14

Structural design

14.1 ACV and SES structural design features

In this chapter specific aspects concerned with structural design of ACV/SES will also be discussed, as follows:

1. a brief introduction of the determination of external forces acting on hovercraft;
2. determination of safety factors during the calculation of internal stresses;
3. calculation of hull structure strength;
4. determination of plate thickness during hovercraft structural design (scantlings);
5. evaluation of ACV/SES vibration and its attenuation.

The structure of ACV/SES can be designed in a similar way to conventional ships, based on the following general procedure:

1. selection of material;
2. selection of safety factors;
3. determine external forces acting on the hull during the operation and maintenance of craft;
4. structural analysis of ACV/SES hull:
   (a) on cushion in waves;
   (b) off cushion in waves;
   (c) moored at its berth;
   (d) lifting strength;
5. optimization of scantlings.

The procedure is similar to that on most high-performance vehicles. The hull is generally constructed of aluminium alloy, GRP or marine grade steel. Further, the main design considerations after selection of the main structural dimensions are motion accelerations, impact loading due to waves and landing, fatigue of highly stressed areas and stresses induced by vibration.

Both ACV/SES have a large volume compared with their displacement and so have a large area of shell plate, i.e. high $\bar{S}$, where $\bar{S} = S/(W_v)^{0.66}$, where S represents the area of shell plates in m^2 and W_v the volumetric displacement of the craft in m^3.

SES

An SES hull is similar to a catamaran with wider beam and thinner hulls. Thus, at the same displacement it has a larger area of shell plate and a heavier structure weight compared with that of a conventional monohull.

Hull weight is related closely to the principal dimensions of an SES and thus also with the cushion pressure/length ratio p_c/l_c. Due to support of the major part of the craft weight by the cushion, an SES structure is less highly loaded and can therefore be designed as a lighter structure per unit area. Materials which lend themselves to SES structures are GRP and welded aluminium alloy due to their low density. It has not been found necessary to use aircraft design techniques to minimize structure weight, for example using carbon fibre reinforcement to GRP, or using high-tensile-strength grades of aluminium. A satisfactory structure can be fabricated from weldable marine aluminium alloys, or glass reinforced polyester or vinylester plastic.

Since ACV/SES operate in a salt water environment, the structural material is required to have good corrosion resistance. In this respect GRP or marine grade aluminium alloy has advantages. If steel is used, corrosion protection has to be applied and regularly maintained.

Steel is less expensive to fabricate than the other materials but incurs a considerable weight penalty. Some use of HT steels can mitigate this. To date there are very few SES, e.g. the 719-II, which have successfully applied a steel hull structure. Steel hull structures may find useful application for larger SES (above 100 m in length) where its higher modulus may allow improved optimization, particularly if the normal approach for Navy frigates is used where the lower-loaded superstructure is constructed in aluminium.

ACV

Since passenger hovercraft have to have low p_c/l_c in order to obtain good take-off performance, small wave-making resistance, added wave drag and roomy passenger cabin, in general designers adopt smaller p_c (i.e. low-density craft) for passenger ACVs and take larger p_c (high-density craft) for military landing craft, which are required to have small overall dimensions for entering into the landing ship's stern dock area and where operational economics are not so sensitive to installed power.

Due to the low density requirements in order to achieve low p_c, designers were earlier obliged to use material with high specific strength, e.g. riveted aluminium alloy and sandwich panel construction, in a similar manner to aircraft. High construction costs follow this approach. Since the early 1980s, as skirt technology has allowed higher p_c values to be adopted, GRP construction has been used in ACVs as well as SES, and welded aluminium structures similar to catamarans have been successfully designed. The normal approach is now to use these lower-cost structural design.

It is useful to note that when designing with high-strength materials, as for aircraft, it is often the case that stiffness rather than strength is the controlling design criterion (cabin floor panels for example). The transition to lower-strength materials therefore also leads to a change to strength governing stiffness as the main structural design criterion. Both criteria should nevertheless be checked during design!

The welding of aluminium structures can be a serious problem for the aluminium plates applied to the structure, particularly for upward and vertical welding. In certain areas manufacturers and designers have been obliged until recently to continue to use riveted or adhesive jointed structures based on aviation technology. So long as this is minimized, the effect on total cost need not be significant.

Summary

The current state of development of hovercraft structures may be summarized as follows:

1. Welded aluminium structures are now widely used on hovercraft (e.g. AP.1-88) to replace the riveted and adhesive structure based on aviation technology (SR.N series), leading to decreased construction cost. Marine grade aluminium alloy is generally of lower strength than aircraft grades. The welds are relatively low strength, since no work hardening can be applied. Shell plates of SES or ACV structures are therefore thicker than a riveted design, resulting in a higher structure weight.
2. With respect to small ACV/SES, thin plates would normally be sufficient for the structural loading requirements. These plates are difficult to weld in some cases and this together with the stiffness requirements for the structure may oblige the designer to replace thin plating with over-thick plates, or use riveted, glued or fibre-reinforced plastic structures. As an alternative, in order to save labour, weight and overall dimensions it is possible to consider alternative structural arrangements to the traditional buoyancy tank approach, or to replace part of the rigid structure with an inflatable structure.
3. It is possible to construct the structure with welded aluminium or GRP. The latter is easy to maintain or repair with simpler construction technology, but it has lower stiffness than aluminium, which prevents the material being used to construct larger craft (above 50 t approx). In the case of large medium-speed SES, marine steel can also now be used as the material in order to reduce the construction cost. The Chinese SES 719-II is an example.
4. With respect to smaller hovercraft, the most significant problem is minimizing structural weight based on extreme external loads such as wave 'slamming' at high speed. In other respects, small hovercraft are dimensioned from stiffness requirements rather than strength.
5. Due to the low structural density, high installed power and high-speed rotating machinery with wide distribution of exciting frequencies (engines, fans, propellers and transmission gearboxes), vibration problems can become very important. Vibration and noise analysis therefore has to be included in the overall design of the craft, to minimize the vibration, and apply appropriate vibration isolation design.

14.2 External forces on hull – introduction to the strength calculation of craft

Determination of external forces

Structural design should start with assessment of the quasi-static trim and equilibrium of forces at maximum speed. Designers normally estimate the external loads while the ACV/SES runs in the required sea state at maximum cushion-borne speed. From the point of view of strength calculation, the estimation of such loads is critical. The distribution of wave-impacting load in longitudinal and transverse directions can be determined by empirical methods, using data from model tests or previously constructed craft and thus the dynamic bending moment acting on the craft can also be determined in a similar way to the method used for the strength calculation of planing hulls [90].

In order to determine external loads accurately, a great deal of model and full-scale ship experiments, similar to those on planing hulls, have to be carried out. So far such data are limited; the main reference to which one can refer for information is the calculation method for determining external loads in the rules for safety requirements of civil hovercraft originally issued in 1962 by the British Aviation Authority [4,115,] which was based upon the calculation method for external loads of sea-planes and the unskirted ACV model SR.N1. Therefore correction for the impact absorption of skirts and the air cushion should be incorporated (see below).

In a similar way to a planing hull, the external loads acting on an ACV or SES are mainly the vertical impact acceleration acting on the hull running in waves and the slamming pressure of waves acting on the bottom frame of the craft. Superposing the dynamic bending moment caused by these loads upon the static bending moment, we can calculate the overall bending moment and shear, according to which the required local strength and overall longitudinal, transverse and torsion strength can be calculated.

Determination of design inertial loads

According to [115], a factor may be defined, η_w, where η_w may be calculated as the relative vertical acceleration, that is, the vertical acceleration divided by gravitational acceleration at point A located at a distance of l_2 from the CG of the craft:

$$\eta_w = (p_w/gW)\,(1 + l_1\,l_2/r_y^2) \tag{14.1}$$

where p_w is the impacting force acting on the hull by the waves (N), W the craft weight (kg), l_1 the distance from the impacting force to the CG of the craft (m), J_y the transverse moment of inertia through the CG of the craft (kg m^2) and r_y the radius of inertia of the hovercraft (m), $r_y = (J_y/W)^{0.5}$. The wave-impacting force can also be written as

$$(p_w/gW) = (K_1\,V_{zk}\,V_s)/[W^{0.333}\,(1 + r_y^{\,2})^{0.666}] \tag{14.2}$$

where K_1 is the coefficient due to the wave-impacting force, and can be written as (Fig. 14.1) [4]

$$K_1 = \begin{cases} 1 + l_1/l_2 & l_1 > 0, \text{ impact occurring forward of LCG} \\ 1 & l_1 < 0, \text{ impact occurring aft of LCG} \end{cases}$$

and V_{zk} is the relative vertical velocity due to the craft and wave motion, at the section on which the impact force acts (m/s),

$$V_{zk} = \pi \ V_w \ \zeta_a / L_w + V_z \tag{14.3}$$

where L_w is the wavelength (m), V_z vertical velocity of hull section on which impact force acts (m/s). In the case of lack of experimental data, $V_z \approx 0.6$ m/s. V_s is the craft speed (m/s) and ζ_a the wave height, which can be written as

$$\left.\begin{array}{ll} \zeta_a = 0.1 L_w & \text{when } L_w < 38 \text{ m} \\ \zeta_a = 0.10 L_w^{0.5} & \text{when } L_w > 38 \text{ m} \end{array}\right\} \tag{14.4}$$

V_w is the wave speed (m/s):

$$V_w = 1.25 L_w^{0.5} \tag{14.5}$$

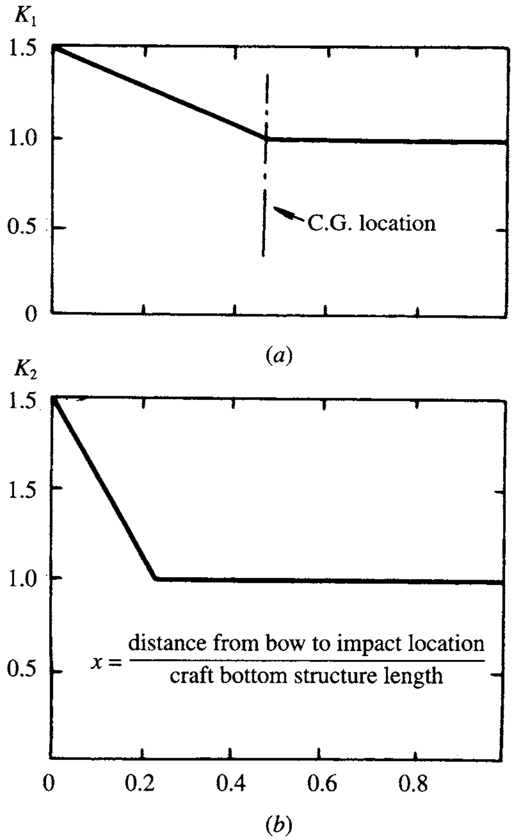

Fig. 14.1 Wave impact loading and pressure coefficients: (a) loading coefficient; (b) pressure coefficient.

J_y is the ACV/SES moment of inertia which can expressed in simplified form as

$$J_y = (W/12)\,(L^2 + 4z_g^2) \tag{14.6}$$

where $L^2 >> 4\,z_g^2$, then expressions can be written as

$$\left.\begin{aligned} J_y &\approx W\,L^2/12 \\ r_y &\approx 0.3L \end{aligned}\right\} \tag{14.7}$$

where L is the craft length (m) and z_g the vertical height of the CG of the craft over the base-line (m).

Determination of wave impact pressure [4][52]

When hovercraft are running at high speed in waves, the impacting pressure due to waves at the acting centre of the waves (e.g. the maximum hydrodynamic pressure) can be written as

$$p_{max} \approx K_3 K_2 \rho_w\ V_s \tag{14.8}$$

where K is an empirical coefficient, take $K_3 = 4.6$–4.9, ρ_w is the water density (Ns^2/m^4) and

$$K_2 = \left.\begin{matrix} 2 - x/(0.22L) & \text{when } x \leqslant 0.22L \\ 1 & \text{when } x \geqslant 0.22L \end{matrix}\right\} \tag{14.9}$$

The data can also be obtained in Fig. 14.1(b).

Reference 4 discussed the linear relation between the wave-impacting force acting on the bow and the craft speed, as in expression (14.8). It was validated by the test results on the ACV SKMR-l (without skirt) in 1963. With respect to the bow slamming condition, if one uses $K = 2.0$, then we have

$$P_{max} \cong V_s \tag{14.10}$$

where V_s is the craft speed (knots) and P_{max} the maximum slamming pressure, lb/in^2 ($\times$ 6894.76 in Pa).

Expression (14.10) is the well-known 1lb–1 knot rule, in other words, the slamming pressure acting on the bottom plate of the bow increases with the speed in the proportion to 1 lb/in^2 per knot of speed. This formula has been used for the design of many early hovercraft.

The calculation of pressures for shell plates of US JEFF(A), JEFF(B) landing craft ACVs and SES-100A and SES-100B are shown in Fig. 14.2. The design conditions for the US ACV are the craft operated at speed of 50 knots in sea state 5, while for the SES it is 60 knots, sea state 5.

British designers took a different point of view on ACV structural design in the 1970s. They proposed that the air cushion and skirt attenuated slamming and that the design slamming pressure could be reduced by a factor of 50%. For the SR.N4 hovercraft ferry, maximum slamming pressure for design of the bow structure was taken as 22–23 lb/in^2 at a speed of 35–40 knots and in sea state 5. It was later found that the bow bottom plates of SR.N4 had been damaged in sea state 5 conditions and as a result, BHC's experts offered a revised attenuation coefficient, i.e. 30% reduction of hydrodynamic pressure, as shown in Fig. 14.3. Table 14.1 shows the calculated conditions for some ACVs during the calculation of hull bow strength.

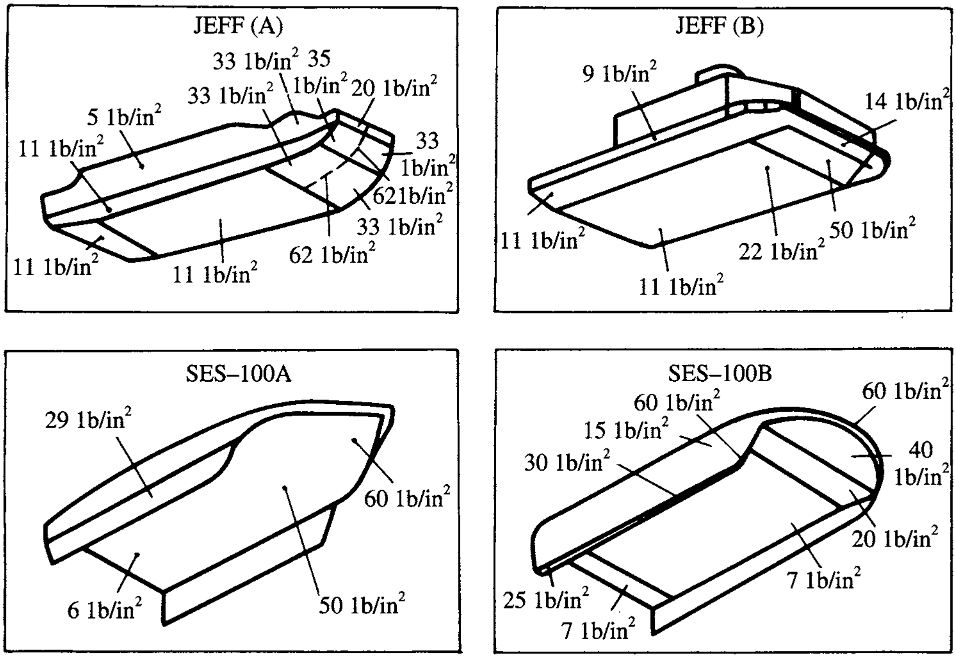

Fig. 14.2 Calculated hydrodynamic pressure on plates of US SES craft due to wave impact.

Having obtained the external load, i.e. the inertial load factor and slamming pressure acting on the bottom plates and frames of the craft, one can check the strength (including the fatigue strength) of structure and mountings for various machines, equipment and instruments as follows:

1. The overall strength of the craft in the case where the wave exerts an impact force at the CG or other longitudinal portions of the craft.
2. The overall longitudinal strength, torsion and transverse strength of craft under the action of wave slamming at one side of the bow or other portion of the craft. Hovercraft and SES structures normally comprise a relatively thin buoyancy tank

Table 14.1 Calculated conditions adopted in calculating the strength of hull of ACV [4]

Features	Unit	SKMR-1 (no skirt)	SR.N2	SR.N4	SR.N4	SR.N5	SR.N6	JEFF(B)
Weight	t	20.40	25.90	168.00	168.00	6.80	9.55	147.00
Craft speed	knots	50.00	55.00	70.00	60.00	60.00	50.00	50.00
Wave height	m	1.50	0.90	0.76	0.37	0.61	0.61	1.98
Maximum slamming pressure	MPa							
Bow		0.415	0.103	0.155	0.155	0.155	0.124	0.345
Stern		0.206	0.041	0.033	0.033	0.033	0.083	0.152
Inertial load factor	g							
Bow		2.00	0.83	0.83	0.45	1.00	1.00	1.00
Midship		1.00	1.80	1.80	0.80	2.00	2.00	2.00
Stern		1.00	0.75	0.75	0.40	0.75	0.75	0.75

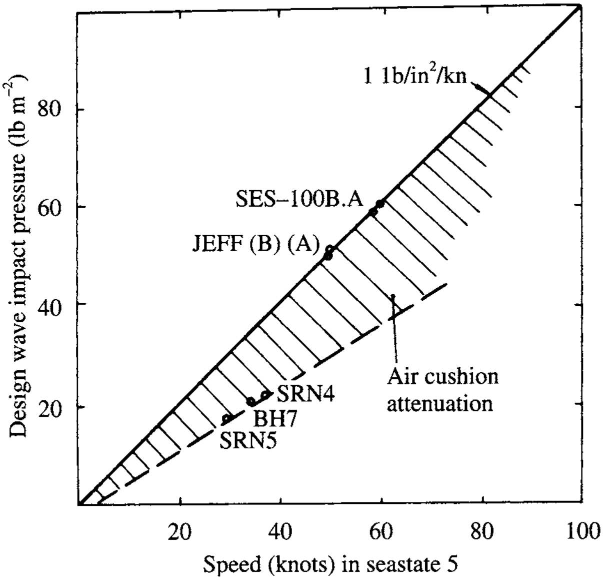

Fig. 14.3 Cushion attenuation coefficient vs craft speed.

above their wet deck, and the main transverse strength is provided by a number of watertight bulkheads and transverse web frames.

On small SES, the overall longitudinal strength is provided by the two sidewalls, while the transverse strength, being controlled by the transverse bulkheads, may require more than that needed to provide damage stability. This may lead to cabin spaces being divided. A large proportion of SES transverse bulkheads are above the upper deck, which moves the centre of area upwards relative to the sidewalls. This results in a further increase in structural scantlings. The effect is inbuilt into the catamaran cross-section of an SES, enhanced by the deeper sidewalls of an SES.

3. Owing to the craft operating in waves at a high speed, hydrodynamic impacts, clearing vertical obstacles, landing and launching and emergency landing etc, during the design of mountings for various instruments and equipment the acceleration acting on the equipment in three directions X,Y, Z has to be considered carefully.

14.3 Brief introduction to the structural calculation used in MARIC[102][103]

Parameters and conditions

During the analysis of the local and overall strength of the craft structure, the following conditions are considered at MARIC:

1. ACV lifted by crane;
2. ACV static on the ground (rigid surface, normally three-point loading);
3. ACV/SES operating cushion borne over ground (ACV) or water surface (ACV and SES);
4. ACV/SES operating on cushion in waves at high speed, including wave slamming, when the following slamming conditions have to be considered:
   (a) wave slamming at the CG of craft;
   (b) slamming at bow/stern instantaneously;
   (c) slamming at bow only.
5. ACV/SES on hull-borne operations:
   (a) in sagging condition;
   (b) in hogging condition.

The conditions listed under (4) and (5) are similar to those applied to conventional ships. The differences between them are the dynamic bending moment acting on the hull caused by the wave slamming of the craft and the hydrodynamic impacting force acting on the shell plates. This requires a different method of calculation for the overall and local strength of craft, so we will introduce briefly the procedure and conditions for strength inspection of hovercraft.

ACV/SES operating on cushion at high speed in waves – wave slamming at the CG or bow/stern instantaneously

In this case, the craft can be considered as not heaving or pitching, therefore the equilibrium conditions for the vertical force are as follows (taking an ACV as the example):

$$W + F_i = A_c p_c + p_w$$

where W is the craft weight, F_i the inertia force of the craft, $A_c p_c$ the total cushion lift and p_w the impacting force of waves, from equation (14.2) and Fig. 14.1. In the above equation, the inertia force can be taken as the weight at the different longitudinal position (ordinate) times the vertical acceleration acting on this position, i.e. the inertial load times the gravitational acceleration g. In general, the craft length can be divided into 20 ordinates for calculation. In the case where the wave slamming is acting on the CG, the impacting acceleration will be constant along the longitudinal axis and without pitching, then we have

$$p_w = \eta_w W \tag{14.11}$$

The impacting length can be taken as (0.145–0.16) l_c, symmetric about the craft's CG. In the case where wave slamming impacts on the craft at the bow/stern instantaneously, then

$$p_{wb} + p_{ws} = p_w = \eta_w W \tag{14.12}$$

where p_{wb} is the wave impacting force at bow (N), p_{ws} the wave impacting force at stern (N), W the craft weight (N), l_s the impacting length at stern (m), l_f the length of front body of craft before the CG (m), l_a the length of rear body of craft before the CG (m) and l_b the impacting length at the bow (m):

$$l_s = (2.34\, l_c\, l_f)/(1.2\, l_f + l_a)$$
$$l_b = (1.95\, l_c\, l_a)/(1.2\, l_f + l_a)$$

For impact at the bow/stern instantaneously, the craft is not pitching, the resultant of both bow/stern impacting force acts on the CG. The equilibrium condition for this force can be written as

$$p_w + \int_{lc} p_c\, B_c\, dl = \sum_{i=1}^{n} (1 + \eta_w) W_i \tag{14.13}$$

where p_c is the cushion pressure (Pa), l_c, B_c the cushion length and beam respectively (m) and W_i the craft weight sharing on ith space (N) and the craft is divided into n spaces along its length. The shear and longitudinal bending moment can be obtained according to this equation,.

Craft operating on cushion at high speed in waves – hull strength in the case of wave slamming at bow

In the case where slamming occurs at the bow, pitching motion will occur and the vertical acceleration is not uniformly distributed along the longitudinal axis; the law of distribution can be calculated according to equation (14.1) and Fig. 14.1, being linearly distributed as follows:

$$p_w + \int_{lc} p_c\, B_c\, dx = \int_{lc} (1 + \eta_x)\, W(x)\, dx \tag{14.14}$$

$$L_b = 0.1 L_c$$

According to this equation and applying the gravitational force, cushion force, inertia force and hydrodynamic impacting force on each longitudinal space, the longitudinal bending moment and thus strength inspection can be obtained. Meanwhile, the local strength analysis also can be carried out based on the wave-impacting pressure. With respect to the inertial loads (η_{wi}) acting on the mechanical and electrical equipment as well as their mountings at various positions along the longitudinal axis can be obtained according to this equation and Table 14.2. During craft landing, the force acting on the landing pads can also be obtained from Table 14.2; this table was obtained from tests and statistical analysis.

14.4 Calculation methods for strength in the former Soviet Union

Analysis of structures is specified by these methods for adequate reserve while floating or on cushion in the design wave conditions, while moored at its berth and while being lifted for maintenance. The analysis methods have been found useful and realistic and can be recommended where the craft type and operational mission are applicable.

Table 14.2 Maximum acceleration acting on hovercraft engines and equipment and the forces acting on landing pads of hovercraft [104]

1	Maximum acceleration acting on engines and equipment	Upward		$3g$
		Down		$4g$
		Forward		$6g$
		Backward		$3g$
		Lateral		$5g$
		Resultant		$6g$
2	Force acting on landing pads	SR.N2, middle pads,	vertical	1.0 × craft weight (W)
			lateral	$0.5W$
		SR.N5, all pads,	vertical	$0.5–0.6W$
			horizontal	$0.17W$
		SR.N4, fore pads,	vertical	$0.5W$
			horizontal	$0.25W$
		SR.N4, other pads,	vertical	$0.4W$
			horizontal	$0.2W$

Useful range of the calculation

This calculation is suitable for craft operating on waterways in (O), (P) and (L) classes. The craft can be operated cushion-borne and hull-borne as passenger, auxiliary transport, or cargo ACV/SES. The classifications O, P and L are for river boats as stipulated by the Soviet government, which corresponds to the A, B and C classes of boats operating in China, on rivers and in estuary waters. The calculations of wave height h (the 1% highest waves) are equal to:

For craft operating on O class waterways $h_w = 2.0$ m
For craft operating on P class waterways $h_w = 1.2$ m
For craft operating on L class waterways $h_w = 0.6$ m

The stiffness of hull and relative speed F_r of such hovercraft should satisfy the following conditions:

$$\left.\begin{array}{l} EI/(DL) > 1.3 \\ V/(gL)^{0.5} > 2.0 \end{array}\right\} \tag{14.15}$$

where E is the elastic modulus on the normal direction (tf/m^2), I the section moment of inertia of the hull structure (m^4) – this only includes the section moment of inertia of the main hull structure in the case of no strong superstructure, otherwise it must include the section of inertia of the superstructure. D is the displacement of craft (t) and L the craft length (m).

The ratio of principal dimensions of an SES has to satisfy the following conditions:

$$\left.\begin{array}{l} L/H < 20 \\ L/B = 3\text{–}6 \\ H/H_{sw} = 2\text{–}3 \end{array}\right\} \tag{14.16}$$

where H is the depth of the upper deck (m) and H_{sw} the depth of sidewalls (m).

Design loads for craft structure, overall bending and torsion

The loads acting on the craft structure during the calculation of overall bending and torsion can be determined using the maximum inertial load coefficient measured at the craft's CG. The inertial load coefficient operating in waves can be obtained from prototype or experimental results of models in various operation modes and various modes of overall deformations. The loads acting on locations other than the CG can be determined as follows:

$$\eta = \{1 + \mu_1 [(x_1 - x_g)(x - x_g)/\rho_1^2 + (y_1 y)/\rho_2^2] + \mu_2 [(x_2 - x_g)(x - x_g)/\rho_1^2 + (y^2 y)/\rho_2^2]\}\eta_g \tag{14.17}$$

where μ_1, μ_2 are coefficients, determined from Table 14.3, x_1, x_2, y_1, y_2 are the coordinates of external force as shown in Fig. 14.4, x_g the longitudinal ordinate of the CG of the craft (m), ρ_1 the radius of inertia of the hull weight about the transverse axis through the CG (m), ρ_2 the radius of inertia of the hull width about the longitudinal axis through the CG (m), η_g the inertial load coefficient acting at the CG of the craft in the case of lack of information during the preliminary design phase. The inertial load coefficient for calculating longitudinal strength can be determined as follows (for cushion-borne operation):

$$\eta_g = 1 + (0.085\, h^{0.5} + 0.04 V/D^{0.333}) \tag{14.18}$$

The external force can be written as

$$\left.\begin{aligned} P_1 &= \mu_1\, D\, \eta_g \\ P_2 &= \mu_2\, D\, \eta_g \end{aligned}\right\} \tag{14.19}$$

Based on these inertial load coefficients, the longitudinal and transverse bending moments can be obtained in a similar way. The location and area of action of the hydrodynamic impacting force during slamming at the CG or bow/stern can be obtained from Fig. 14.4 and Table 14.3.

The torsion moment M_t can be determined by integrating the torsion moment intensity, which is the algebraic sum of the moment intensity m_1, m_2 and distribution moment m_3, induced by the supporting force P_1, P_2 and the mass inertia of the craft about the longitudinal axis respectively, i.e.

$$\left.\begin{aligned} m_1 &= \mu_1\, D\, \eta_g\, y_1/l_1 \\ m_2 &= \mu_2\, D\, \eta_g\, y_2/l_2 \\ m_3 &= -W(x)\, (\mu_1\, y_1 + \mu_2\, y_2) \end{aligned}\right\} \tag{14.20}$$

The distribution of moment intensity m_1, m_2 along the craft length can be determined as in Fig. 14.4 and Table 14.3. Moment m_3 distributes along the whole length of the craft. $W(x)$ represents the distribution of craft weight along the longitudinal axis.

Overall bending moment acting on the midship section

In preliminary design, the overall bending moment acting on the midship section M_o can be determined as follows.

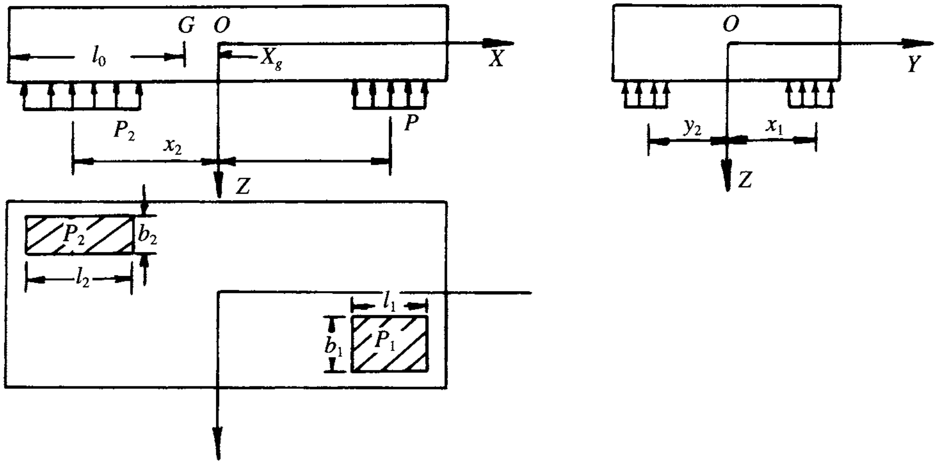

Fig. 14.4 Some parameters for determining overall bending moment and torsion load of SES.

Table 14.3 Some parameters for determination of overall bending and torsion moment acting on a structure

Characteristic value	Cushion-borne operation in waves				Hull-borne operation in waves			
	Longitudinal bending		Transverse bending	Torsion	Longitudinal bending		Transverse bending	Torsion
	Sag	Hog	Sag		Sag	Hog	Sag	
$L1$	$0.2L$	$0.4L$	$2\,l_o$	$0.2L$	$0.2\,L$	$0.4L$	$2\,l_o$	$0.2L$
$L2$	$2\,l_o$	$2\,l_o$	$2\,l_o$	$0.2\,l_o$	$0.2\,L$	0	$2\,l_o$	$0.2\,L$
$b1$	B	B	ε_1	ε_2	B	B	ε_1	ε_2
$b2$	B	B	B	B	B	0	ε_1	ε_1
$x1$	$0.4\,L$	x_g	x_g	$0.4L$	$0.4L$	x_g	x_g	$0.4L$
$x2$	x_g	x_g	x_g	x_g	$-0.4\,L$	0	x_g	$-0.4\,L$
$y1$	0	0	ε_2	ε_2	0	0	ε_2	ε_2
$y2$	0	0	0	0	0	0	$-\varepsilon_2$	$-\varepsilon_2$
μ_1	$(\eta_g - 1)/\eta_g$	$(\eta_g - 1)/\eta_g$	$(\eta_g - 1)/\eta_g$	$(\eta_g - 1)/\eta_g$	2/3	1	1/2	2/3
μ_2	$1/\eta_g$	$1/\eta_g$	$1/\eta_g$	$1/\eta_g$	1/3	0	1/2	1/3

Note:
For ACV: $\varepsilon_1 = 0.2B$, $\varepsilon_2 = 0.4B$.
For SES: $\varepsilon_1 = B_{sw}$, $\varepsilon_2 = 0.5\,(B - B_{sw})$.
B_{sw} = width of sidewalls at the bow, and
B = width of midship section at design water-line.

ACV and SES cushion-borne operation

$$M_o = [K_s \pm 0.5\,(0.15 \pm K_s)\,(\eta_g - 1)]\,D\,L \tag{14.21}$$

where K_s is the coefficient for longitudinal bending moment in calm water, (+) represents the hogging mode, (−) represents sagging mode, and η_g the inertial load coefficient, which can be determined by equation (14.18), or using prototype and model test results.

ACV hull-borne operation

$$M_o = \pm\, 0.5\,(0.15 \pm K_s)\,\eta_g\, D\, L \tag{14.22}$$

SES hull-borne operation

$$M_o = [K_s \pm 0.5\,(0.15 \pm K_s)\,(\eta_g - D_{sw}/D)]\, D\, L \pm 5.1 B_{sw}\,(L/10)^2\, h \tag{14.23}$$

where D_{sw} is the displacement provided by the sidewalls and h the wave height.

The maximum shear can be written as $N_o = 4\ M_o/L$. The overall bending moment and shear for every section of a craft can then be determined as in Fig. 14.5.

Determination of transverse bending moment of ACV/SES in preliminary design

This can be determined as follows.

ACV/SES cushion-borne operation

$$M_o' = [K_s' - 0.5\,(0.15 - K_s')(\eta_g' - 1)]\, DB \tag{14.24}$$

ACV hull-borne operation

$$M_o' = -\, 0.5\,(0.15 - K_s')\, DB\, \eta_g' \tag{14.25}$$

SES hull-borne operation

$$M_o' = -\, 0.5\,[0.25 - 0.5\,(B_{sw}/B - K_s')]\, DB\, \eta_g' \tag{14.26}$$

where K_s' is the coefficient for transverse bending moment in calm water,

$$K_s' = M_{os}'/DB$$

M_{os}' the transverse bending moment in calm water (tm), B the width of midship section at designing water-line (m) and η_g' the inertial load coefficient, determined by prototype or model test. The maximum shear can be written as

$$N_o' = 4M_o'/B \tag{14.27}$$

Calculation for local loading

The local load acting on the bottom and sidewalls of an ACV/SES can be determined according to the following conditions:

1. air cushion pressure (in the case where water does not contact the structure directly);
2. hull slamming ;
3. reaction force of supports.

These forces can be calculated as follows.

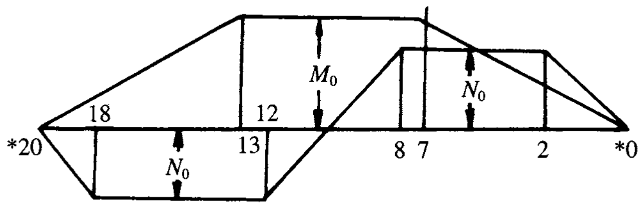

Fig. 14.5 Distribution of overall bending moment and shear forces at different craft stations.

Air cushion pressure

In the case where the hull does not contact the water surface, the distribution of pressure under the bottom along the craft length can be expressed as in Fig. 14.6 and along the transverse direction can be written as a uniform distribution:

$$\left.\begin{aligned} P_1 &= 2D\,\eta_g/S_c \\ P_2 &= D\,\eta_g/S_c \end{aligned}\right\} \tag{14.28}$$

where S_c is the cushion area (m^2). The design cushion pressure should be at least 30% greater than the cushion pressure supplied by the lift fan in the case of no air leakage.

Distribution of wave impact force along the craft length

During slamming on the craft bottom, the distribution of hydrodynamic pressure along the craft length can be determined as in Fig. 14.7, but it is uniformly distributed along the transverse direction. The impact force acting on section 0, 10, 20 (bow, midships and stern) can be taken as

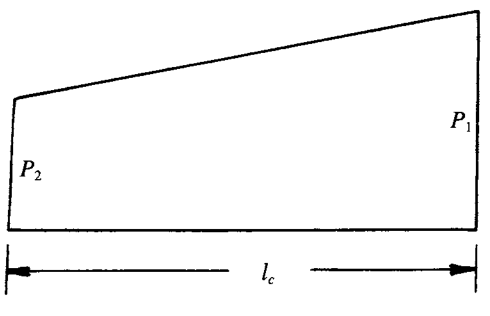

Fig 14.6 Distribution of cushion pressure in longitudinal direction due to slamming in waves.

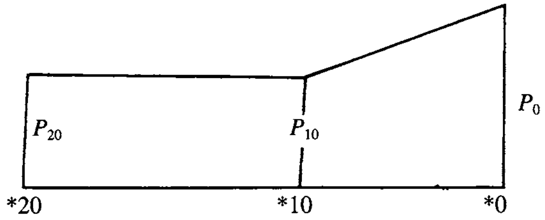

Fig. 14.7 Pressure distribution in longitudinal direction due to slamming of craft bottom in waves.

$$P_0 = KD\eta_g/(0.3\ LB)$$
$$P_{10} = KD\eta_g/(0.4\ LB) \qquad (14.29)$$
$$P_{20} = KD\eta_g/(0.4\ LB)$$

where K is the coefficient due to non-uniformity, and can be written as

K= 1 for the calculation of frames
K= 3 for the calculation of stiffness and frames between station 0 and 10
K= 1.25 for the calculation of stiffness and frames at station 20

Hydrostatic pressure acting on the bottom P_b and the sidewalls P_{sw}

$$P_b = T + h/2 - h_{sw} \qquad (14.30)$$
$$P_{sw} = T + h/2 - z$$

where h is the design wave height (m), z the vertical height from the base-line to the design location of the side plates (m), T the draft of craft in hull-borne operation, which can be measured from the lower edge of the bottom plates of the sidewall (or from the bottom in the case of no sidewall) to design water-line (m), P_b the hydrostatic pressure acting on the bottom, water head in metres (1m H_2O = 9.8 kPa) and h_{sw} the sidewall depth (m).

Cushion pressure

This can be calculated as a uniform distribution along the vertical direction and the distribution along the longitudinal axis can be calculated as shown in Fig. 14.6.

Design load on deck plates

The following pressure head values are recommended:

Passengers and crew spaces in a craft, walkways, etc.	0.50 m H_2O
The deck area where passenger chairs are accommodated	0.35 m
Superstructure deck plates and stiffeners	0.30 m
Superstructure deck beams	0.10 m
Design uniform load of front of deck house and window are:	
For 'O' class craft	2.00 m
For 'P' class craft	1.00 m
For 'N' class craft	0.50 m
Design uniform load on side plates and windows on first floor of superstructure	0.30 m

Calculation of strength for craft in docking and lifting situation

During the calculation of strength for craft in docking and lifting situations, the vertical velocity of the craft affecting the mounting or block and the dynamic load caused by cranes have to be taken into account. In general, the inertial load coefficient should be taken as η_g = 1.25.

14.5 Safety factors

Practical experience with hovercraft is much less than that of conventional ships, therefore, as yet there are no fully consistent calculation rules and regulations for

Table 14.4 Typical safety factor applied to the strength calculation of structure of ACV/SES [4]

Load condition	Safety factor cf. yield strength	cf. ultimate strength
On cushion	1.0–1.5	1.5–2.0
Emergency	1.0	1.5
Damaged	1.0	1.0
Towing, lifting, pushing	1.5–2.0	2.0–3.0

designers' reference. Reference 4 suggests that the safety factors for strength calculation of structure can be written as in Table 14.4.

Reference 105 suggested the following factors, which are summarized in Table 14.5.

14.6 Considerations for thickness of plates in hull structural design

In general, ACV/SES are constructed of stiffened plate structures. A key parameter in determining the dimensions is to determine minimum plate thickness; here we will discuss methods to determine the necessary thickness of plates.

Step 1

At first, designers have to determine the minimum thickness of plates. Particularly for small ACV/SES, the plate thickness is not determined according to the strength of the structure, but to other requirements related to stiffness, practical construction requirements, operational durability, overhaul life of craft and corrosion of plates, etc. Reference 105 recommended that minimum thickness of plates should be as shown in Table 14.6, in which the plate thickness of some SES are also listed.

Step 2

The local thickness of plates in the region of engine mountings, propeller supports, water-jet installations and other regions in which plates will experience serious corrosion, should be thickened by at least 40%.

Step 3

In the case where the thickness of plates is less than 3 mm the frame spans should not be greater than 300 mm. Spans should not be greater than 400 mm in other conditions.

Step 4

In the lower regions of sidewalls, the thickness of plates has to be thickened or strengthened in addition to other requirements so that after strengthening the thickness should not be less than double the thickness of the shell plates.

Table 14.5 The safety factors suggested by Ref. 105

Item	Name and character	Character of calculation stress under action of the loads	Ratio between admissible and maximum stress
1	Hull and superstructure framing participating in longitudinal or transverse overall bending (including window frames)	Normal stress and shear stress due to the overall longitudinal and transverse bending	0.5
		Resultant normal stress and shear stress due to the overall longitudinal and transverse bending	0.7
2	Longitudinal framing participating in the overall longitudinal bending and resisting local load (longitudinal cargo deck and bottom panel)	Resultant normal and shear stress due to the overall longitudinal bending and bending on single stiffeners, mid-span/at supports.	0.75/0.90
3	Beam participating in the overall bending and resisting local load (framing of cargo deck, bottom, and sidewalls)	Resultant normal stress due to the overall bending moment and local bending of panel and stiffeners, mid-span/at supports	0.80/0.90
4	Shell plates and bulkhead plates of the hull and superstructure. Tank bulkheads	Normal and shear stresses due to the local loads, mid-span/at supports	0.80/0.90
5	Stiffeners of hull and superstructure not participating in the overall bending	Normal and shear stresses due to the local loads, mid-span/at supports	0.75/0.90
6	Hull structure and superstructure beams not participating in overall bending	Normal and shear stresses due to the local loads, mid-span/at supports	0.80/0.90
7	Watertight and tank bulkhead stiffeners	Normal and shear stresses due to the local loads, mid-span/at supports	0.80/0.95
8	Watertight and tank bulkhead stiffeners	Normal and shear stresses due to the local loads, mid-span	0.85
9	Pillar and bracing stability	Normal stresses due to local loadings. Single frames/cross braces	0.5/0.75 but not $> \sigma_{0.2}$

Notes:

1. In this table maximum stress can be taken as

$$\sigma_0 = K\,\sigma_{0.2} \quad \text{while in extension}$$
$$\sigma_0 = \sigma_{kp} \quad \text{while in compression}$$

where $\sigma_{0.2}$ is the assumed yield point of material equivalent to the residual deformation of 0.2%; σ_{kp} the critical stress of stiffeners considering the correction of the elastic modulus, K a coefficient,

$$K = \begin{cases} 0.9 & \text{riveted structure} \\ \left.\begin{array}{ll} 0.6 & 2 < t < 3\ \text{mm} \\ 0.7 & 2 < t < 3\ \text{mm} \\ 0.8 & 2 < t < 3\ \text{mm} \end{array}\right\} & \text{welded structure} \end{cases}$$

and t is the thickness of the connecting plates (mm).

2. In this table maximum shear stress $\tau_0 = 0.57\,\sigma_0$.

Table 14.6 Comparison of minimum plate thickness recommended by the registers of former USSR [105] with that of Chinese hovercraft

Item		Craft											
		$L \leqslant 20$ m; craft class:			$20 \leqslant L \leqslant 40$ m; craft class:			$L > 40$ m; craft class:		Chinese river SES with aluminium hull			28 m SES with steel hull
		L	P	O	L	P	O	P	O	Bow	Mid	Stern	
1	Bottom plates	1.5	2.0	2.5	2.0	2.5	3.0	3.0	3.5	4	3	2.5	3.0–2.5
2	Side plates	1.5	2.0	2.5	1.5	2.0	2.5	2.5	3.0	2.5	2.5	2.5	3.0–2.5
3	Deck plates	1.5	1.5	2.0	1.5	2.0	2.5	2.5	3.0	2.5	2.5	2.5	3.0
	Cabin floors	1.5	1.5	2.0	1.5	2.0	2.5	2.5	3.0	2.5	2.5	2.5	3.0
4	Sidewall plates	2.5	3.0	3.0	3.0	3.5	4.0	4.5	5.0	3.0	3.0	3.0	2.5
5	Plenum chamber plates	1.0	1.0	1.5	1.5	1.5	2.0	2.5	3.5				
6	Superstructure shell plate	0.8	0.8	1.0	1.0	1.0	1.5	1.5	1.5				

14.7 Hovercraft vibration

The importance and complexity of hovercraft vibration

Hovercraft vibration is a complicated problem, for the following reasons.

Vibration with a severe and superharmonic excitation source

The installed power is high for ACV/SES even though the displacement is small, hence the specific power is as high as 15–60 kW/t and with a high harmonic exciting force. For instance, on ACVs with turbine propulsion, the speed of a gas turbine is about 10 000 r.p.m. and the speed of air propeller about 1000 r.p.m. and lift fan about 500–1500 r.p.m. There are also other power transmission gearboxes and shafting systems mounted on the craft, therefore the out-of-balance exciting force (moment) induced by the engines and other non-equilibrium dynamic forces provided by some machinery will cause exciting forces (moments) on an ACV.

After a period of operation the dynamic equilibrium of air propellers and lift fans may deteriorate because of some wear or minor damage to the complicated shaft system, owing to installation errors and inaccurate centring. All of these will be vibration sources and complicate the problems of vibration.

Low natural frequency

Owing to the relatively flexible hull structure, superstructure and machinery mountings of hovercraft, the natural frequency of such structures is low. For this reason the natural frequency of mountings is low even though the mountings themselves are strong enough. Meanwhile the static and dynamic stresses on the structure are also large.

High operational speed

Hovercraft with low specific weight often operate at high speed and in high seas in comparison with conventional ships. The propeller blades of Chinese air-cooled diesel propelled ACVs have broken twice during operation. Break-up of cooling fan blades, engine mountings and thrust ring mountings has also happened to ACVs, due to violent vibration of the main engines and air propellers.

Failure of lift fan mountings, the break-up of transmission gearboxes and universal joints has also occurred to SES. With respect to oil and water pipes, the failure of these used to happen to the ACV/SES because of vibration and fatigue. Table 14.7 lists the malfunction of various machines and components due to the vibrations, summarized from practical operations.

The situation of hovercraft in the past was that during demonstrations users were always interested in ACV/SES special characteristics, but once the craft were used in service, the users would get very annoyed, as a lot of malfunctions occurred to the early craft, mainly due to the vibration. These vibrations have been the main problems facing almost all design, manufacture and operation units concerned with the ACV/SES in China. In the case where the vibration problems can be solved smoothly, not only will the rate of sorties be greatly enhanced, but also the noise level will be reduced thus improving the ride comfort of hovercraft.

Table 14.7 The malfunctions often occurring to ACV/SES caused by vibration

Items	Malfunction	Craft type	Frequency of occurrence	Main reason for malfunction
1	Exhaust pipe breakage	ACV/SES	Medium	Weak exhaust pipes, lack of elastic supports and jointing
2	Hydraulic, fuel and lubrication system piping failure	ACV/SES	High	Violent vibration, lack of elastic joining on pipes
3	Lubricating oil radiator damage	ACV	High	Poor anti-erosion and anti-vibration capability of radiators
4	Transmission gearbox damage	ACV/SES	Medium	High vibration, particularly high vibration with low frequency at starting of engines
5	V drive power transmission damage	SES	Low	
6	Engine and bearing mountings damage	ACV/SES	Medium	High vibration of engines, weak hull and mounting strength
7	Air propeller GRP blade root failure	ACV	Low	Comprehensive factors of vibration
8	Air propeller GRP blade high vibration	ACV	Medium	Rupture of dynamic equilibrium on air propellers
9	Electrical system malfunctions	ACV	Medium	Induced by erosion and vibration
10	Rudder bearing mounting failures	ACV/SES	Low	High vibration at stern, weak structure
11	Water leakage at window and door seals	ACV/SES	Medium	High vibration and weak structure, unsuitable seal designs
12	Cracking of passenger chair mounts	ACV/SES	Medium	High vibration and weak structure

Note: Weak = not stiff enough.

The problems mentioned above can be solved as soon as careful design, manufacture and maintenance are in place, as is the case for many ACV/SES made in China and in the West. The vibration problems discussed here do not imply there are specific vibration problems concerning the shaft system or structure on ACV/SES. Rather, there are comprehensive problems which have to be mastered and paid more attention to by designers during design, construction and even operation. This is similar to the design problems faced by designers of high-speed warships and monohull or catamaran fast ferries.

Such problems most often concern the vibration of the structure, main engine and shaft systems, reduction gears, mountings of bearing and engines, various equipment and instruments, pipelines and their joints, etc. which includes the selection and determination of permissible standards for vibration, the co-ordination between mechanical and structural designers in order to avoid high vibration levels during the selection of main engines, drawing the general arrangements, construction profile and deck plans and the design of various subsystems such as propellers and shaft systems. This is the so-called general design of vibration absorption.

So far we do not have an accepted common vibration standard for ACV/SES. ACV/SES differ from conventional ships, aeroplanes or helicopters and wheeled vehicles. The permissible vibration standard is more difficult to define. Here we list some vibration standards specified for marine vessels and wheeled vehicles for reference. Table 14.8 summarizes the vibration standard ISO 2372 and ISO 3945.

It would be best if ACV/SES could meet the requirement C of class IV. Figure 14.8 shows the vibration standard for conventional ships published by the Bureau Veritas of France [106], in which (a) shows the vibration standard for diesel engines and (b) that for rotating machinery. As far as ACV/SES with flexible structure, elastic mountings and couplings are concerned, it is clear that this standard level is high, but can be used as a reference. Table 14.9 details the technical standards for vibration in classification rules and construction regulations for former Soviet marine vehicles (1974) [107]; these standards are also high for ACV/SES.

We could propose the vibration standard for helicopters and perhaps such standards will be lower, considering the measurement of external force acting on helicopters. Installation of main engines in the helicopter is at the stern and the design of vibration absorption is more precise for the helicopter than the ACV/SES, which are now constructed in shipyards in China. For this reason, we do not recommend the aviation standard. We prefer to use the vibration standards for conventional ships for reference and then make engineering decisions based upon test results of prototypes.

Design for vibration absorption

Owing to the importance and complicated nature of hovercraft vibration, the considerations for vibration absorption should take in the whole course of craft developments from preliminary design, construction to sea trials. Thus it can be called the general design for vibration absorption.

It is difficult to calculate the natural frequency of bearing mountings, because the boundary conditions of supports are complex. Taking the intermediate shafts of the propeller shaft system as an example, they are only a section of the propeller shaft system, which are supported on the bearing mountings, then to the panel of superstructure, to the main structure of hull and finally supported by the buoyancy tank.

Table 14.8 The vibration standard ISO 2372 and ISO 3945 of the International Standard Organization [108]

Ranges of radial vibration severity		Quality judgement for separate classes of machines			
Range	RPM velocity in the range 10–1000 Hz at the limiting mm/s	Class I	Class II	Class III	Class IV
0.28					
	0.28				
0.45		A			
	0.45				
0.71					
	0.71		A		
1.12					
	1.12	B		A	
1.8					
	1.8		B		A
2.8					
	2.8	C		B	
4.5					
	4.5		C		B
7.1					
	7.1			C	
11.2					
	11.2	D			C
18					
	18		D		
28					
	28			D	
45					
	45				D
71					

Note:
Machine classes are defined as follows:
Class I Small machines to 20 hp
Class II Medium-size machines 20–100 hp
Class III Large machines 600–1200 r/min, 294 kW and larger, mounted on rigid supports
Class IV Large machines 600–1200 r/min, 294 kW and larger, mounted on flexible supports
Acceptance classes:
A=Good; B=Satisfactory; C=Unsatisfactory; D=Unacceptable.

Thus analysis of vibration will be best determined by progressive approximation, i.e. beginning from the analysis of arrangement of the main engines, transmission shaft system and longitudinal structure arrangement as well as the vibration conditions and progress to calculation of shaft system vibration natural frequency, engine mountings and structure and finally the tests of vibration on such systems during construction and sea trials.

Only at sea trials can designers fully determine the characteristics of vibration of a particular hovercraft. Sometimes, in order to reduce vibration, local revision (stiffening) of the structure and mountings might be carried out during the sea trials. Prototype hovercraft trials therefore always play a very important role.

(a) For diesel and reciprocating engines:
  (1) For slow engines up to 150 rpm check that $\Delta s < 0.5$ mm at the bearings and foundations.
  (2) For piping mounts and miscellaneous units $\alpha < 1.5$g.
  (3) (A) Good
    (B) Normal operating condition
    (C) Requires survey to check
    (D) Not admissible.

	v(mm/s)	A	B	C	D
Slow	< 750 kW		4.8	11	18
Fast			4.8	11	18
Slow	< 750 kW		4.8	11	30
Fast			7.0	18	50

(a) For rotating engines and line shafting measured at the bearings or foundations:
  (1) For low-speed line shafting to 150 rpm it should be checked that $\Delta s < 0.5$ mm (vibration amplitude).
  (2) For the piping mounts and miscellaneous units acceleration $\alpha < 1.5$g

	v(mm/s)	A	B	C	D
Slow	< 15 kW				
Fast	< 75 kW		1.1	2.8	7.0
Slow	< 750 kW		1.8	4.5	11.0
Fast	> 750 kW		2.8	7.0	18.0

Fig. 14.8 Acceptable vibration levels for conventional ships proposed by Bureau Veritas of France [104].

The following three considerations have to be borne in mind with respect to vibration study:

1. investigation and analysis of the exciting force;
2. calculation of natural frequency for various structural components, in order to avoid resonance with rotating components;
3. vibration isolation.

In the whole course of vibration absorption design the three issues mentioned above have to be considered as shown in Fig. 14.9, a block diagram for general design for vibration absorption of hovercraft as explained below.

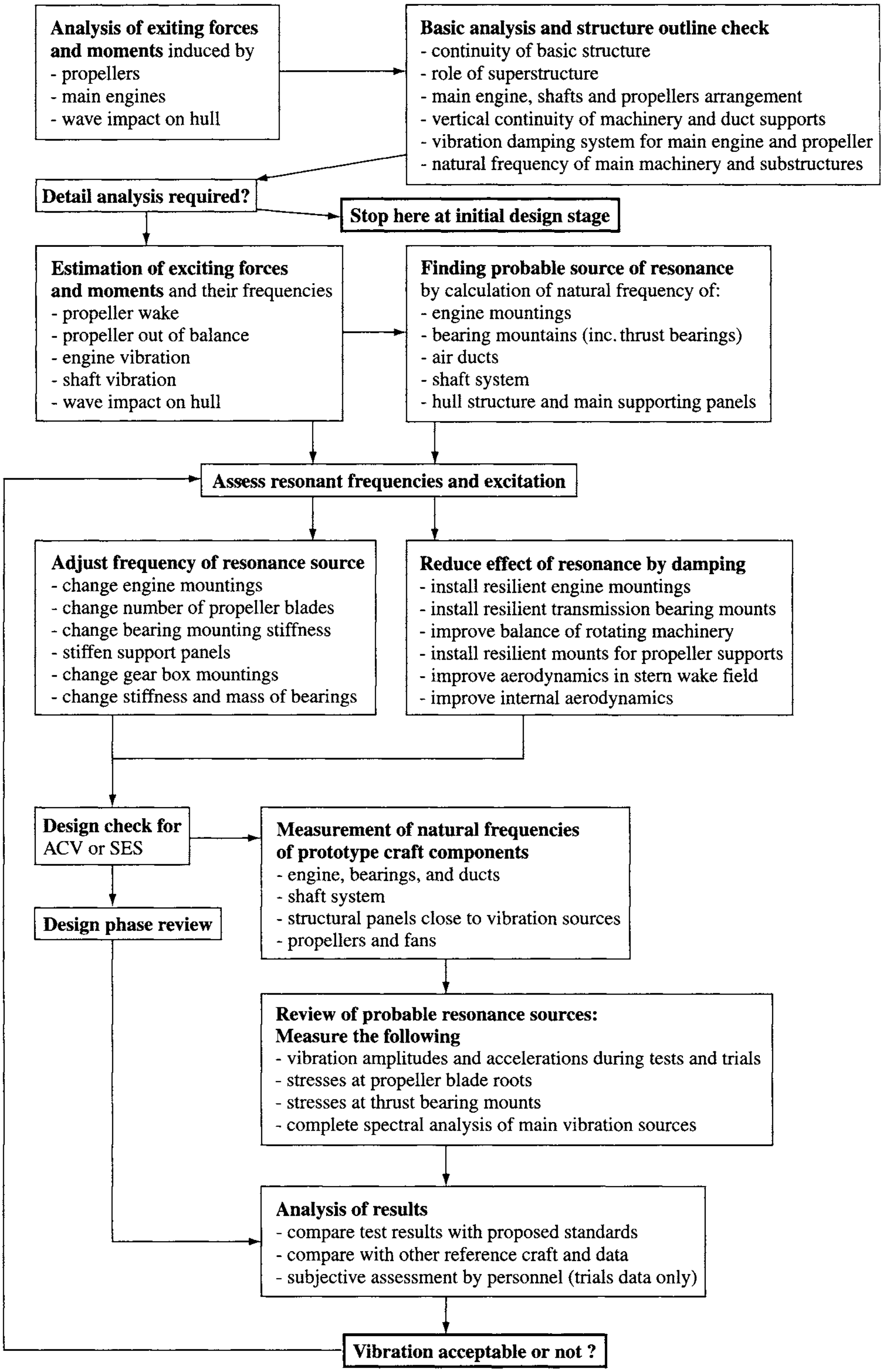

Fig. 14.9 Block diagram for vibration damping design ACV/SES.

Table 14.9 The permissible vibration and construction rules of marine vehicles in USSR [107]

Name of structure machine and equipment	Vibration axis	Frequency range (cycles/min)	Permissible amplitude (mm)	Remarks
Rigid member at stern	X, Y, Z	< 200	0.8	
			$[1.5 \times 10^4 + 85n]/n^2$	
Propeller and intermediate shafts	Y, Z	< 1500	$0.5–2.8n \times 10^{-4}$	
Gas turbine with reduction gearbox	X, Y, Z	< 850	0.35	Vibration frequency tuned to propeller revolutions
(top of reduction gearbox and bearings of turbine) and thrust bearing (top part) of turbine and diesel engines		> 850	$[0.25 \times 10^8]/n^2$	
Auxiliary machinery and heat exchangers				
Without vibration isolation	X, Y, Z		0.25	
With vibration isolation	X, Y, Z		0.5	
Navigation and radio equipment	X, Y, Z	< 300	1.0	
		300–1500	$300/n$	
Diesel engine without vibration isolation (top of diesel)	X, Y, Z	< 1000	0.5	Vibration frequency tuned to propeller revolutions
		> 1000	$[0.5 \times 10^6]/n^2$	
Diesel engine with vibration isolation (top of diesel)	X, Y, Z	< 1000	0.3	Vibration frequency tuned to shaft revolutions
		> 1000	$300/n$	
Rotating machine with vibration isolation	X, Y, Z	400–2000	$0.2–6.5 \times 10^{-5}\, n$	
		> 2000	$[0.28 \times 10^6]/n^2$	

Notes:
1. X, Y, Z denote the longitudinal, transverse and vertical vibration respectively.
2. n represents the frequency of vibration (cycle/min).
3. In the case of machine and equipment with vibration isolation, the vibration amplitude of alternative deformation of vibration isolation should not exceed the permissible value proved by the former USSR Register Bureau.
4. The vibration frequency for this standard begins from 30 cycles/min.

Preliminary design phase (or extended preliminary design)

In the preliminary design phase, the following principles have to be considered.

Exciting force

At first, designers have to consider whether the craft will operate in rivers or along a coast at sea, at low speed or high speed; if the wave-impacting force (moment) should be studied; what kind of engine will be installed in the craft, petrol, diesel or gas turbine; the degree of non-equilibrium; the exciting force induced by propellers and lift fans, etc. Based upon such considerations, designers then judge the vibration problems and whether they are likely to be troublesome or ordinary problems to solve.

General arrangement

Based on the initial craft design, the configuration of machinery within the structural arrangement, particularly the supports and foundations, the natural frequency and magnitude of forces acting on mountings of bearings, need to be reviewed, to assess the possibility of vibration isolation, and prepare a specification. It is normal on hovercraft to mount at least the main and auxiliary engines resiliently and sometimes the main gearboxes (see Chapter 16 for more details about local mechanical design). Since the calculation for vibration is very difficult, in this phase empirical rules are normally followed, based upon the analysis of previous craft prototype vibration.

Detail design phase

The following sequence of analysis is followed :

1. Estimation and analysis of exciting forces.
2. Calculation of natural frequency of various members, such as the overall vertical vibration of the hull at full loaded and light displacement; the local vibration of stiffeners, panels, shell plates, deck plates and bulkhead plates; calculation of natural frequency of mountings of main engines, bearings, gearboxes and air propeller ducts, etc.; natural frequency of shaft systems.
3. Referring to the critical operational frequency as shown in Fig. 14.10, it can be seen that during calculation of the frequency of the exciting force, the speed of transmission shaft, i.e. shaft frequency, double of shaft frequency, the speed of shaft times the number of blades and so on need to be considered. The probability of resonance vibration between the exciting force and various mountings is high, but can be avoided at the operational range of engine speed as long as it is given careful consideration.
4. In this calculation, the following profile of vibration resonance between the various exciting forces and mountings and components have to be checked.
   (a) The allowance between the shaft speed and the natural frequency of vibration of the shaft system at the first mode is at least 20%.
   (b) On cushion-borne and hull-borne operation, the natural vibration frequency at the first mode of the hull structure should exceed the frequency of the exciting force.
   (c) The natural frequency of bottom plates and stiffeners at the stern should be higher than propeller speed by 50 and 30% respectively. The natural vibration frequency of plates and stiffeners at engines should be greater than the speed of the crankshaft and double that of crankshaft speed plus 50% and 30% respectively.
   (d) The propeller shaft speed times the number of blades should avoid the natural frequency of vibration of the stiffeners in the region where the propeller is located.
   (e) In the region of the lift fan, the natural frequency at the first mode of the local stucture of the hull should avoid the speed of the lift fan times the number of blades and so on.
   (f) The allowance between the natural frequency at the first mode of bearing mountings and the shaft frequency should be at least 30–50%.
   (g) The allowance between the natural frequency at the first mode of thrust bearing mountings and the shaft frequency of the propeller should be at least 30–50%.

(h) The blade frequency of the propeller should avoid double shaft speed and n times shaft speed with the number of blades and so on.
The frequency of the foregoing calculation should include the vibration frequency in vertical, transverse and longitudinal directions.

5. In the case of resonance, the frequency adjustments should be made first, i.e. changing the stiffness of mountings at resonance, since this is the simplest job, then changing the mass of the rotating component, thus changing the general arrangement, or shaft arrangement, which it is preferable to avoid.

Changing the stiffness of structure at which the resonance occurs, for instance in the case where the resonance occurs to the panel of the bottom structure at the stern, then the stiffness of such a structure should change and so on.

If the frequency adjustments fail to provide the desired effect, then designers are obliged to take measures to reduce the exciting force :

- changing the number of propeller blades;
- implement vibration absorption for main engines and shaft system;
- use elastic coupling;

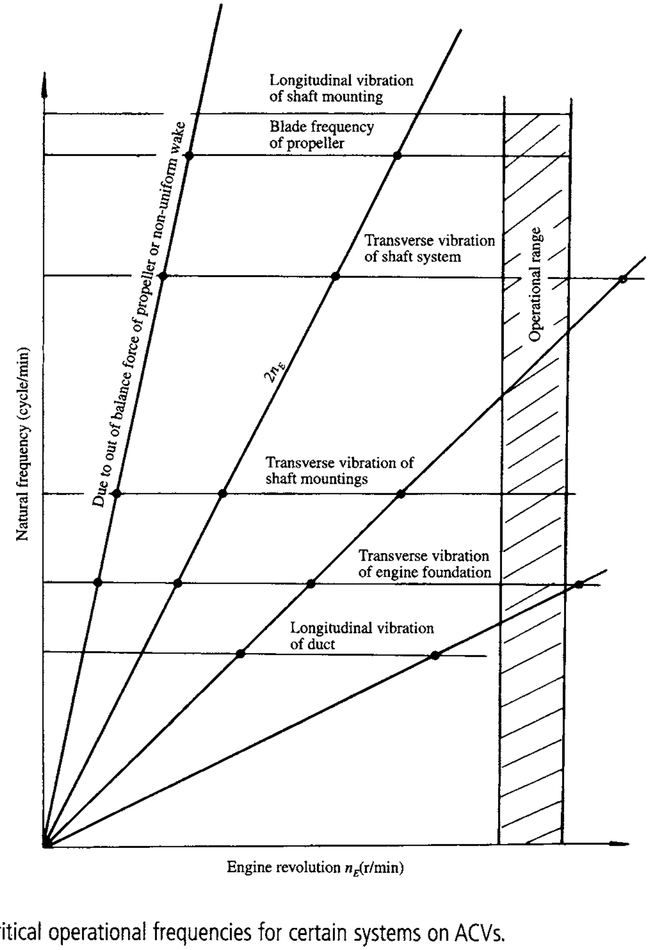

Fig. 14.10 Critical operational frequencies for certain systems on ACVs.

- belt transmission and so on can be adopted on the shaft system in order to dampen the exciting force.

Construction

In the case where the problems mentioned above have been solved, or it is predicted that these problems might be solved, then the production design and construction can be undertaken. If the designers anticipate that trouble may occur in the vibration absorption design, such as large exciting forces being in existence, frequency difficult to adjust and vibration absorption difficult to measure, etc. then attention has to be paid during construction to the static and dynamic balance, strict centering and vibration isolation of equipment and instruments and mounting of pipelines.

Table 14.10 Assessment of mechanical vibration [40]

Frequency of vibration	Primary reasons	Secondary reasons
Shaft frequency	Shaft not properly balanced	1. Eccentric gear journals 2. Poor shaft centring 3. Shaft not straight 4. Faulty transmission belt 5. Resonance 6. Unbalanced attached components 7. Problems with electrical systems
Shaft frequency × 2	Eccentricity for mechanical loosening	1. In the case of large axial vibrations then the eccentricity will be the main reason 2. Unbalanced attached components 3. Resonance 4. Faulty transmission belt
Shaft frequency × 3	Eccentricity	In general it is due to the out of centre and excessive slack in the axial direction
< Shaft frequency	Vibration on oil membranes, normally at 0.43 × shaft frequency	1. Faulty belt transmission 2. Background vibration 3. Low mode harmonic response 4. Beat vibration
Shaft frequency harmonics	1. Gear failure 2. Poor belt transmission 3. Aerodynamic action 4. Hydrodynamic action 5. Electric problems 6. Roller bearing failure 7. Air pressure fluctuation 8. Mechanical loosening 9. Reciprocating inertia forces 10. Combustion force	1. The number of teeth of failed gear × shaft frequency 2. 1 to 4 × shaft frequency, for defective belts 3. Number of fan blades × shaft frequency 4. Number of impeller blades × shaft frequency 5. In the case of mechanical loosening, 2, 3, 4, × shaft frequency and other harmonics 6. Imbalanced force at high order at inertia moment
Harmonic vibration frequencies not related to shaft harmonics	Failure of journal bearing	1. Unstable vibration on bearing 2. Vibration due to the friction of poorly lubricated journal bearing 3. Vibration with high random due to cavitation, turbulence and backflow 4. Vibration due to friction

Static hovering tests, and during trials

During tests of craft hovering statically and on trials, tests also have to be carried out and compared with the vibration standard mentioned above. In the case where the test results are not satisfactory, then the vibration source should be found by spectral analysis with attempts to adjust frequency by changing mounting characteristics, etc. to reduce the exciting force once again. Failure diagnosis can be undertaken according to Table 14.10 [44]. Meanwhile, the overall and alternating stress at the blade roots of air propellers, mountings and bearings and engines should be measured as good practice.