect on the motion response. The suspended hull and the elastomer coated hull designs both demonstrated a change in the acceleration magnitude transmitted to the human body, to the modelled slam event when compared to the regular aluminium hull response. Although influencing the motion response, the modelled elastomer hull design was found to be detrimental to performance, exposing the occupants to a greater acceleration magnitude than that of a regular aluminium hull. The motion mitigation provided by the suspended hull design was found to reduce the magnitude and onset rate of the shock. Such a system has the potential to alleviate some of the issues of WBV and repeated shock associated with high speed marine craft operations, however further research is needed.

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