12

Lift system design

12.1 Introduction

In Chapter 2 we introduced air cushion theory and its development to date. Modern air cushion systems are based on the plenum chamber principle, though generally using lower skirt fingers or segments which encourage an element of air jet sealing to improve free air gap and so reduce drag. The normal starting point for lift system design is to assume a cushion system which includes segments or fingers as the primary cushion seal and an upper skirt including a bag or loop which acts as an air distribution duct, and spring/damping system. If another system (e.g. extended segment system) is used, then the same cushion system elements have to be provided by other means.

Thanks to improvements in the design of flexible skirts, the air gap under the segment tips necessary for minimized drag, and thus the required air inflow rate, is small. The lift power of modern hovercraft is not excessive in proportion to the total craft installed power compared with hovercraft built in the 1960s. The decrease in bag pressures used (typically down from 1.4 p_c to 1.1 p_c), which lead to the development of more responsive skirts, also results in minimized fan total pressure and therefore reduced power.

The specific power of ACV/SES, i.e. lift power plus propulsion power divided by craft all-up weight, has been reduced significantly, from 90 kW/t for SR.N1 in 1960, to 50 kW/t for SR.N4 in 1970 and 30 kW/t for both SR.N4 MK 3 and AP1.88 in 1982. Since that time the specific power for lift systems has stabilized.

Meanwhile, lift system design must be approached carefully, because for an ACV, the lift power is still approximately 1/3 of the total craft installed power. Figure 12.1 shows the typical distribution diagram for the lift system and the pressure distribution of the lift system, in which q_0 denotes the pressure recovery of air inflow of the craft in motion, $k_1 q_1$ denotes the inflow pressure loss and H_j denotes the overall pressure head of the fan. Airflow is through air ducts with a diffusion loss in section (3), then into a skirt bag with head loss due to sudden diffusion, and to the bag pressure p_t (since the air velocity is very small, therefore the dynamic pressure in the skirt bag is also very small); subsequently the air flows via the skirt bag and into the cushion, at cushion pressure p_c.

G. H. Elsley [92] investigated the power distribution of small high-speed ACVs operating at the speed of 1.5 times hump speed. From Table 12.1 it can be seen that

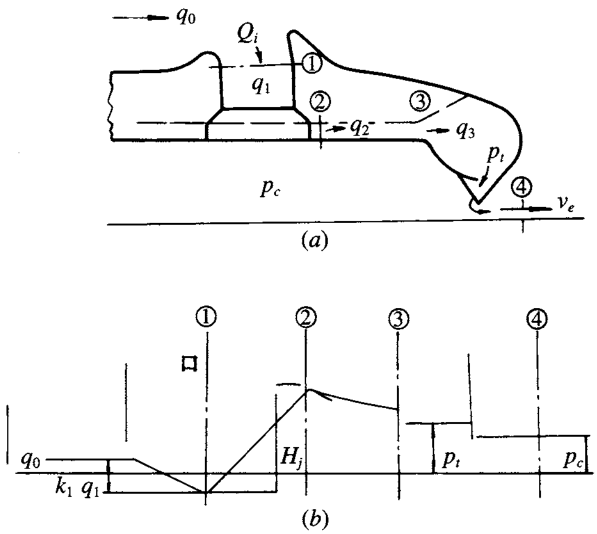


Fig. 12.1 Layout of a typical ACV lift system and the distribution of pressure in different positions.

Table 12.1 Power analysis for three small ACVs at 1.5 × hump speed in head winds and waves [100]

No.	Power consumption item	% Installed power	% Cumulative power saving from 10% reduction in this item	Related items
1	Cushion flow losses	14	4.3	1, 2, 3, 7, 8
2	Skirt drag	24	4.0	4, 8
3	Propeller/propulsor loss	25	2.5	8
4	Intake, fan and ducting losses	15	1.5	1
5	Wave-making drag	6	1.0	5, 8
6	Skirt pressure loss	7	0.8	1, 3
7	Profile drag	5	0.8	6, 8
8	Momentum drag	4	0.7	7, 8

saving cushion flow and skirt drag will obtain the best results, because it not only saves on the power of this item but also other items, e.g. 1, 3, 7 and 8, etc. This illustrates the important role which cushion air flow has for saving flow rate.

In this chapter our aim is to define the cushion as an air supply and ducting system, so as to be able to select an appropriate type of lift fan (axial, mixed flow or centrifugal) to feed it. Fans can then be designed and built to the required specification by specialist suppliers, or a standard unit selected from an existing range. Standard fans used for hovercraft are generally units designed for ventilation systems for buildings. There are a number of suppliers world-wide of axial and mixed flow fans with non-metallic lightweight blades which are reasonably efficient. Centrifugal fans suitable for ACV/SES on the other hand are usually built to specification, since for industrial use steel is used rather than the aluminium which is preferred for an ACV.

We will discuss the following aspects:

1. the determination of cushion parameters concerned with the performance of the craft, e.g. the relative flow rate $\bar{Q}$, the bag cushion pressure ratio p_t/p_c and the cushion pressure/length ratio p_c/l_c;
2. design of air inlet and outlet, as these seriously affect the loss of total pressure head;
3. air duct design;
4. air fan design.

12.2 Determination of air flow rate, pressure and lift system power

Determination of air flow

Reference is made to sections 2.3–2.5 for background.

The flow coefficient $\bar{Q}$, can be written as

$$\bar{Q} = \frac{Q}{S_c\,(2p_c/\rho_a)^{0.5}} \tag{12.1}$$

where Q is the air flow rate (m^3/s), S_c the cushion area (m^2), p_c the cushion pressure (N/m^2) and ρ_a the air density (N s^2/m^2). The value of $\bar{Q}$ is dependent upon the design requirements for the craft speed, speed loss in waves and control of heaving acceleration. It has to be taken into account comprehensively in the overall design of a craft. The effect of air flow on overall performance of a craft has been described in earlier chapters, but can be estimated as follows in preliminary design or initial project design stages.

Statistical method

Based on craft actually constructed, a statistical analysis can be made as follows (ref. 15, see also section 2.5):

$$\bar{Q} = 0.015\text{–}0.030 \quad \text{(ACV) [15]} \tag{12.1a}$$

$$\bar{Q} = 0.005\text{–}0.010 \quad \text{(SES)} \tag{12.1b}$$

which can also be written as follows [108]:

$$Q/W = 5.0\text{–}10.0\ \text{m}^3/\text{s/t} \quad \text{(ACV) [93]} \tag{12.1c}$$

$$Q/W = 0.5\text{–}1.3\ \text{m}^3/\text{s/t} \quad \text{(SES)} \tag{12.1d}$$

The flow should be assumed to be at the upper limit of this expression to start with, while the flow rate coefficient for river hovercraft will be rather lower as shown in Table 12.2.

Model experimental method

In a towing tank, the calm water drag and vertical acceleration in waves at different $\bar{Q}$ can be measured and the optimum flow rate (or range, for a craft operating envelope) selected. Figure 12.2 shows the drag curve of British SES model HM-2 at different flow rates.

Designers can also make decisions using full-scale prototype test results on calm water and waves. This method is rather more costly and time-consuming than building a scale model. It is normally used if the original skirt design gives different results from the model tests, as a method to optimize craft performance.

Approximate equations

An initial estimate of the cushion flow rate required for an ACV at project design stage can be obtained according to the required equivalent hovering height H_e/l_c, based on past experience from completed ACVs. H_e/H_c can be determined statistically as in equation (12.2) [4, 93]:

$$H_e/l_c = 0.05/W^{0.5} \tag{12.2a}$$

Table 12.2 Flow coefficient for various Hovercraft

Craft	Country	Type	Cushion length l_c (m)	Cushion area S_c (m^2)	Craft weight W (t)	Cushion pressure p_c (N/m^2)	Flow rate Q (m^3/s)	$\bar{Q}$ bar
HM.216	UK	SES	11.95	58.29	18.38	3570	29.80	0.0067
HM.218	UK	SES	14.39	70.19	27.30	3390	21.50	0.00412
HM.221	UK	SES	17.44	85.07	37.20	3339	26.80	0.00427
713	China	SES	17.5	87.30	28.00	3200	50.00	0.008
711-IIA	China	ACV	10.09	52.32	6.40	1290	51.70	0.0217
SR.N6	UK	ACV	14.8	78.00	10.00	1280	75.10	0.0213
SR.N4	UK	ACV	39.7	780.00	200.00	2570	453.40	0.00906
JEFF (A)	USA	ACV	28.0	335.00	157.00	4690	362.70	0.0124

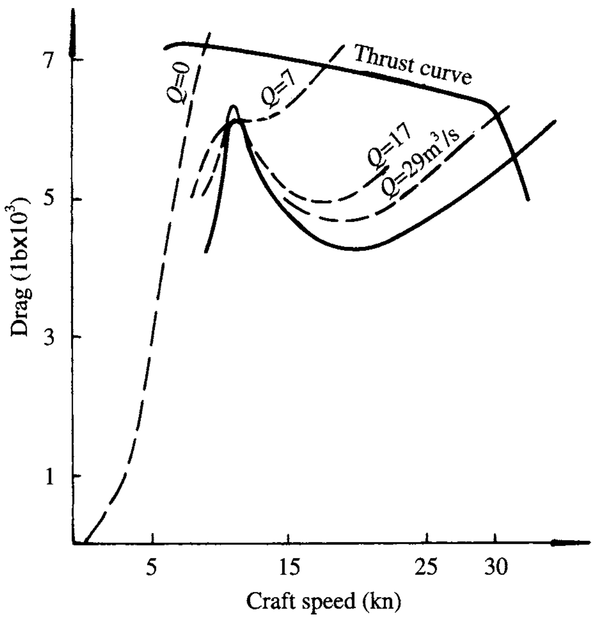


Fig. 12.2 Influence of flow rate Q on drag of SES model HM-2.

or

$$H_e/l_c = k_1\,(p_c/l_c)^{0.5} \tag{12.2b}$$

where H_e is the equivalent air gap under the skirts, including delta between fingers (ft), $l_c = S_c/B_c$, W is the craft weight (t), p_c/l_c the cushion pressure/length ratio (lb/ft^3), S_c the cushion area and k_1 a constant, which we can take as $0.0035 < k < 0.0046$ or as shown in Fig. 12.3.

After obtaining the equivalent air gap, the air outflow Q can be calculated as

$$Q = \phi\, l_j\, H_e\, (2p_c/\rho_a)^{0.5} \tag{12.3}$$

where l_j is the peripheral length of air leakage (m) and ϕ the flow coefficient (take ϕ = 0.6).

Flow rate for minimum calm water drag Q_{minR}

Owing to momentum drag of the cushion air increasing with flow rate and skirt drag increasing inversely with flow rate, ref. 98 proposed the following relation for the flow rate Q_{minR}, for minimum drag:

$$Q_{\text{minR}} = [0.34\, b/a]^{0.75} \tag{12.4}$$

$$a = \rho_a V_s$$

$$b = C_{\text{dsk}}\, l_j^{1.68}\, (2p_c/\rho_a)^{0.17}\, S_c^{0.5}\, 0.5\rho_w\, V_s^2$$

where C_{dsk} is the skirt drag coefficient of craft on calm water,

$$C_{\text{dsk}} = 1.5 \text{ (typically)}$$

Meanwhile, the flow at which the minimum total installed power can be obtained [98] can be written as

$$Q_{\text{minp}} = [0.34b/(a + (p_c\,\eta_p/V_s\,\eta_l))]^{0.75}$$

where η_p is the overall propulsion efficiency and η_l the overall lift efficiency.

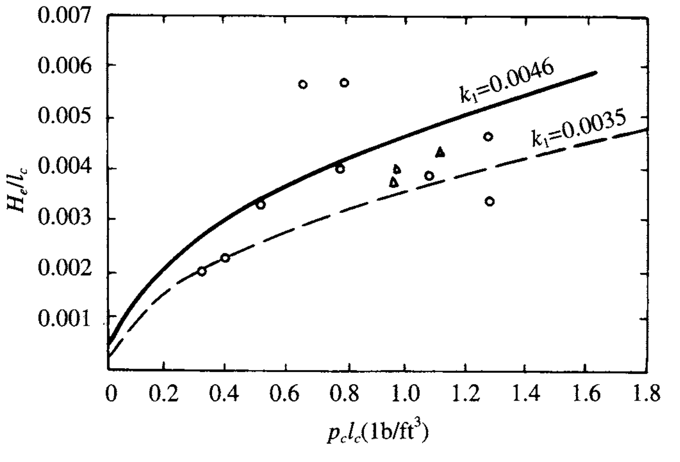


Fig. 12.3 Relative air clearance vs p_c/l_c.

These equations should only be used for reference as a starting point in determination of ACV cushion air flow. Meanwhile it may be noted that all of the physical values in these expressions are imperial dimensions, e.g. ft for length and lb for weight, etc., while the equivalent in SI units is given on the right-hand side.

Determination of overall pressure head of fans

The determination of overall pressure at the fan requires the calculation of the various pressure drops in the 'pipe' system. To begin this, at first one has to determine the bag cushion pressure ratio p_t/p_c of the skirt to meet the requirements for stability and seaworthiness. Early hovercraft designs, in order to prevent the plough-in of craft, in general had a high bag/cushion pressure ratio.

In the early 1980s, once the mechanism of skirt deformation in the bow area was fully understood, it was shown that plough-in could be prevented by means of improving the type and configuration of skirts and air ducts as well as regulating the ACV dynamic trim. Therefore designers now often adopt a low bag/cushion pressure ratio, such as 1.09 for ACV model 7202.

With respect to the bag/cushion pressure ratio of SES, in order to minimise the losses in air ducts and lift fan system, sometimes an air duct or bow/stern fan might be deleted, and the bag/cushion pressure ratio may be even lower, as the air duct system will be greatly simplified. A typical calculation for the air ducts of ACV/SES can be written as follows.

Air duct calculation for ACVs

The arrangements of ACV fans and air ducts differ greatly and dependent on the passenger cabin arrangement, machinery bay, transmission shaft system, fans and propellers. The calculation of air duct head loss and the flow distribution have to be analysed for the specific craft design. A typical example can be seen in Fig. 12.1. This figure shows that one fan blows the pressure air via the pressure chamber into the bags (such air ducts have been used on Chinese ACVs 711-IIA, 716-II, 7206 and 7210 for example). Sometimes the air ducts of skirt bags are connected with each other, so the overall pressure in various bags is basically equal, thus the problems are simplified. An arrangement is shown in Fig. 12.4, namely the air duct system becomes a parallel pipeline system and its branch pipes are:

1. via the air feed holes of the bow and fore side bag into the cushion;
2. via the air feed holes of the stern and rear side bag into the cushion;
3. via the air feed holes of the longitudinal stability skirt bag into the cushion;
4. via air feed holes of the transverse stability skirt bag into the cushion.

According to the Bernoulli equation and knowing the cross-section area of the duct sections, the flow rate in air ducts Q_1, Q_2, Q_3, Q_4, etc. can be calculated as shown in section 8.4.2.

During the initial performance analysis, before the craft geometry is fixed, the calculation can be worked out according to the different bag/cushion pressure ratios in various air ducts, in order to get different skirt stiffnesses. The equation will have multiple degrees of freedom, so the flow equation is best solved by an iteration method.

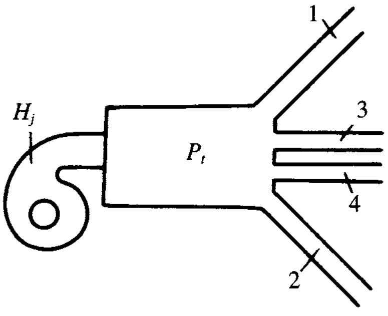


Fig. 12.4 Layout of simulated pressure distribution of typical lift systems (H_j = fan pressure, P_t = bag pressure): (1) air from bow and fore bag, via its bow into cushion; (2) air from stern and rear bag, via its hole into cushion; (3) air from longitudinal stability skirt bag and via its holes into cushion; (4) air from transverse stability bag and via its holes into cushion.

In order to reduce the work, during the preliminary design, the overall fan pressure on an ACV can be estimated by simplified expressions as follows [94]:

$$H_j = p_c\,(p_t/p_c) + k_d\,(Q/d_2^2)^2 + k_E q_E - k_R q_A \tag{12.5}$$

where d_2 is the diameter of the fan impeller, k_d the air duct head loss coefficient from outlet of fan to skirt bag (as shown in Table 12.3), k_E the head loss coefficient of fan air inlet (as shown in Table 12.4), k_R a coefficient due to the head recovery of air inlet (as shown in Table 12.5), q_E the dynamic head of fan air inlet, which can be written as

$$q_E = 0.5\rho_a\,[4Q/(\pi d_E^2)]^2 \tag{12.6}$$

d_E is the diameter of the fan inlet and Q the air inflow rate,

$$q_A = 0.5\rho_a\,(V_s + V_w)^2 \tag{12.7}$$

where V_s is the craft speed and V_w the wind speed (it is shown above as in head wind condition). The first item of the right-hand side of equation (12.5) represents the bag

Table 12.3 Loss coefficient in air ducts k_d [94] (including SES)

Year	Craft weight (t)	Type of air flow distribution (air duct system): Free diffusion into pressure chamber (on craft SR.N4, BH.7)	Mixed controlled and free diffusion (on craft SR.N6)	Centrifugal fans with short outlet diffusers (on craft JEFF(A), VT 1, 3KSES)	Axial fans with conical diffusers (on craft HD.2)
1980	100	0.037	0.0114	0.0063	0.0035
	10 000	0.037		0.0055	0.0030
1990	100	0.035	0.0110	0.0055	0.0032
	10 000	0.035		0.0045	0.0028
2000	100	0.033	0.0105	0.0050	0.0030
	10 000	0.033		0.0042	0.0026

Table 12.4 Loss coefficient of fan inlet k_E (including SES)

Year	Craft weight (t)	Configuration of fan inlet: Opening inlet, integrated fan system (on craft SR.N6, SR.N4, BH.7)	Opening inlet, circular inlet (on craft JEFF(B), HD2, N300, N500)	Air ingested from diffusion (on craft JEFF(A), VT1, VT2, CC-8, 3KSES, SES-100A, SES-100B)
1980	100	0.016	0.01	0.016
	10 000	0.014	0.009	0.014
1990	100	0.012	0.009	0.012
	10 000	0.011	0.008	0.011
2000	100	0.010	0.008	0.010
	10 000	0.009	0.007	0.009

Table 12.5 Pressure recovery coefficient at inlet of fan k_R (including SES)

Year	Craft weight (t)	Configuration of fan inlet: Rearwards facing fan inlet (on craft HD.2)	Integrated fan and propeller system (on craft SR.N6, SR.N4, LACV-30, BH.7)	Flush inlet (on craft JEFF(A)&(B), VT 1 and 2, SES-100B, N300 and 500, 3KSES, CC-7, MV-PP5, Aist, etc.)	Scooping inlet (on craft SES-100A)	Fan forward inlet (on craft HM.2, Gus)
1980	100	− 0.2	0.4	0.5	0.7	1.0
	10 000	− 0.2	0.5	0.6	0.8	1.0
1990	100	− 0.2	0.45	0.55	0.8	1.0
	10 000	− 0.2	0.55	0.65	0.85	1.0
2000	100	− 0.2	0.5	0.6	0.85	1.0
	10 000	− 0.2	0.6	0.7	0.9	1.0

pressure, the second the head loss between the fan outlet to the skirt bag, the third the head loss at the air inlet and the fourth item denotes the head recovery at the fan inlet.

Calculation of overall pressure and lift system power for SES

The typical air duct system of an SES is shown in Fig. 12.5(a) (similar to that on craft 717) and an equivalent pipe system is shown as Fig. 12.5(b), i.e. the parallel pipeline of the air supply system. The equation, similar to that in the ACV lift system, can also be solved by the Bernoulli equation. Thus according to the required air flow and the configuration of air ducts, the overall head H_j and lift power N_{el} (in kW), can be written

$$N_{el} = H_j\ Q_i/[1000\eta_F\ \eta_M] \qquad (12.8)$$

where H_j is the fan overall pressure (N/m^2), Q_i the air inflow of the fan (m^3/s), η_F the fan efficiency and η_M the transmission efficiency. The calculation concerning the fan

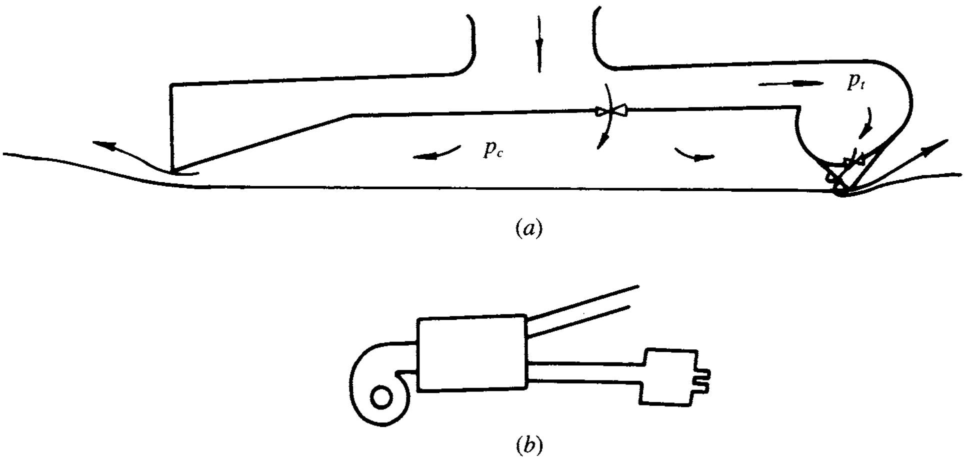


Fig. 12.5 A typical equivalent air duct system for SES: (a) air duct configuration; (b) equivalent parallel air network for calculations.

recovery pressure coefficient and pressure loss at inlet, etc. can be calculated as detailed above.

12.3 Design of fan air inlet/outlet systems

Inlet system

From Figure 12.4, it can be seen that the air inlet pressure recovery will play a very important role for an ACV with large lift air flow rate, and so a significant saving can be obtained by using a well-designed air inlet. It will be less important for SES due to the lower lift system air flow rates.

The British Hovercraft Corporation has carried out much research work [95] in this area. The ACV series SR, such as SR.N4, SR.N5, SR.N6, all used an integrated system for lift system fans and air propellers mounted above the cabin structure, as shown in Figs 12.7 and 12.8. Reference 96 specially introduced the effect of the geometrical parameter of the inlet on the air inlet pressure recovery. They investigated the effect of inclination angle of the inlet and lip configuration on the air pressure recovery, as shown in Fig. 12.10. Figure 12.9 shows the effect of inlet position on the lift power.

All of these factors can give a reference for inlet design, which may be summarized as follows:

1. During design of the lift fan inlet, the advantage of an air scooping inlet should be considered. Figure 12.8 shows the effect of air inlet position on the lift power. The difference of lift power between a positive air recovery inlet and negative inlet will be up to 5–8%.
2. The ingestion effect of air in the laminar layer close to craft structure surfaces

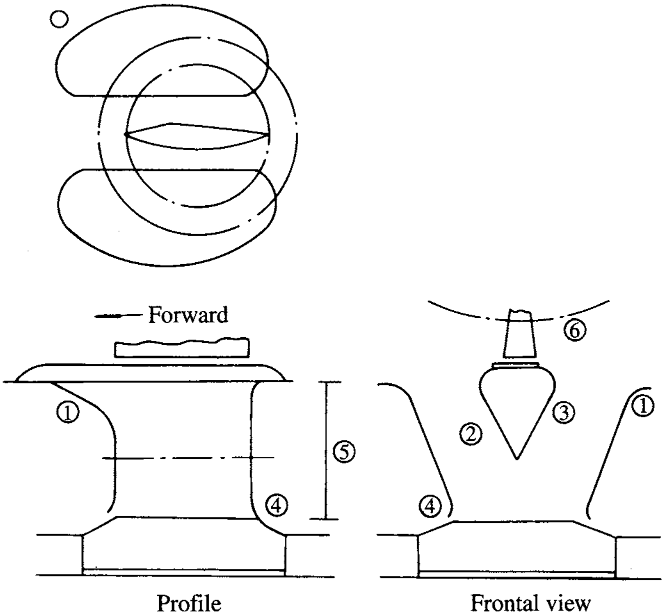


Fig. 12.6 Design considerations for air inlets of integrated lift/propulsion systems. 1: increased forward lip radius; 2: air inlet fairings; 3: centre body for smooth flow; 4: close fitting inlet to fan; 5: height of air inlet duct; 6: influence of air propeller/pylon.

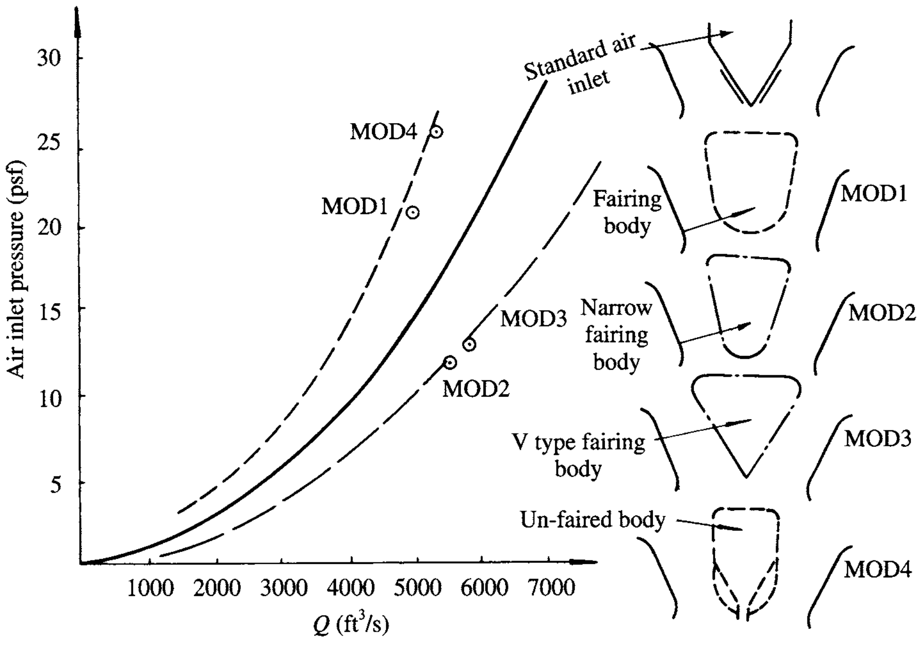


Fig. 12.7 Influence of gearbox shape on air inlet systems pressure losses.

should also be considered. In general, the inlet front lip should be designed with a larger radius of curvature, as 1 in Fig. 12.6 and the foreward part of A in Fig. 12.9, and slightly protruding in order to increase the air pressure recovery coefficient. Figure 12.9(c) is a typical example investigating the air inlet pressure recovery [96].

The researchers in this reference carried out three projects, two types of inlet foreward lips A, C and one inlet rear protruding part B compared to the standard symetrical fan inlet. The maximum air inlet recovery of the combination of these three parts could therefore be obtained. The figure shows that the second project, namely the standard inlet plus foreward lip A and rear lip B, is found to be the optimum. Moreover, it will be better for the angle between the centre line of fan inlet and the oncoming flow angle a to be smaller than 90° in order to obtain better ram pressure recovery. This is not necessary in the case of small craft and at large inlet ram pressures, i.e. at higher values.

3. A definite height of inlet tunnel is required to obtain straight air streamlines in the inlet to enhance the fan efficiency. Too high an inlet tunnel will increase the craft height, therefore it is a trade-off between the two requirements, as shown at 5 in Fig. 12.6.
4. In general, the transmission gearbox is located in the fan inlet, for practical reasons. It is therefore suggested that the gearbox is carefully designed to reduce its size and thus minimize air inflow blockage, particularly at the contracting section of the inlet. MARIC have experience with a gearbox and fan arranged exactly at the contracting section of the fan inlet of an ACV model, which consequently could not hover up. This was improved after raising the position of the gearbox.

Designers are also recommended to pay attention to the shape of the centre body of the gearbox. Figure 12.7 shows results from study work carried out by BHC. The difference of inlet pressure loss is 10% between the non-smooth gearbox model

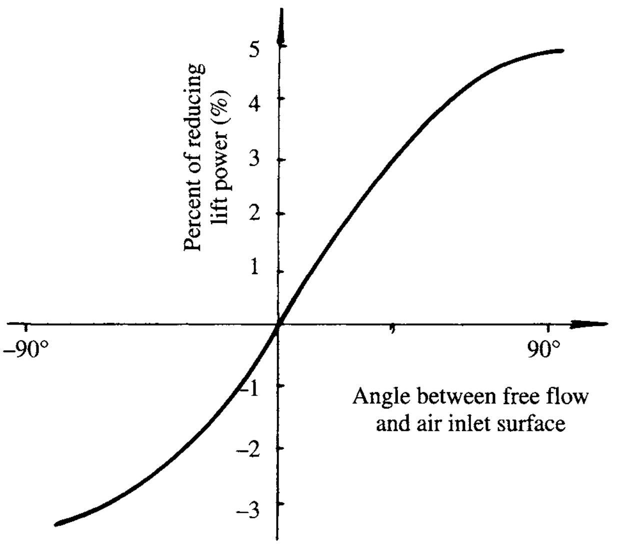
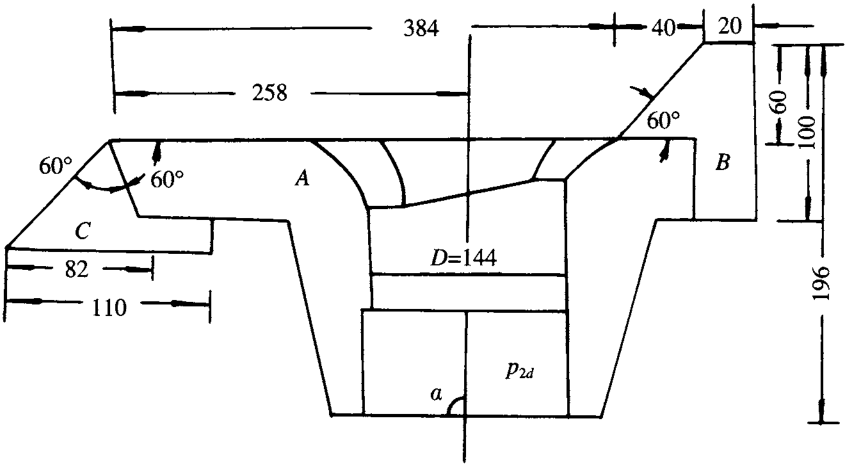


Fig. 12.8 Lift power versus air inlet location.

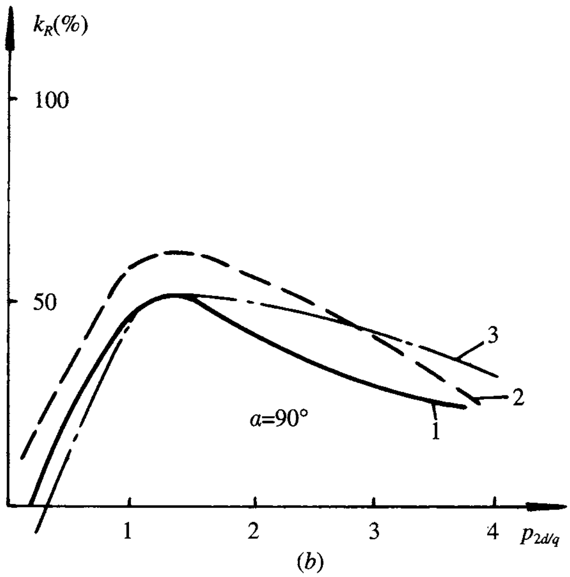
MOD 4, blocking at the contracting section of model MOD 1, and the narrow centre body with less blockage at contraction section model MOD 3. Thus it can be seen that about 10% of lift power may be saved.

5. Increasing the air inlet duct area to decrease the velocity of air inflow will give a significant reduction of inlet losses, because the dynamic pressure at the air inlet increases in square proportion to the inflow velocity, thus increases inversely in



(*a*)

Fig. 12.9 (a) Influence of inclination angle of air inlet on ram air pressure recovery: (a) dimension and variations of air inlet geometry.



(*b*)

Fig. 12.9 (b) ram air pressure recovery coefficient vs air inlet configuration. 1: standard air inlet plus A; 2: standard air inlet plus B; 3: standard air inlet plus A and C.

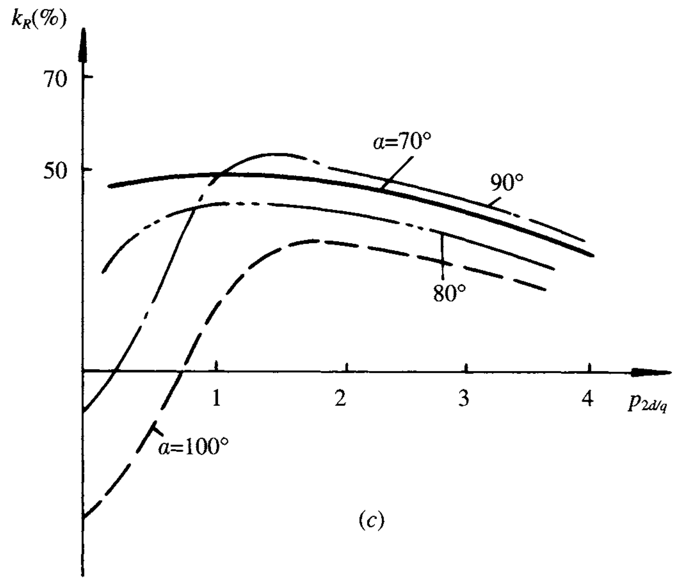


Fig. 12.9 (c) ram air pressure recovery vs inclination angle of air inlet.

fourth proportion with the diameter of air inlet. Moreover, discontinuity of section area for inflow should be avoided as far as possible in order to reduce pressure losses, as shown in Fig. 12.6.

6. Continuity of cross-section of the fan inlet in the axial direction should be preserved with the foreward disc of the fan impeller in order to avoid air flow separation and leakage of pressurized air as well as reduction of fan efficiency as shown in Fig. 12.10.
7. In general, the fan air inlets might be arranged at the craft rear or stern in order to reduce the craft aerodynamic form drag. If this arrangement is used the distribution of air inflow between the fans and air propellers has to be considered carefully.

In our experience, if some care is taken this problem can be solved successfully and vibration of shaft systems, engine mountings, air ducts and air propellers due to the non-uniformity of air inflow avoided. This is because air density is less than water density, so as to make a smaller vibration exciting force. However, it is better to carry out model test investigations in a wind tunnel with the actual arrangement in order to avoid violent vibration at this part.

ACV fan air outlet system

At the fan air outlet, the air flow velocity and dynamic pressure are highest. In addition the air flow will be very turbulent unless a downstream volute and guide vanes are fitted. For this reason, it must be designed carefully.

Outflow has to be led with no air blockage due to sudden changes of cross-sectional area and diffused gently to the craft air duct system. Air blockage sometimes may occur to the craft, because of a large amount of pipeline arranged in the plenum chamber, but it is unlikely to be a problem in the case where the blocking structures

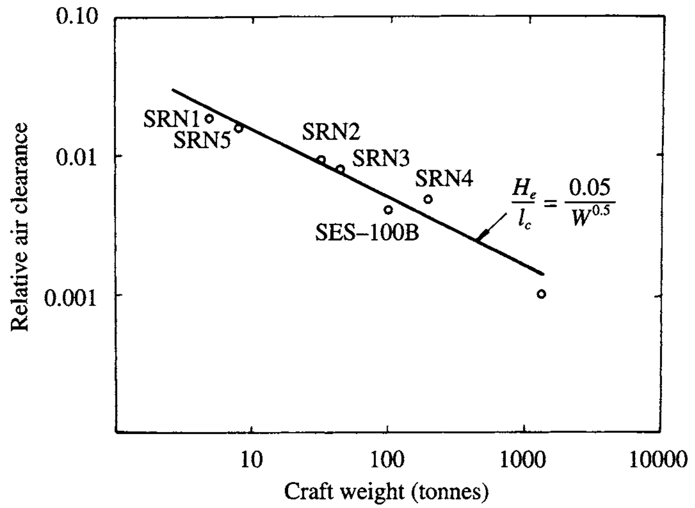


Fig. 12.10 Relative air clearance H_c/L_c for ACV vs craft weight.

are slight and far from the fan outlet. Therefore the fan outlet has to be designed carefully with the aid of model experiments for measuring loss of pressure.

The skirt bag can also be used as the plenum chamber for diffusion of pressurized air. MARIC has a lot of experience on this subject. Due to the lack of care during design and construction of some craft, there were pipes and frames blocking the outflow, which caused reduction of pressure and air clearance under the skirt so as to lead to deterioration of take-off through hump speed. The craft sometimes lost speed as low as to hump speed in the case of head wind, overload and incorrect handling.

In general there are three air outlet system designs which may be used on an ACV, these are as follows.

Vertical arrangement of fans

In this case, the fan is arranged perpendicular to the ground or waterline. The air is blown out from the fan outlet directly into the skirt bag or air cushion and uses the skirt as the diffusion duct. Since the fan can diffuse the pressurized air in the volute system, then the air velocity at the outlet can be decreased significantly in order to reduce the pressure loss, which can also be calculated accurately. This system can obtain a high air outlet efficiency.

Horizontal arrangement of fan (without volute)

This is an arrangement such as used for the BHC SR.N4 and SR.N6. The key requirement of such an arrangement is that the corridor from the air duct has to be roomy not only in the tangential direction, but also in the radial direction. All stiffeners or frames, pipes, cables, etc. have to be arranged far from the outlet. Although these fans are without a volute to diffuse air progressively into the cushion, the streamline is diffused freely in an intermediate plenum chamber formed by the craft buoyancy tank and hull structure. Experiments prove that this system can give high air duct efficiency due to the low air velocities.

Horizontal arrangement of fan with part volute (or guide plate)

Often due to various requiremnts for craft layout, the air streamline in the horizontal arrangement of fans cannot provide free diffusion, but needs to be diffused smoothly via a part volute or flow-guide plates. In this case particular attention has to be paid by designers, because the existence of machines, equipment and structure will affect the layout of volute and flow guide plates so as to reduce the air duct efficiency and so skirt air gap, stability, seaworthiness, anti-plough-in capability, etc.

SES fan outlet system

The lift fan flow rate of an SES is smaller than that of an ACV, thus the size of fans for an SES is also smaller. For this reason, SES fans are generally accommodated in the machinery bay and sometimes include the inflow system in the engine bay. Differing radically from an ACV, only enough area of air inflow for the engine bay has to be regarded in order to avoid high velocity of air inflow and too large a pressure loss.

Noblocking objects or plates should be arranged in front of the fan air inlet in order to avoid inflow blockage, particularly in the case where the craft is running on rough seas, as the air inlet blockage will affect the air inflow and increase the loss of pressure at the air inlet. Consequently, it will affect the fan characteristic indirectly, making it steeper, which strengthens the cobblestone effect in the case where craft are running in short-crested waves.

To meet these requirements for the design of air outlet systems, a number of measures may be taken. They can be summarized as follows:

1. A minimum distance of the air outlet from the water surface has to be maintained in order to prevent the water causing damage to the fan impellers. It is best if some guide plates are arranged at the fan outlet which can act as green water deflectors, or the air duct from the outlet can be curved in order to prevent spray hitting the fan blades.
2. The area of the outlet is suggested to be widened as far as possible in order to avoid sudden air diffusion at the outlet, leading to a large pressure loss. A efficient diffuser shape also will flatten the fan air duct characteristic and reduce the cobblestoning effect.
3. Action should be taken to optimize the bow and stern bag cushion/pressure ratios and the flow distribution for the bow/stern bag and air cushion. It is difficult to give general recommendations. Definite conclusions can be determined according to the craft size, craft speed and requirements for seaworthiness.

   An SES may have a separate air feed to the stern skirt loops, or adjustable (during maintenance) feed holes into the loop. If the stiffness of this skirt is too high, problems may be encountered with acceleration through hump speed and cavitation or air injestion of the propulsors in heavier seas.

   It is normal for most of the cushion air to be fed through the bow bag or into the segments at their top, from holes positioned similarly to the optimum for feed from a loop above the segments.

12.4 Lift fan selection and design

The function of the lift fan is to provide:

- sufficient pressurized air to support the craft weight on the cushion;
- the required flow rate in order to maintain the design skirt tip air gap, and in addition meet the requirements due to wave pumping, heave motion pumping and the change of cushion air requirements as the craft moves over differing terrain;
- a suitable fan pressure/flow characteristic curve so that it will not stall during craft operation. The characteristic has to meet the slope of overpressure with respect to flow rate at the design point, and also to consider the characteristics at off design points, which will affect the seaworthiness at larger heaving amplitudes.

Before selecting and/or designing the fan(s), one first has to consider how to select the appropriate type, i.e. which type of fan will be suitable for the craft design, whether axial, mixed flow or centrifugal flow.

The considerations are introduced below, followed by a discussion of the design characteristics of each lift fan type.

Selection of fan type

There are three generic types of fans which may be used for ACVs and SESs. The centrifugal fan possesses the feature that high pressure head may be maintained at small flow rates. The axial fan produces a large flow and relatively low pressure head. Mixed flow fans have characteristics in between the centrifugal and axial fan types. Typical characteristics of these types of fan are shown in Fig. 12.11.

Each of the three types can be adopted for ACVs, as they normally have a cushion with low to medium pressure. SESs and air cushion platforms tend to operate with higher cushion pressures and so generally centrifugal or mixed flow fans are most appropriate. The application ranges defined by the specific speed of the fan at design flow (N_s) are

$$\begin{aligned} N_s &< 3 \quad \text{Suitable for centrifugal fans} \\ 2 < N_s &< 4 \quad \text{Suitable for mixed flow fans} \\ N_s &> 3 \quad \text{Suitable for axial fans} \end{aligned}$$

However, the selection of a lift fan is also strongly dependent upon the tradition and custom of the research institutes and manufacturers. For example, the former Soviet Union and France more often use the axial fan, while in the UK the centrifugal fan is used for ACVs except small recreational craft, and centrifugal or mixed fans for SES. In China it is usual to install an industrial centrifugal fan, partly due to its easy procurement.

The non-dimensional specific speed of a fan can be written as

$$N_s = nQ^{0.5}/(gH)^{0.75} \tag{12.9}$$

where N_s is the specific speed, H the overall pressure of the fan (metres of water head), Q the flow rate of the fan (m^3/s) and n the speed of the fan impeller (revolutions/s). Meanwhile the non-dimensional specific speed can also be written as

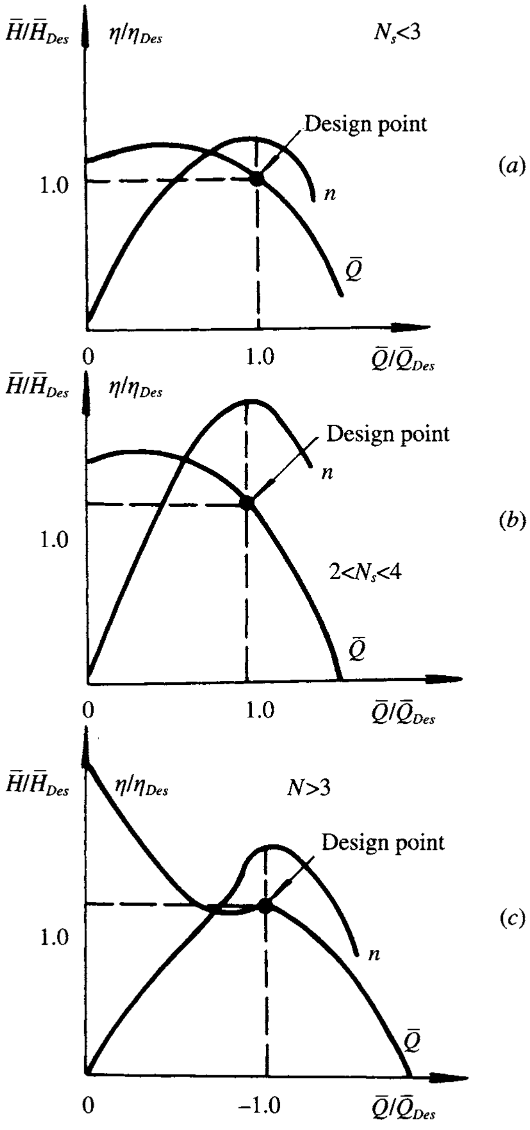


Fig. 12.11 Characteristics of three types of fans: (a) centrifugal fan with backward inclined blades; (b) mixed flow fan; (c) axial fan.

$$N_s = \bar{Q}^{0.5}/\bar{H}^{0.75} \tag{12.9a}$$

where $\bar{Q}$ is the flow rate coefficient

$$\bar{Q} = Q/nD^3$$

and $\bar{H}$ the pressure coefficient

$$\bar{H} = H/[\rho_a n^2 D^2]$$

where D is the diameter of the fan impeller (m), H the overall pressure head (N/m^2) and ρ_a the air density (Ns^2/m^4).

Considerations for off-design operation

Due to the flat fan pressure-inflow characteristic of a centrifugal fan, it will also be generally suitable for off-design operation. A typical centrifugal fan design is shown in Fig. 12.12.

Blade vibration may occur to an axial fan at low air flow rates due to the separation of flow at the blade surface. This causes a peak and hollow on the pressure/flow characteristic of the fan, i.e. $dH/dQ > 0$, which means that in this region the operation mode is unstable. This is the disadvantage of the axial fan. Certainly craft motions in a seaway can cause vibration and eventual fatigue to an axial fan blade.

Most small craft use this type of fan for their lift system, as their cushion pressure is relatively low (around 5–20 psf or 240–960 N/m^2). The fans used have polypropylene or nylon (PA-11) blades, which are relatively light. Extreme pressure fluctuations can cause the fan to burst, but due to the low mass of the blades, damage to the craft structure is normally minimal. In contrast, burst of a centrifugal fan is likely to be a major event and needs careful design of the surrounding ducting to prevent damage to the craft and personnel.

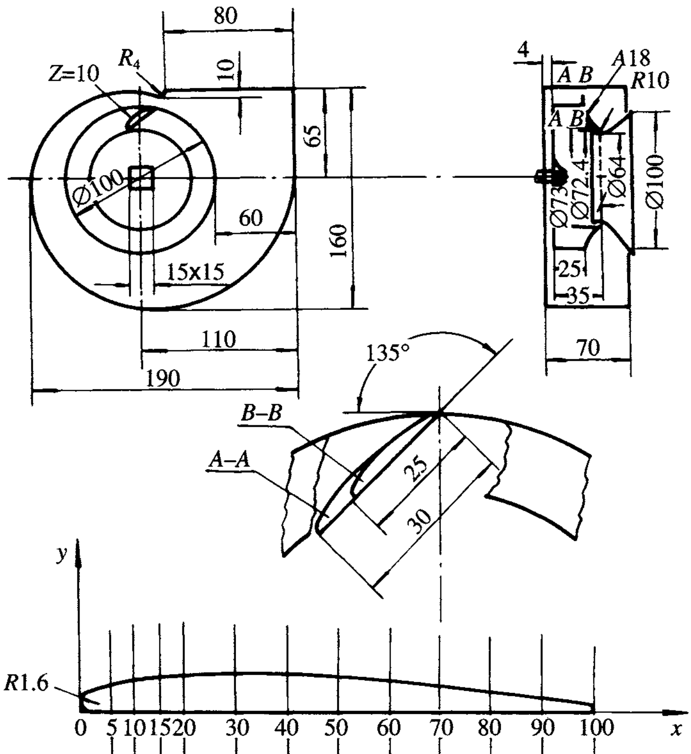


Fig. 12.12(a)

x	0	5	10	15	20	30	40	50	60	70	80	90	100
y	16	4.6	6	6.8	7.25	7.6	24	6.85	6.08	5.1	3.95	2.65	1.2

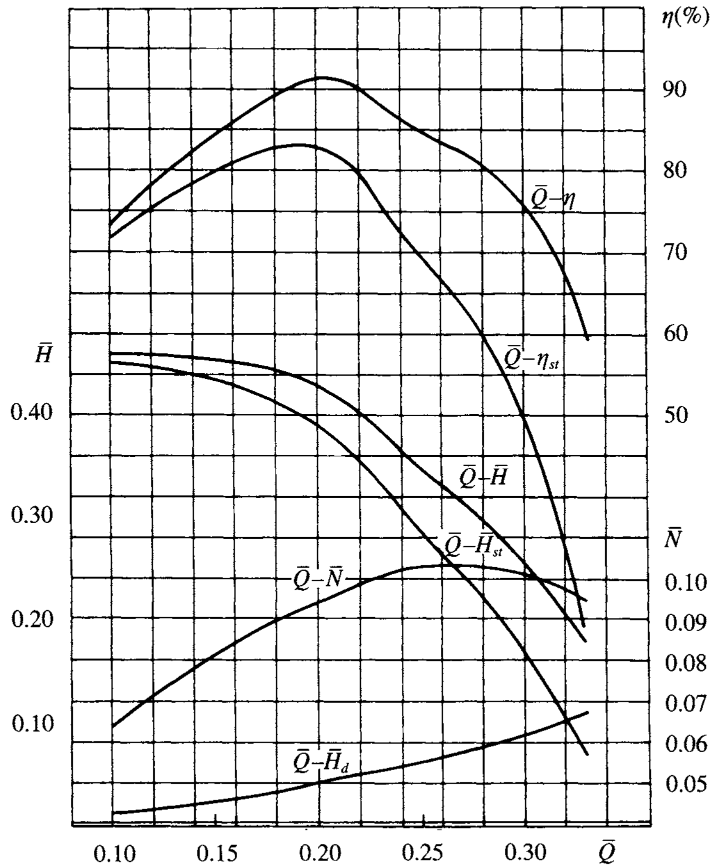


Fig. 12.12(b) Aerodynamic characteristics of Chinese centrifugal fan model 4-72, its configuration and streamlined blade offsets.

Fan choice

The principal choice for an ACV or SES designer is likely to be which commercially available fan type will be used. The major source of fans is the heating and ventilating industry. The first task therefore is to investigate what fans are locally available, whether direct from local manufacturers or from distributors, which are of a size which appears applicable to the initial craft design. Currently, fans of all three major types are available, though the ACV designer may have to uprate the fan, either by allowing increased stresses compared with building industry practice, or by building a completely new hub to hold the blades in the case of some axial fans.

Since the HVAC industry design considerations are normally related to system noise and efficiency, fans in HVAC systems are normally very lightly loaded. Use of higher specific speed can normally provide the requirements for an ACV. Care should be taken to select fans which are designed around aerofoil blades and not flat plate blades. Also steer away from blades cast in aluminium as these will be sensitive to vibration and fatigue.

A fan burst of a cast aluminium fan is generally very dangerous due to the large energy stored in the blades themselves. Steel fans (normally centrifugal, formed from plate) are generally too heavy for ACVs and select themselves out. Should a fan in either material be selected, then care must be taken to install a guard around the fans which can retain the energy of a failed blade. In the case of an axial fan, a mesh guard at the intake is required, and a duct with walls sufficient to absorb the energy. An aluminium sheet ring around the fan blade tips, or additional woven roving layers in the GRP ducting, will normally be sufficient. A centrifugal fan would require a steel plate ring around the volute.

GRP moulded blades can also be sensitive to fatigue and cannot be recommended for axial fans, though there have been several successful applications for centrifugal and mixed flow GRP fans. Attention has to be paid to finishing these fans, by sealing all surfaces with gel coat, so as to prevent water being drawn into the fan along glass strands and putting the fan out of balance.

Axial and centrifugal fan blades of PA-11 impregnated with short glass strands have been successfully applied. These blades are stiffer than PA-11 alone. They can shatter more easily than the basic polymer in temperatures below zero. Craft which are to be used in cold climates therefore need to use materials such as polypropylene and polyurethane.

After selection of fan type, the fan, air duct and air clearance characteristics have to be drawn out as shown in Fig. 12.13. Based on the non-dimensional characteristics and size of the selected fan as well as the fan speed, one can determine the dimensional characteristics of the fan, subtract the pressure loss in air ducting, then obtain the characteristic of the air cushion. The logic for this procedure is shown in a chart form as Fig. 12.14.

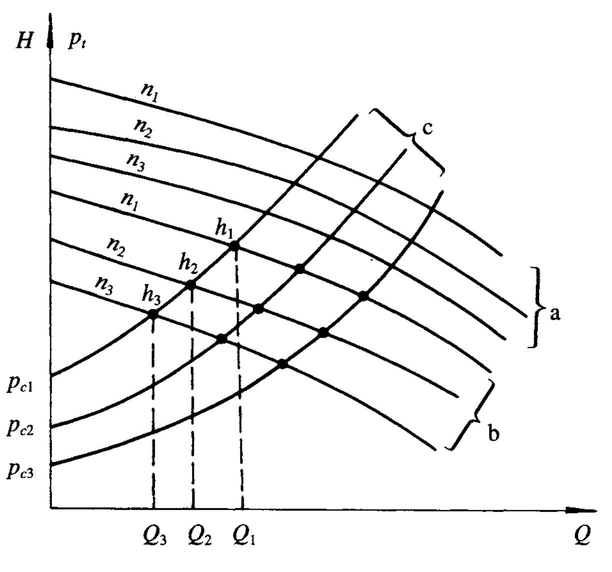


Fig. 12.13 Fan, air duct and air clearance characteristic curves of ACV/SES: (a) fan; (b) air duct; (c) air clearance (under-skirt gap).

Meanwhile, based on Chapter 2 and previous sections of this chapter, the relation can be obtained between the bag pressure and flow rate at different cushion pressures p_c, i.e. the characteristics related to differing air clearance. Thus from the intersection point of clearance and air duct characteristic curves, one may obtain the air clearance under the skirt (or flow rate) at different craft weight and engine speeds. This group of curves is very important for checking the performance of craft.

We introduce here the fan characteristics and configurations widely used in China, i.e. the characteristic of industrial fan model 4-73, 4-72 and modularized fan design. A comparison between foreign and Chinese fans is also made. Table 12.6 shows the fans mounted on some ACV/SES.

Modularized design of centrifugal fans

The industrial centrifugal fan models 4-73, 4-72, etc. have been widely used in China and a large amount of experimental results and data have been obtained which verify

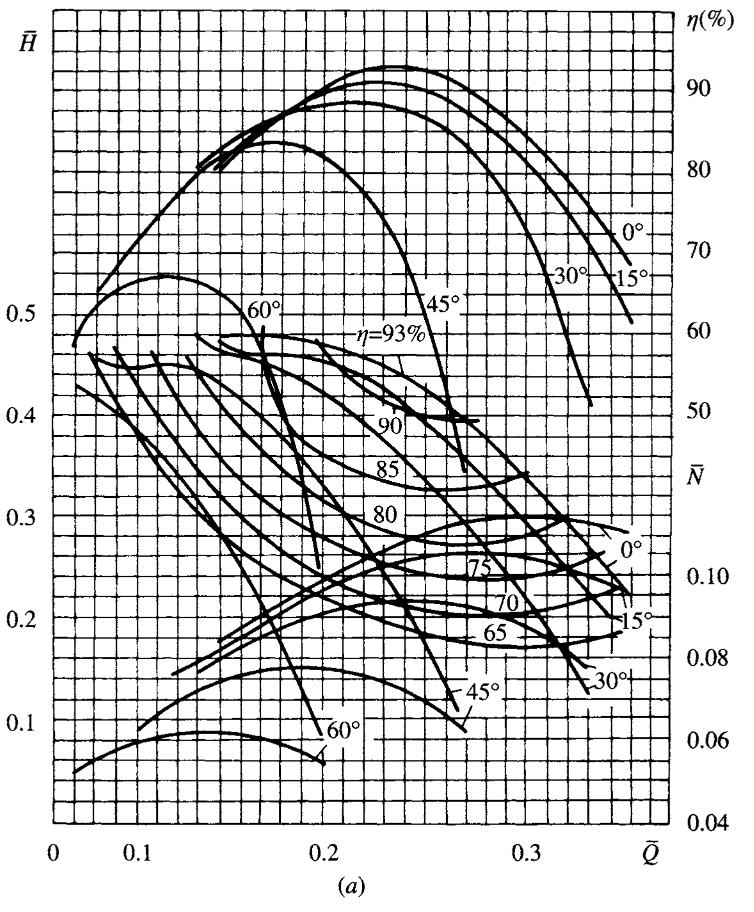
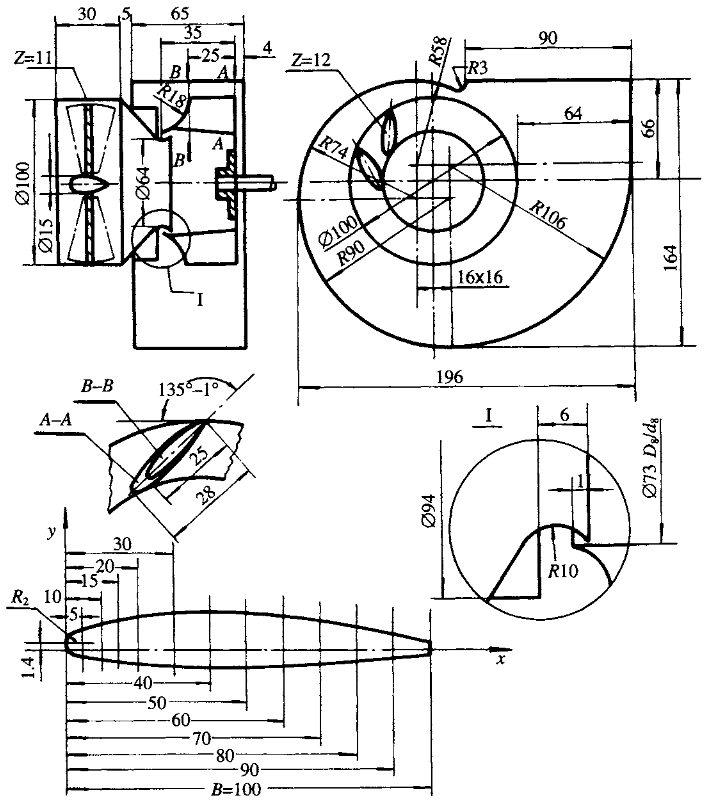


Fig. 12.14(a) Aerodynamics characteristics of Chinese centrifugal fan model 4-73, its configuration and streamlined blade offsets.



(b)

Fig. 12.14(b) Chinese fan model 4-73 geometric data.

x	5	10	15	20	30	40	50	60	70	80	90	100
y	4.6	6	6.8	7.25	7.6	7.4	6.85	6.08	5.1		2.65	1.2
$-y$	1.15	1.5	1.7	1.81	1.9	1.85	1.71	1.52	12.8	0.99	0.66	0.3

that the characteristics of these fans are suitable for the ACV and SES, so in general we apply the modularized design method and take the industrial centrifugal fan as the prototype to design ACV fans [97, 98]. Some fans, those on ACVs with high load density, or air cushion platforms with special requirements, are outside the range of such standard fan types. Then new fans have to be designed.

During the modularized design of a centrifugal fan, the following steps should be taken.

Selection of fan type by means of specific speed

The dimensional specific speed of lift fan can be written as follows:

$$N_s = n\, Q^{0.5}/H^{0.75} \qquad (12.10)$$

where n is the fan speed (r/min), Q the inflow rate of the fan (m^3/s) and H the overall pressure of the fan (kg/m^2). Thus the dimensional specific speed can be obtained according to the required Q, H and speed of fan. Then designers can select the characteristic curve of an available industrial fan and check to see if the design point is

Table 12.6(a) Lift fans mounted on some ACV/SESs [4] – basic data

Craft	Builder	Fans (No.)	Fan type	Tip diameter (m)	No. blades	Design speed (RPM)	Overall pressure (Pa)
VA-1	Dowty Rotol	2	Centrifugal	1.31	17	875	1250
VA-2	Dowty Rotol	2	Centrifugal	1.67	17	870	3400
VA-3	Dowty Rotol	2	Centrifugal	3.35	19	430	3112
HM.2	Hovermarine	5	Mixed Flow	0.61	11	2900	3351
HM.2			Centrifugal				2394
SKIP	General Dynamics	1	Axial	1.65			
SKMR-1	Bell Aerospace	4	Axial	1.98	10	1200	2973
VRC-1	British Vehicle Research Corporation	2	Centrifugal	0.99	8	1140	962
SES 100B	Bell Aerospace	8	Centrifugal	1.22	12	1700	5745
SES 100A	Aerojet General	3	Axial	1.21	19	2500	7804
SR.N6	BHC	1	Centrifugal	2.13	12	800	3591
SR.N4	BHC	4	Centrifugal	3.50	12	700	5745
N-500	Sedam	4	Axial	1.85	12	900	2968
		2	Axial	3.60	12		
VT.1	Vosper Thornycroft	8	Centrifugal	1.54	12	1050	
JEFF-A	Aerojet General	8	Centrifugal	1.22	12	2450	8139
Sormovich	Krasnoye Sormovo	1	Axial	2.74	12		4309
713	Shanghai HDSY	2	Centrifugal	1.0	12		4400
717C	MARIC	4	Centrifugal	0.6	12		4900
711-II	Shanghai HDSY	1	Centrifugal	1.8	12		2900

Table 12.6(b) Lift fans mounted on some ACV/SESs [4] – statistics

Craft	Builder	Flow rate (m^3/s)	Fan efficiency	Fan Shp kW/0.735	Specific speed (N_s)	Impeller weight (kg)	Cushion pressure (Pa)	Overall pressure (Pa)
VA-1	Dowty Rotol	22.6	0.86	55	2.43	27.6	820	1250
VA-2	Dowty Rotol	21.8	0.78	128	1.11	59.0	958	3400
VA-3	Dowty Rotol	75.6	0.79	400	1.09	304.0	1518	3112
HM.2	Hovermarine	5.61		30	2.34	8.2	2202	3351
HM.2		5.1						2394
SKIP	General Dynamics					28.5		
SKMR-1	Bell Aerospace	73.6	0.79		3.20		2250	2973
VRC-1	British Vehicle Research Corporation	8.86				58.1	814	962
SES 100A	Aerojet General	18.4	0.76		2.95		4549	5745
SES 100B	Bell Aerospace	66.0	0.68	195	1.35	95.2	4788	7804
SR.N6	BHC	75.0	0.83		1.82		1675	3591
SR.N4	BHC	113	0.75		1.38	680.0	2394	5745
N-500	Sedam	481	0.75		3.99			2968
VT.1	Vosper Thornycroft			175	1.75			
JEFF-A	Aerojet General	45.3	0.80	785	2.35	59.0	4596	8139
Sormovich	Krasnoye Sormovo	113					1963	4309
713	Shanghai HDSY	12.5	0.85	190			3120	4400
717C	MARIC	5	0.85	30			2500	4900
712-II	Shanghai HDSY	51.7	0.84	239			2160	2900

located at a high efficiency region. If not an alternative choice may be made and rechecked, as an iterative process.

Determining the type of fan, the non-dimensional flow and head as well as the fan efficiency at the design point can then be calculated (H, Q, η, etc.):

$$\bar{H} = H/[\rho_a u_2^2] = 3600H/[\rho_a \pi^2 n^2 D_2^2] \quad (12.11)$$

$$\bar{Q} = Q/Fu_2 = 240Q/[\pi^2 n D_2^2] \quad (12.12)$$

where F is the area of the fan impeller disc (m^2),

$$F = \pi/(4D_2^2)$$

u_2 is the circular velocity of the fan impeller (m/s),

$$u_2 = (\pi D_2 n)/60$$

and D is the impeller diameter (m). Meanwhile the power output of the fan can be obtained as

$$N_f = QH/[1000 \eta_f](kW)$$

where η_f is the fan efficiency.

The calculation mentioned above is suitable for selecting the fan type. During the calculation of circular velocity u_2, it is suggested that designers have to take the strength of the impeller blade and the noise of the fans into account. For the blades with an aerofoil profile, in general we take $80 < u_2 < 110$ m/s.

Selecting the impeller diameter

After selecting the fan type one can select the impeller diameter to position the design point of the fan according to the fan characteristic, required air inflow, overall pressure head of the fan and given fan speed. It may be noted that the actual operation points of a lift fan are not often situated at the design point.

In general only a small air flow is needed when the craft is running on calm water, in order to obtain the optimum craft running attitude. In the case where craft are operating in waves, captains often throttle up the lift engine in order to reduce the vertical acceleration of the craft, i.e. reduce the vertical motion and the wave pumping effect. For instance, the fan speed of SES model 713 operating on calm water is 1250 r.p.m., but 1400–1500 r.p.m. in waves.

As a general rule, fan flow rate increases in linear proportion to the speed, while pressure increases in square proportion and the power increases in cubic proportion; thus the flow rate of the fan in waves will increase 1.12 times, pressure increase 1.25 times and power increase 1.4 times, taking SES-713 as the example.

Craft weight is always nearly constant, so that the cushion pressure also approximately stays constant. The main change is fuel usage, making the craft gradually lighter. The operation point on the dimensional characteristic curve will therefore slip to the right-hand side of the curve, i.e. at larger inflow condition.

The operation points will in general not be located at the design point of curves, since this is normally set for calm water, or for a small sea state rather than the maximum. Therefore during the design of a lift fan, designers have to take off-design points into account to locate these off-design operation points also within the region of high efficiency, moreover at a flattening section of the H–Q characteristic curve so as to reduce vertical motion.

Designers can select several impeller diameters, D_2, to get the corresponding u_2, $\bar{Q}$, $\bar{H}$, then choose a suitable D_2 and consequently plot the operational characteristic curve of the fan. Using the characteristic of the fan, one can recalculate the characteristic curve of the air duct and compare this with the characteristic of air clearance, then the operation point at various craft weights and fan speeds can be obtained as shown in Fig. 12.13.

Figures 12.12 and 12.14 identify the fan configuration, aerodynamic characteristics and blade offsets for the streamline type of centrifugal fan models 4-73 and 4-72. One can carry out the design (selection) of fans based on these figures.

Technical issues to take into account for lift fan design and manufacture

Choice of impeller speed and diameter

It is very important to select the optimum speed and diameter of impellers. From the point of view of craft general arrangement, the impeller diameter should be decreased for higher craft design speeds, to minimize frontal area. However, the decrease of diameter will need to be compensated by an increase in the number of fans to produce the same airflow volume and their speed will have to be increased so as to support the required pressure head. For this reason, designers have to make a tradeoff between the number, diameter and speed of lift fans to select the most suitable combination.

Fan characteristics at low flow rate

Because the required flow of fans on hovercraft operating over calm water (particularly for SES) is small, i.e. at small Q and sometimes may be $\bar{Q} < 0.1$, complementary experiments with very small flow have to be carried out if unstable operation is suggested by the fan H/Q characteristic. From the point of view of safety and plough-in resistance of craft running in waves, it is suggested setting the pressure characteristic at low flow at twice that at the design point [94]. This cannot be obtained on many hovercraft, as this would require the fan to be operated too far down its efficiency curve and so a compromise must be reached.

Fan balancing

It is not enough to carry out static balancing of a fan. Owing to the wide impeller blades and the lower speed used in steady-state fan tests ($\approx$ 500 rev/min), it is important to check the fan balance at a range of speeds, if possible up to the operating conditions on the craft. MARIC have a lot of experience on this point. By not carrying out fan dynamic balancing carefully enough, some fans, shaft systems and air propellers have been damaged after a period of operation, causing the deterioration of equilibrium of rotating machines. For example:

1. By not carrying out dynamic balancing tests of fans for the craft model 719, fan vibration amplitude was very large at the speed of 1200 rev/min, causing hull vibration and alarm in crews and passengers.
2. With respect to the air impeller composed of GRP of the ACV model 722, the dynamic equilibrium of propeller was destroyed after a time of operation, because water and oil were absorbed into the air propeller blades non-uniformly, thus causing damage to propeller bearing mountings, etc.

Similar experience has been had with fans on craft in the UK through the 1970s and 1980s.

Installation of lift fan

Lift fans have to be mounted carefully according to the specified geometry between impeller and volute (Figs 12.13 and 12.14). Attention should be paid to the size of clearance between the air inlet and impeller (tip clearance and uniformity) as shown in Fig. 12.15. This has been proved in the test of fan model 4-72. For instance, the reduction of radial clearance from 0.5 to 0.3% will give an efficiency increase from 87 to 89% and efficiency will enhance to 91% if the clearance was decreased further to about 0.1%. This latter is probably not practical for fans on large craft.

Air flow rate for fans with double inlet

Due to the difficulty of arrangement of fans, sometimes two lift fans will be fitted on to one backplate to become a fan with a double inlet. To our knowledge, the flow will be 90% of the sum of flow of two fans, i.e. $Q_1 = 1.8Q_0$, where Q_1 denotes the flow of a double inlet fan and Q_0 denotes the flow of a single fan inlet and the corresponding overall pressure will stay unchanged, which was validated by a test of fans on SES model 713. This fan arrangement has also been successfully used on craft such as the VT.1, VT.2, AP1-88, LCAC, etc.

Noise reduction

In order to reduce fan system noise it is suggested to put isolation material on fan volutes. This has been tested on the SES model 719-II and gave good results.

Fan characteristic curves

As is mentioned above, it is better to install fans with a flat pressure–flow characteristic curve in order to get small heaving stiffness and damping, thus minimized motion of craft in waves, particularly the cobblestoning effect of craft running in short-crested waves at high craft speed.

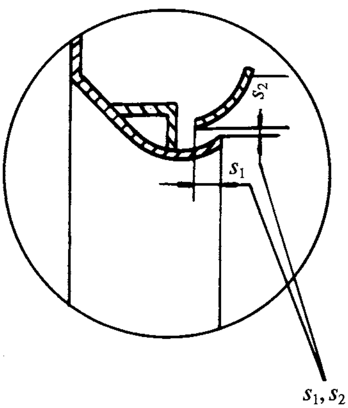


Fig. 12.15 Data for clearance between fan impellers and air inlet casing, which has to be considered carefully during installation of fans.

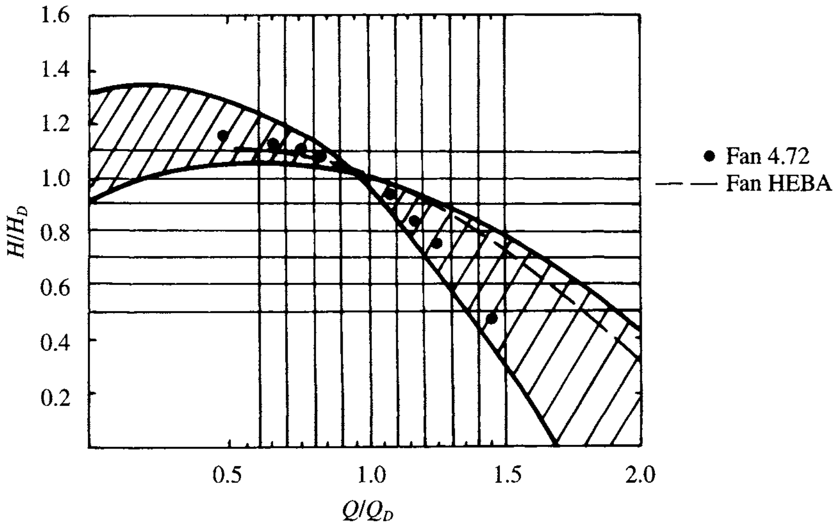


Fig. 12.16 Fan overall pressure-flow rate characteristics of fans used in China and abroad.

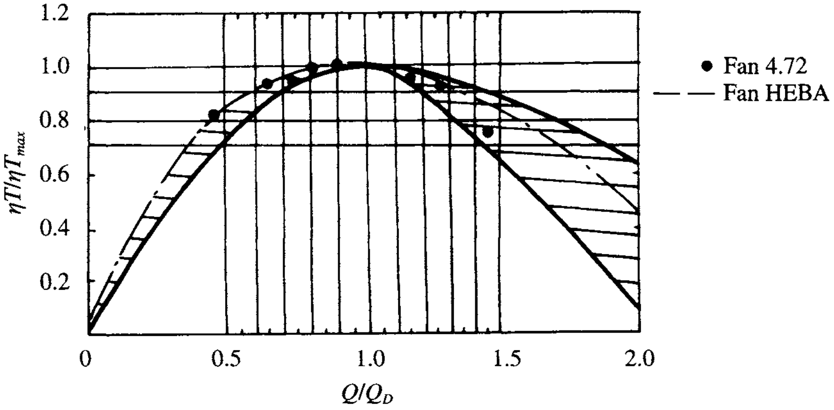


Fig. 12.17 Fan efficiency-flow rate characteristics of fans adopted in China and abroad.

Figures 12.15 and 12.16 [94] introduce the high efficiency HEBA fan which is widely used in Western countries for ACV/SES. It is surprising that the characteristic curves for these characteristics are so close to each other (the shaded area shown in the figures). The high efficiency fan type HEBA-B, as shown in the figures, is typical and shows the flat H–Q curve and η–Q curve.

Characteristic curves for Chinese manufactured industrial fans, which are used as the lift fans of ACV/SES, are shown in Figs 12.17 and 12.16 with black points. It can be seen that the overall pressure head/flow rate characteristics are rather steep and the proportion of overall pressure at the maximum efficiency region with the maximum overall pressure of the fan is about 0.83, but not 0.5, which leads to the following results:

1. Due to the steep characteristic at the region near the design point, heave stiffness and damping are larger, thus causing larger vertical acceleration which will be strongly sensitive to the 'cobblestone effect'.
2. Once the flow rate reduces, the craft has a lack of vertical restoring force and is easy to plough in.

The efficiency of fan models 4-72 and 4-73 is high and with a wide region for high efficiency, but the high efficiency fan (HEBA) will be better due to the aspects mentioned above.