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GUIDELINES FOR COMPARTMENT NOISE PREDICTION OF CRUISE SHIPS

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Brief Introduction

As the pearl of shipping industry, cruise ships are characterized by large volume, large passenger capacity, complex system, and high entertainment and comfort requirements. As an important indicator of comfort, compartment noise of cruise ships is of great significance to passengers' amusement experience. Compartment noise prediction of cruise ships has become a key technology in the design and construction of cruise ships.

In order to avoid taking necessary noise reduction remedial measures after the completion of cruise ship construction, and to reduce the impact on ship compartment arrangement, light weight and shipbuilding cost, and to efficiently build a cruise ship with low noise and high comfort, CCS has developed the Guidelines for compartment noise prediction of cruise ships.

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Chapter 1 GENERAL

1.1 General requirements

1.1.1In accordance with the noise comfort index stipulated in the Rules for Cruise ships, the guidelines develop the compartment noise prediction methods of cruise ships and the noise control measures for air conditioning ventilation system.

1.1.2 The Guidelines are applicable to the compartment noise prediction of sea-going cruise ships, as well as the noise prediction of ro-ro passenger ships, scientific research ships and other ship types.

1.1.3 Unless otherwise stated, the Guidelines only serve as the refinement and supplement for related conventions and rules and cannot replace any of them.

1.1.4 It is suggested that compartment noise prediction should be carried out at the design stage of cruise ships, and appropriate vibration and noise reduction measures should be taken, so as to avoid taking remedial measures after the completion of construction.

1.1.5 The Guidelines only provide the compartment noise prediction method at the design stage of cruise ships, and the final measured results are to prevail.

1.2 Plans and documents

1.2.1 The following information is at least to be provided for the compartment noise prediction of cruise ships:

(1) General arrangement;
(2) Loading manual;
(3) Cabin arrangement plan;
(4) Construction profile;
(5) Insulation arrangement plan;
(6) Deck covering arrangement plan;
(7) Equipment arrangement plan;
(8) Engine room arrangement;
(9) Interior sound insulation material parameter list;
(10) Machinery parameter list;
(11) Machinery noise excitation spectrum table
(12) Whole ship air conditioner room arrangement plan;
(13) Whole ship air conditioner duct arrangement plan;
(14) Mechanical ventilation arrangement plan;
(15) Other plans and documents as deemed necessary by CCS

1.3 Types of cruise compartments

1.3.1 Cruise compartments are sorted in accordance with Table 1.3.1.

Types of cruise compartments Table 1.3.1

Type of compartment	Location of compartment	Remarks
Passenger space	Passenger cabin	
	Passenger public space	Generally referring to the recreational and

		leisure cabin of cruise passengers, including restaurants, gymnasiums, dance halls, bars, multi-function halls, cinemas, banquet halls, galleries, supermarkets, grocery stores, libraries, card rooms, garden lounges, bandstands, beauty salons, massage rooms, smoking rooms, etc.
	Passenger open deck recreation area	Generally including open air swimming pools, fairgrounds and other areas.
Crew space	Crew cabin	
	Crew public space	Generally referring to the cabins in the crew public space, including officer mess rooms, crew mess rooms, conference rooms, officer recreation rooms, crew recreation rooms, etc.
	Crew work area	Generally refers to crew work cabins, including various types of offices, duty rooms, etc.
Equipment control room	Equipment control room	
Machinery Space	Machinery Space	
Medical space	Medical space	

1.3.2 The compartment noise limits of cruise ships are to comply with the requirements of 4.6.2 of Chapter 4 of the Rules for Cruise ships. Cruise ships applying for CCS class notation of comfort are to comply with the requirements of 5.3.4 of Chapter 5 of the Rules for Cruise ships.

1.4 Definitions and terms

1.4.1 Sound pressure level L_p: 20 times the logarithm to the base 10 of the ratio of the sound pressure to a reference sound pressure, expressed in decibels (dB), given by the following equation:

$$L_p = 20\lg\frac{p}{p_0}$$

where: p— the sound pressure, in Pa;

p_0— the reference sound pressure, being 20 μPa in air.

1.4.2 Sound power: Sound energy which passes through an area perpendicular to the direction of sound transmission per time unit, expressed in watts (W).

1.4.3 Sound power level: 10 times the logarithm to the base 10 of the ratio of the sound power to a reference sound power, expressed in decibels (dB), given by the following equation:

$$L_w = 10\lg\frac{w}{w_0}$$

where: L_w— sound power level, in dB;

w— sound power, in W;

w_0— reference sound power, in W, being 1×10^{-12}W in air.

1.4.4 A-weighted sound pressure level: The quantity measured by a sound level meter in which the frequency response is weighted according to the A-weighting curve.

1.4.5 Fan specific sound power level: The sound power level generated by the fan at unit air volume (m^3/min) and unit air pressure (Pa), in decibels (dB).

1.4.6 Sound absorption coefficient: The ratio of the absorbed sound energy to the incident one, given by the following equation:

$$\alpha = \frac{W_r}{W_i}$$

where: W_i— incident sound energy;

W_r— absorbed sound energy.

1.4.7 Acceleration level L_a : 20 times the logarithm to the base 10 of the ratio of the acceleration to a reference acceleration, expressed in decibels (dB), given by the following equation:

$$L_a = 20\lg\frac{a}{a_0}$$

where: L_a— acceleration level, in dB;

a— effective value of acceleration, in m/s^2;

a_0— reference acceleration, in m/s^2, generally taken as $1 \times 10^{-6} m/s^2$.

1.4.8 Velocity level: 20 times the logarithm to the base 10 of the ratio of the velocity to a reference velocity, expressed in decibels (dB), given by the following equation:

$$L_v = 20\lg\frac{v}{v_0}$$

where: L_v— velocity level, in dB;

v— effective value of velocity, in m/s;

v_0— reference velocity, in m/s, generally taken as 1×10^{-9}m/s.

1.4.9 Airborne noise: Noise transmitted from the interaction between operating machineries and power equipment and air or airflow movement into air.

1.4.10 Structure-borne noise: The noise generated by vibration and transmission of solid components of a cruise ship (e.g. machinery bases, hull structures, ventilation ducts, ect.) under excitation conditions. The noise volume is usually expressed in velocity levels or acceleration levels.

1.4.11 Mechanical impedance: Plural ratio of excitation force to speed response, in N · s/m.

1.4.12 Natural attenuation of noise: In the transmission process of the noise of ventilation and air conditioning system, part of the sound energy is converted into heat energy due to the friction of airflow and duct wall and part of the sound energy is reflected back to the source due to the change of duct section and structure, so that the attenuation of noise is caused, which is expressed in decibels (dB).

1.4.13 Sound insulation: The difference between the incident sound power level at one side of a bulkhead or any other acoustic outfitting component and the transmitted one at the other side, given by the following equation:

$$R = 10\lg\frac{W_i}{W_t}$$

where: R — sound insulation, in dB;

W_i — total sound energy;

W_t — transmitted sound energy.

Chapter 2 NOISE SOURCES AND TRANSMISSION PATHS OF CRUISE SHIP COMPARTMENTS

2.1 Main noise source of cruise ship compartments

2.1.1 Sound sources of mechanical equipment, including diesel engine, propulsion motor, generator, thruster , air conditioning system, fan, boiler, air compressor, pump, etc.;

2.1.2 Sound source of entertainment equipment, including entertainment facilities, fitness and amusement equipment at dance halls, cinemas, open air entertainment area and other spaces;

2.1.3 Propulsion system sound source, which is mainly propeller pulsation pressure;

2.1.4 Aerodynamic sound sources, including power equipment inlet and outlet, ventilation system inlet and outlet as well as eddy, etc.;

2.1.5 External sound sources, mainly noise due to docking and cargo handling;

2.1.6 Other sound sources, such as broadcast, whistle, footfall of crew and passengers as well as sound of door opening and closing.

2.1.7 The Guidelines only involve cruise ship-related noise sources, such as noise of mechanical equipment, entertainment equipment, propulsion system and aerodynamic sound source, excluding wind, wave and ice noise, alarm and public address system, as shown in Figure 2.1.7.

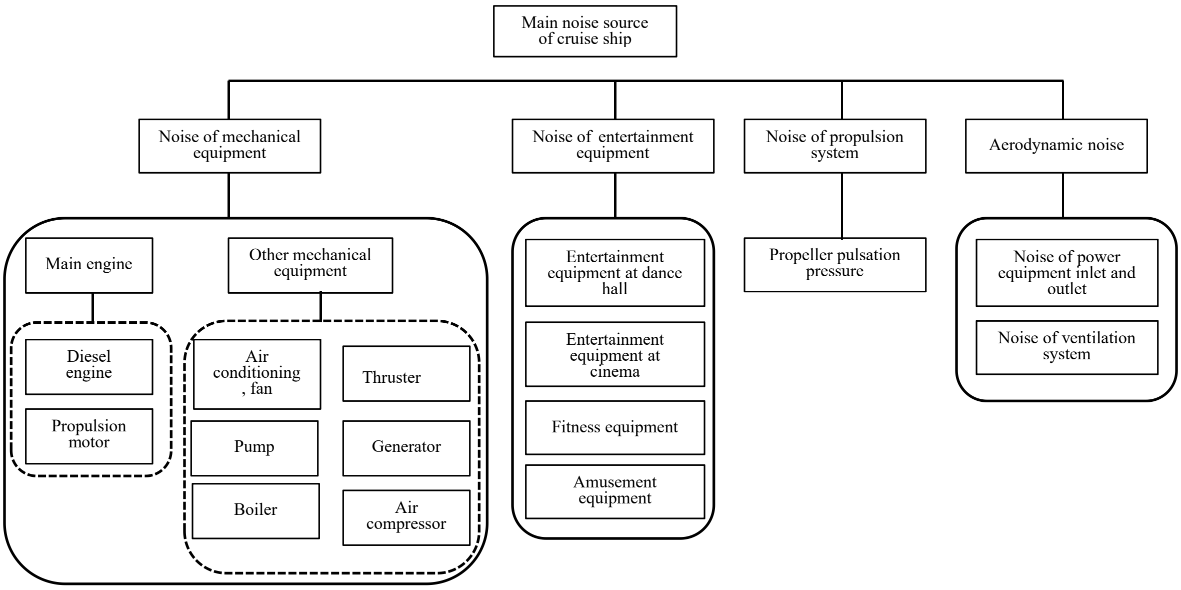


Figure 2.1.7 Main noise source of cruise ship compartments

2.1.8 The main noise sources of cruise cabins are to take the measured data of equipment structure-borne noise and airborne radiation noise as the input data. If there is no measured result, it can also be calculated in accordance with the empirical formula in 2.4.

2.2 Compartment noise transmission paths

2.2.1 Noise on cruise ships is mainly transmitted by means of air medium and hull structure and spread as airborne radiation noise and structure-borne noise. Paths of noise transmission on board ships are shown in Figure 2.2.1.

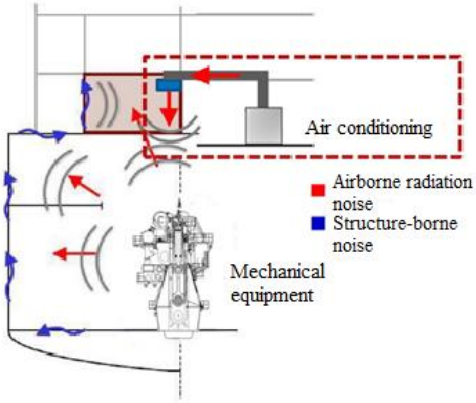


Figure 2.2.1 Diagram of noise transmission paths on board ships

2.2.2 Airborne radiation noise

2.2.2.1 Airborne noise of mechanical equipment is mainly spread in compartments near noise source or with noise source (e.g. engine room, air conditioning room) by following two means:

(1) Outward radiation by penetrating wall plate;

(2) Spreading to other compartments through all possible means and openings (e.g. seam, stairway, funnel, door and window, etc.).

2.2.2.2 The ventilation noise of the cruise ducts is mainly generated by the air conditioning ventilation system and transmitted to each cabin through the ventilation ducts and various components.

2.2.3 Structure-borne noise is mainly transmitted to hull structure from sound sources such as engines by means of foundation vibration or the structure vibration of the outer bottom plating caused by the propeller excitation.

2.3 Noise source of typical compartments

2.3.1 Except for the special compartments, the noise generated by air conditioning ventilation system is one of the main noise sources in all compartments, the frequency of which is concentrated in the medium and high frequencies. The structure-borne noise of the main engine and other equipment is concentrated in the medium and low frequencies while the airborne noise is concentrated in the medium and high frequencies. Noise source of typical compartments are shown in Table 2.3.1.

Noise source of typical compartments **Table 2.3.1**

Type of compartment	Location of compartment	Noise source
Passenger space	Passenger cabin	Mainly including ventilation noise, running noise of electrical equipment and noise transmitted to the cabin by machinery equipment.
	Passenger public space	The main noise sources of gymnasium include the running noise of ventilation equipment, air conditioning equipment and fitness equipment. The main sound source of dance hall, bar, multi-function hall, cinema includes the running noise

		of stereo equipment. The main sound source of gallery, supermarket, grocery store, library, card room, garden lounge, concert hall, beauty salon, massage room, smoking room and other public spaces includes the noise transmitted to the cabin by machinery equipment.
	Passenger open deck recreation area	Mainly including the running noise of entertainment equipment, such as running noise of karting, bumper cars, amusement park, water park, deck surfing and other entertainment facilities.
Crew space	Crew cabin	Mainly including ventilation noise, running noise of electrical equipment and noise transmitted to the cabin by machinery equipment.
	Crew public space	Mainly including the noise transmitted to the cabin by ventilation equipment and machinery equipment.
	Crew work area	Mainly including the running noise of the internal equipment in all service spaces and the noise transmitted to the cabin by machinery equipment.
Equipment control room	Equipment control room	Mainly including the noise from the equipment in the control room and the noise transmitted to the cabin by machinery equipment.
Machinery Space	Machinery space	Mainly including the noise produced by the machine itself.
Medical space	Medical space	Mainly including the noise transmitted to the cabin by machinery equipment.

2.4 Calculation method of noise source

2.4.1 Calculation of excitation of bottom plating above propeller

2.4.1.1 Acceleration level of bottom plating above propeller caused by propeller excitation is to be calculated according to the following formula:

$$L_a = 10\lg(MN) + 40\lg D + 30\lg n_e + 25$$

where: L_a — acceleration level of bottom plating above propeller, in dB, reference acceleration $a_0 = 10^{-6}$ m/s^2;

M— quantity of propellers;

N— quantity of propeller blades;

D— diameter of propeller, in m;

n_e— rated rotational speed of propeller, in r/min.

2.4.2 Calculation of diesel engine noise

2.4.2.1 Acceleration level of foot of medium and high speed diesel engines:

$$L_a = -20\lg m + 20\lg P_e + 30\lg\frac{n}{n_e} + 140 + C_a$$

where: L_a — acceleration level of diesel engine, in dB, reference acceleration $a_0 = 10^{-6}$ m/s^2;

m— mass of diesel engine, in kg;

P_e— rated power of diesel engine, in kW;

n_e— rated rotational speed of diesel engine, in r/min;

n— working speed of diesel engine, in r/min;

C_a— octave band correction for vibration of diesel engine, in dB, see Table 2.4.2.1.

Octave band correction for vibration of diesel engine Table 2.4.2.1

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Correction (dB)	5	11	16	21	27	29	27	22

2.4.2.2 Where the diesel engine is equipped with vibration isolation device, the correction of vibration isolation device is to be considered.

2.4.2.3 Radiation sound power level of diesel engine is to be calculated according to the following formula:

$$L_w = 10\lg P_e + 58 + C_w$$

where: L_a — radiation sound power level of diesel engine, in dB, reference sound power $w_0 = 10^{-12}W$;

P_e— rated power of diesel engine, in kW;

C_w— octave band correction for airborne noise of diesel engine, in dB, see Table 2.4.2.3.

Octave band correction for airborne noise of diesel engine Table 2.4.2.3

Octave band center frequency (Hz)	With blower	63	125	250	500	1000	2000	4000	8000
Below 600r/min	Yes	21	27	28	26	24	20	13	4
Below 600r/min	No	18	24	25	23	21	17	10	1
600-1500r/min	Yes	20	17	22	33	31	25	20	9
600-1500r/min	No	24	26	24	26	25	23	19	13
1500r/min and above	Yes	24	31	31	30	32	30	23	16
1500r/min and above	No	21	28	28	27	29	27	20	13

2.4.2.4 The sound power level of exhaust noise of diesel engine is to be calculated according to the following formula:

$$L_w = 77 + 10\lg\left(P_e \times n_e\right) + 30\lg\left(\frac{n}{n_e}\right) + 10\lg\left\{1\Big/\left[\left(\frac{f_r}{f}\right)^3 + \left(\frac{f}{f_r}\right)\right]\right\}$$

where: L_a — sound power level of exhaust noise of diesel engine, in dB, reference sound power $w_0 = 10^{-12}W$;

f_r— firing rate, to be calculated according to the following formula:

$$f_r = N_c \times n \times 2 \Big/ (60 \times S)$$

f — octave band center frequency, in Hz;

N — number of cylinders;

S — number of strokes.

2.4.3 Electric machines

2.4.3.1 Calculation of acceleration level of electric machines

$$L_a = 10\lg P_e + 7\lg n_e + 62 + C_a$$

where: L_a— acceleration level of electric machine, in dB, reference acceleration $a_0 = 10^{-6}$ m/s^2;

P_e— rated power of electric machine, in kW;

n_e— rated rotational speed of electric machine, in r/min;

C_a— octave band correction for vibration of electric machine, in dB, see Table 2.4.3.1.

Octave band correction for vibration of electric machines Table 2.4.3.1

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Correction (dB)	11	14	14	16	17	18	18	18

2.4.3.2 Calculation of radiated sound power level of electric machines

$$L_w = 13\lg P_e + 15\lg n_e + 6.6 + C_w$$

where: L_w—radiated sound power level of electric machine, in dB, reference sound power w_0 = 10^{-12} W;

P_e— rated power of electric machine, in kW;

C_w— octave band correction for airborne noise of electric machine, in dB, see Table 2.4.3.2.

Octave band correction for airborne noise of electric machines Table 2.4.3.2

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Correction for speed above 600 r/min (dB)	6	10	14	15	15	14	8	1
Correction for speed below 600 r/min (dB)	0	5	10	15	15	14	8	1

2.4.4 Air conditioning ventilation system

2.4.4.1 For acceleration level of air conditioning equipment, see Table 2.4.4.1, reference acceleration $a_0 = 10^{-6}$ m/s^2.

Acceleration level of air conditioning equipment Table 2.4.4.1

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Acceleration level (dB)	100	110	105	105	110	115	115	120

2.4.4.2 For radiated sound power level of air conditioning equipment, see Table 2.4.4.2, reference acceleration $w_0 = 10^{-12}$ W.

Radiated sound power level of air conditioning equipment Table 2.4.4.2

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Radiated sound power level (dB)	108	108	112	110	101	100	95	95

2.4.4.3 Calculation of sound power level of fans

(1) The sound power level is to be calculated from the fan specific sound power level:

$$L_w = L_{wc} + 10\lg(QH^2) - 20$$

where: L_{wc}—fan specific sound power level, in dB. In the absence of specific sound power level information, 24dB can be taken for medium and low pressure centrifugal fans;

Q— air volume, in m^3/h;

H— total pressure, in Pa.

(2) The sound power level is to be calculated from the fan power:

① Low pressure fans: $L_w = 91 + 10\lg(P)$

② Medium and high pressure fans: $L_w = 87 + 10\lg(P) + 10\lg(H) - 10$

where: P— power of fan, in kW;

H— total pressure, in Pa.

(3) The sound power level is to be calculated from the average sound pressure level of the fan:

The average sound pressure level of the fan $\overline{L}_p$ is to be measured under certain sound field conditions, and the sound power level of the fan is to be obtained in accordance with different sound field conditions.

① Free sound field conditions: $L_w = \overline{L}_p + 20\lg(r) + 11$

② Semi-free sound field conditions: $L_w = \overline{L}_p + 20\lg(r) + 8$

where: r— distance between measuring point and fan sound source, in m;

③ Reverberation sound field conditions: $L_w = \overline{L}_p + 10\lg(V) - 10\lg(T) - 14$

where: V— volume of measurement reverberation chamber, in m^3;

T— reverberation time of measurement reverberation chamber, in s.

④ Conditions of sound field in duct: $L_w = \overline{L}_p + 10\lg(S) + \frac{5.8}{H}\sqrt{\frac{T}{293}}$

where: S— sectional area of measurement duct, in m^2;

T— thermodynamic temperature, in K;

H— total pressure in duct, in Pa.

⑤ General indoor sound field conditions: $L_w = \overline{L}_p - 10\lg\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right)$

where: Q— directivity factor of sound source;

r— test distance, in m;

R— test room constant, $R = \frac{S\overline{\alpha}}{1-\overline{\alpha}}$。.

(4) The sound power level of axial fan can be calculated according to the following formula:

$$L_w = 19 + 10\lg(Q) + 25\lg(H) + S$$

where: Q— air volume, in m^3/h;

H— total pressure, in Pa;

S — correction for working condition, determined by fan efficiency, vane number and vane angle, see Table 2.4.4.3(1).

Correction for sound power levels of axial flow fan under different conditions (dB)

Table 2.4.4.3(1)

Vane number z	Vane angle θ (°)	Q/Qm						
		0.4	0.6	0.8	0.9	1.0	1.1	1.2
4	15	—	3.4	3.2	2.7	2.0	2.3	4.6
8	15	-3.4	5.0	5.0	4.8	5.2	7.4	10.6
4	20	-1.4	-2.5	-4.5	-5.2	-2.4	1.4	3.0
8	20	4.0	2.5	1.8	1.9	2.2	3.0	—

4	25	4.5	2.0	1.6	2.0	2.0	4.0	—
8	25	9.0	8.0	6.4	6.2	8.0	6.4	—
Note: Qm is the air volume of the highest efficiency point, generally taken as Q/Qm=1.								

(5) For calculation of frequency band sound power level of fan, the sound power level of each frequency band is to be approximately calculated according to the following formula:

$$L_{wf} = L_w + \Delta b$$

where: Δb — correction for sound power level of each frequency band, in dB, see Table 2.4.4.3(2).

Correction for sound power level of each frequency band Δb (dB) Table 2.4.4.3(2)

Center frequency (Hz) / Type of fan	63	125	250	500	1000	2000	4000	8000
Centrifugal fan (front curved vane)	-2	-7	-12	-17	-22	-27	-32	-37
Centrifugal fan (back curved vane)	-5	-6	-7	-12	-17	-22	-26	-33
Axial flow fan	-9	-8	-7	-7	-8	-10	-14	-18

2.4.4 Reduction gearboxes

2.4.4.1 Radiated sound power level of reduction gearbox:

$$L_w = 68 + 10\lg(P_e) - 10\lg\left[\frac{fg}{g} + \left(\frac{f}{fg}\right)^2\right] + \Delta$$

where: L_w— radiated sound power level of reduction gearbox, in dB, reference sound power w_0 = 10^{-12} W;

f— rotational speed frequency of gear;

g— number of gear teeth;

Δ— influencing coefficient of manufacturing error of gearbox, see Table 2.4.4.

Influencing coefficient of manufacturing error of gearbox Table 2.4.4.1

Grade	B3	C1	C2	C3	D1	D2	D3
Δ	0	2.5	5	7.5	10	12.5	15

2.4.5 Boilers

For radiated sound power level of boiler, see Table 2.4.5, with reference sound power being w_0 = 10^{-12} W.

Radiated sound power level of boiler Table 2.4.5

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Radiated sound power level (dB)	96	97	94	92	92	85	83	85

2.4.6 Pumps

2.4.6.1 Calculation of acceleration level of non-reciprocating pump

$$L_a = 10\lg P_e + 81 + C_a$$

where: L_a— acceleration level of pump, in dB, reference acceleration $a_0 = 10^{-6}$ m/s^2;

P_e— rated power of mechanical equipment driving pump, in kW;

C_a—octave band correction for vibration of non-reciprocating pump, in dB, see Table 2.4.6.1.

Octave band correction for vibration of non-reciprocating pump Table 2.4.6.1

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000

Correction for centrifugal pump (dB)	8	21	19	23	24	20	24	23
Correction for gear pump (dB)	21	34	32	37	38	34	44	45

2.4.6.2 Calculation of acceleration level of reciprocating-type piston pump

$$L_a = 10P_e + 30\lg\frac{P_p}{3000} - 34 + C_a$$

where: L_a— acceleration level of pump, in dB, reference acceleration $a_0 = 10^{-6}$ m/s^2;

P_e— rated power of mechanical equipment driving pump, in kW;

P_p— rated pressure of pump unit, in Pa;

C_a—octave band correction for vibration of reciprocating-type piston pump, in dB, see Table 2.4.6.2.

Octave band correction for vibration of reciprocating piston pump Table 2.4.6.2

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Correction (dB)	7	10	9	7	10	7	4	7

2.4.6.3 Calculation of radiated sound power level of non-reciprocating pump

$$L_w = 10P_e + 15\lg n_e + 16 + C_w$$

where: L_w—radiated sound power level of pump, in dB, reference sound power w_0=10^{-12}W;

P_e— rated power of mechanical equipment driving pump, in kW;

n_e— rated rotational speed of pump, in r/min;

C_w— octave band correction for airborne noise of non-reciprocating pump, in dB, see Table 2.4.6.3.

Octave band correction for airborne noise of non-reciprocating pump Table 2.4.6.3

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Correction for centrifugal pump (dB)	25	26	26	27	29	26	23	18
Correction for gear pump (dB)	35	36	36	37	39	36	33	28

2.4.6.4 Calculation of radiated sound power level of reciprocating piston pump

$$L_w = 30\lg\frac{P_p}{3000} - 40 + C_w$$

where: L_w—radiated sound power level of pump, in dB, reference sound power w_0=10^{-12}W;

P_p— rated pressure of pump unit, in Pa;

C_w— octave band correction for airborne noise of reciprocating piston pump, in dB, see Table 2.4.6.4.

Octave band correction for airborne noise of reciprocating piston pump Table 2.4.6.4

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Correction (dB)	11	15	21	29	25	22	15	9

2.4.7 Air compressors

2.4.7.1 Calculation of acceleration level of air compressor

$$L_a = 10\lg P_e + 102 + C_a$$

where: L_a— acceleration level of air compressor, in dB, reference acceleration $a_0 = 10^{-6}$ m/s^2;

P_e— rated power of air compressor, in kW;

C_a— octave band correction for vibration of air compressor, in dB, see Table 2.4.7.1.

Octave band correction for vibration of air compressor Table 2.4.7.1

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Correction (dB)	7	10	9	7	10	7	4	7

2.4.7.2 For radiated sound power level of air compressor, see Table 2.4.7.2(1) and Table 2.4.7.2(2), reference sound power $w_0 = 10^{-12}$ W.

Radiated sound power level of reciprocating air compressor Table 2.4.7.2(1)

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Radiated sound power level (dB)	108	108	112	110	101	100	95	95

Radiated sound power level of centrifugal air compressor Table 2.4.7.2(2)

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Below 7.5kW (dB)	95	98	102	102	93	92	85	82
7.5kW~75kW (dB)	100	102	107	107	98	97	90	87
Above 75kW (dB)	105	108	112	112	108	102	95	92

Chapter 3 NOISE PREDICTION FOR COMPARTMENTS

3.1 General requirements

3.1.1 In order to calculate compartment noise on cruise ships and evaluate feasibility and effectiveness of measures for damping vibration and attenuating noise, this Chapter specifies noise prediction methods for compartments at different design stages of cruise ships.

3.1.2 Frequently used compartment noise prediction methods

3.1.2.1 Sound power loss and sound pressure level of compartments are to be evaluated based on simplified empirical evaluation formulas and by analyzing acoustic route from specific ship compartment to sound source.

3.1.2.2 For numerical prediction, the finite element method or other methods such as boundary element method is to be used for low frequency domain and the statistical energy method is to be used for medium and high frequency domain.

3.1.2.3 The database method based on a parent ship applies to those cases with a large amount of measured data of parent ships and parent equipment.

3.2 Noise prediction based on empirical formulas

3.2.1 General requirements

3.2.1.1 The prediction method based on empirical formulas applies to calculation of compartment noise on cruise ships at initial design stage.

3.2.1.2 In the noise prediction based on empirical formulas, characteristics of airborne noise and structure-borne noise of main noise sources are to be provided by the designer. If not available, calculation may be carried out by referring to the empirical formula in 2.4 of Chapter 2.

3.2.1.3 Calculation process is shown in Figure 3.2.1.3.

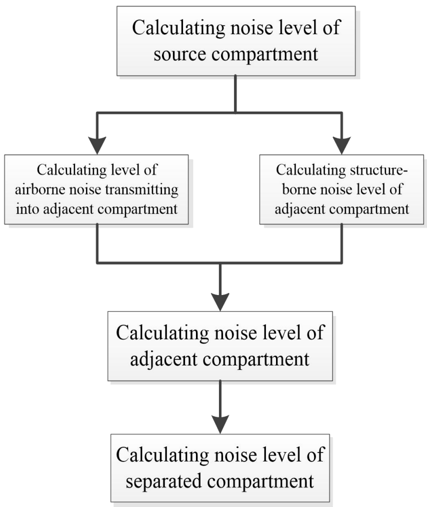


Figure 3.2.1.3 Prediction process based on empirical formulas

3.2.2 Noise prediction based on empirical formulas

3.2.2.1 The method using empirical formulas is to calculate compartment noise caused by each main noise source and combines all noise components to get the total airborne noise level of the compartment.

3.2.2.2 Noise level in source compartment is to be calculated according to the following formula:

$$L_A^{(1)} = L_w + 10\lg\left(\frac{Q}{4\pi r^2} + \frac{4\left(1-\alpha^{(1)}\right)}{S\alpha^{(1)}}\right)$$

where: $L_A^{(1)}$ — noise level of source compartment, in dB;

L_w — sound power level of sound source, in dB;

Q — directivity factor of sound source, Q = 1 for sound source located at the geometrical center of the compartment; Q = 2 for sound source located at floor center or on a wall of the compartment; Q = 4 for sound source located at midpoint of a sideline of the compartment; Q = 8 for sound source located at a corner of the compartment;

r — distance from compartment center to sound source, in m;

$\alpha^{(1)}$ — sound absorption coefficient of source compartment;

S — total area of interior walls of source compartment, in m^2.

3.2.2.3 The level of airborne noise transmitting into an adjacent compartment is to be calculated according to the following formula:

$$L_A^{(2)} = L_{AP}^{(1)} - R + 10\lg\frac{F}{A} + \Delta R$$

where: $L_{AP}^{(1)}$ — noise level at center of enclosure of source compartment, in dB; to be determined in accordance with 3.2.2.2;

R — sound insulation of bulkhead, in dB;

F — area of bulkhead, in m^2;

A — total sound absorption area in the room, in m^2;

ΔR — correction considering acoustic covering on the bulkhead.

ΔR is related to structural type of acoustic covering;

① ΔR = 10 dB for rigid installation;

② ΔR = 5 dB when sound absorbing layer is installed;

③ ΔR = 5 dB when effective vibration isolation parts are installed;

④ ΔR = 2 dB when vibration isolation parts and sound absorbing layer are installed.

3.2.2.4 The structure-borne noise level in the adjacent compartment is to be calculated according to the following formula:

$$L_S^{(2)} = N_{\lambda 0} - 15\lg\frac{S_1^{(2)} + S_2^{(2)}}{2} + 10\lg\left(1 - 0.9\frac{F_3^{(2)}}{F_{\mathrm{II}}^{(2)}}\right)$$

where: $N_{\lambda 0}$ — total level of vibration at foot of equipment, in dB, to be calculated according to the following formula:

$$N_{\lambda 0} = 10\lg\left(\sum 10^{(L_{Pi} - \Delta_i)/10}\right)$$

L_{Pi} — level of vibration of i^{th} octave band center frequency at foot of equipment;

Δ_i — correction, in dB, to be calculated according to Table 3.2.2.4;

Correction Δ_i **Table 3.2.2.4**

Octave band center frequency (Hz)	63	125	250	500	1000	2000	4000	8000
Correction value (dB)	13	11	3	3	0	0	0	0

$S_1^{(2)}$ and $S_2^{(2)}$— maximum and minimum thicknesses of panels surrounding compartment considered, in mm;

$F_3^{(2)}$—area of surrounding panels covered by enveloping surface, in m²;

$F_{\mathrm{II}}^{(2)}$— total area of panels surrounding compartment, in m².

3.2.2.5 Total noise level in adjacent compartment is to be determined according to the following formula:

$$L_T^{(2)} = 10\lg\left(10^{\frac{L_A^{(2)}}{10}} + 10^{\frac{L_S^{(2)}}{10}}\right)$$

where: $L_{\mathrm{A}}^{(2)}$— level of airborne noise transmitting into adjacent compartment;

$L_{\mathrm{S}}^{(2)}$— structure-borne noise level in adjacent compartment.

3.2.2.6 After noise level in adjacent compartment is determined, noise level in separated compartment is to be determined according to the following formula:

$$L_T^{(N)} = N_{\lambda 0} - 4.5n_1 - 1.5n_2 - 10\lg\frac{t^{(n)}}{t^{(1)}} - 10\lg\frac{\alpha^{(n)}}{\alpha^{(1)}}$$

where: n_1 — number of nodes below main deck in the shortest route from separated compartment to source compartment;

n_2 — number of nodes above main deck in the shortest route from separated compartment to source compartment;

$t^{(1)}$— average thickness of partitions of source compartment, in m;

$t^{(n)}$ — average thickness of partitions of separated compartment, in m;

$\alpha^{(1)}$ — sound absorption coefficient of source compartment;

$\alpha^{(n)}$ — sound absorption coefficient of separated compartment.

3.3 Noise prediction based on numerical calculation

3.3.1 General requirements

3.3.1.1 The numerical calculation method applies for noise calculation of compartments on cruise ships in detailed design and production design stage.

3.3.1.2 Finite element, boundary element or meshless method is generally used for low frequency domain noise calculation; statistical energy method is generally used for medium and high frequency domain noise calculation.

3.3.2 Prediction process

3.3.2.1 For cruise ship compartment noise prediction based on numerical calculation, the noise spectrum and distribution of relevant noise sources are to be determined first. Then, according to the structure design of the cruise ship and in conjunction with the use of the internal finishing and vibration and noise reduction materials, the acoustic model of each frequency band is established. Finally, the noise level in any given room is calculated. The numerical calculation process of cruise ship noise prediction is shown in Figure 3.3.2.1.

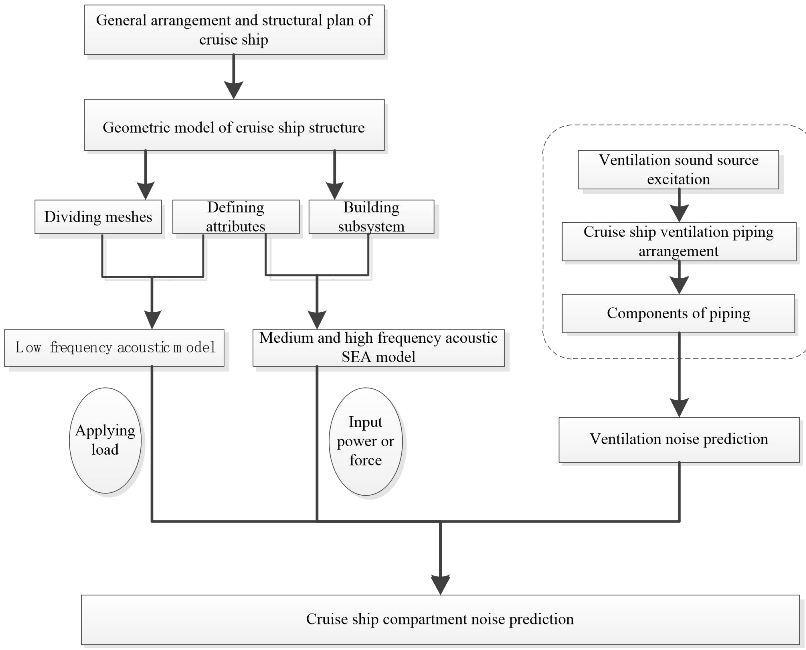


Figure 3.3.2.1 Prediction process of compartment noise of cruise ships based on numerical method

3.3.2.2 The cruise ship noise source is to be determined and the relevant noise source spectrum data are to be obtained (such as the octave band structure noise spectrum of mechanical equipment, and noise power spectrum of air conditioners, etc. provided by the manufacturer. Relevant spectrum data may also be obtained through the on-site measurement of the parent ship).

3.3.2.3 Cruise ship noise prediction model is to be modelled, including:

(1) establishing the finite element model and mass model of cruise ship structure in low frequency domain;

(2) establishing the acoustic finite element (FE) or boundary element numerical model of cruise ship in low frequency domain and setting acoustic parameters;

(3) establishing statistical energy analysis (SEA) model in medium and high frequency domain and setting relevant acoustic parameters. See Figure 3.3.2.3 for statistical energy model.

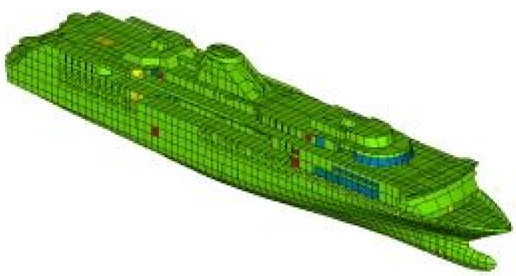


Figure 3.3.2.3 Schematic diagram of cruise ship statistical energy analysis model

3.3.2.4 Full frequency domain cruise ship noise prediction is to be carried out, including:

(1) In the low frequency domain, using the finite element method to analyze the hull vibration noise frequency response;

(2) In medium and high frequency domain, using SEA method to analyze the statistical energy of compartment noise.

3.3.2.5 The sound pressure level and transmission path of noise in the compartment of concern is to be determined, and the noise level is to be assessed according to the rules. At the given octave band and the designated position, the sound pressure level of noise in each compartment is to be calculated according to the low frequency and medium frequency models, and the total sound pressure level of the compartment of concern is to be checked according to the relevant Rules for Cruise Ships to determine whether the compartment noise is acceptable, and to determine the design scheme for vibration and noise reduction.

3.3.3 Noise prediction for low frequency domain

3.3.3.1 Ship vibration noise analysis based on the finite element method is to scatter hull structure and sound field into elements, establish and solve kinetic equation matrix of the whole ship system to get calculation results of node vibration response as well as calculation results of sound pressure. The finite-element kinetic equation matrix is as follows:

$$\left(\begin{bmatrix} K_{ss} & K_{sa} \\ 0 & K_{aa} \end{bmatrix} + j\omega \begin{bmatrix} C_s & 0 \\ 0 & C_a \end{bmatrix} - \omega^2 \begin{bmatrix} M_{ss} & 0 \\ -\rho_0 K_{sa}^T & M_{aa} \end{bmatrix}\right) \cdot \begin{bmatrix} u_i \\ p_i \end{bmatrix} = \begin{bmatrix} F_{st} \\ F_{at} \end{bmatrix}$$

where: K_{ss}—structural stiffness matrix;

K_{aa}—acoustic stiffness matrix;

K_{sa}— coupling stiffness matrix;

C_s—structural damping matrix;

C_a— acoustic damping matrix;

M_{ss}— structural mass matrix;

M_{aa}— acoustic mass matrix;

ρ_0— fluid density;

$\{u_i\}$— structural displacement vector;

$\{p_i\}$— sound pressure vector;

$\{F_{st}\}$— force vector acting on structure;

$\{F_{at}\}$— acoustic excitation vector.

3.3.3.2 Structure modeling is to be carried out in accordance with the following requirements:

(1) In general, the whole ship is to be modeled.

(2) Relevant element types are to be determined according to such factors as boundary conditions of actual structure and characteristics of material media.

(3) Structures such as deck, bulkheads, floors, web beam and girders are to be simulated using plate elements.

(4) Element sizes of hull structure and sound field are to meet relevant division requirements. For linear elements, there are to be at least six elements within minimum wave length; for quadratic elements, there are to be at least three units within minimum wave length. If calculated highest frequency f_{max} is known, the length of all linear elements is not to be greater than $\frac{c}{6f_{max}}$, where c is sound speed velocity.

(5) The interface of hull structure and sound field requires that their sizes are similar, and that

the coupling relation between structural elements and sound field elements be defined.

(6) Side structure is divided into two parts with waterline as benchmark, and elements below water need to be connected with the external flow field and meet the non-reflective boundary condition.

3.3.3.3 Property of material is to comply with the following requirements:

(1) The property of material mainly means characteristics of hull structural materials and characteristics of medium of sound field. Characteristic parameters of relevant material are to be selected as element property.

(2) Characteristics of hull structural materials mainly include modulus of elasticity, Poisson's ratio, density and loss factor.

(3) Element property mainly includes material characteristics, medium characteristics, thickness, additional mass, sound absorption coefficient of hull structure and sound field, which are determined by element type.

(4) Material properties are provided by the manufacturer or a relevant material manual is to be referred to, and structural parameters are mainly obtained from design drawings;

(5) After defining the material property and structural property, the center of gravity of the model is to be checked.

3.3.3.4 Noise source excitation input is to comply with the following requirements:

(1) Noise source parameters are in the form of a low-frequency continuous spectrum.

(2) Measured data of equipment vibration and noise are usually to be used as input data.

3.3.3.5 Compartment noise calculation is to comply with the following requirements:

(1) The frequency interval for calculation is generally not more than 1 Hz.

(2) Sound pressure in the middle of compartment is to be taken as noise calculation result of the compartment (continuous spectrum). If the sound pressure in the compartment changes obviously, sound pressure is to be taken at more location;

(3) Sound pressure level L_{pi} of continuous spectrum is to be converted to sound pressure level $L_p = 10\lg\left(\sum 10^{L_{pi}/10}\right)$ which is corresponding to octave band or 1/3 octave band center frequency.

3.3.3.6 The noise of low frequency compartment is obviously affected by structural resonance, so the secondary noise caused by structural resonance is to be avoided.

3.3.4 Noise prediction of medium and high frequency domain

3.3.4.1 Ship sound vibration analysis based on statistical energy method is to divide the ship into subsystems and obtain compartment noise of the whole ship system by establishing and solving energy equilibrium equation of each subsystem. In general, the following requirements are to be met:

(1) Energy equilibrium equation of ship system is as follows:

$$\omega\{[D][Y][D]^T[N]^{-1} + [Z]\}[E] = \{W\}$$

where:
$[D]$ — connection matrix of $n \times \lambda$ order;
$[Y]$ — coupling loss matrix of $\lambda \times \lambda$ order;
$[Z]$ — internal loss matrix of $n \times n$ order;
$[N]$ — mode matrix of $n \times n$ order;
$[E]$ — subsystem energy matrix of $n \times 1$ order;
ω — considered frequency;

$\{W\}$— external input energy matrix of $n \times 1$ order;

λ — number of connections between subsystems.

(2) Energy E_i of each subsystem is to be obtained by solving energy equilibrium equation to get vibration parameter of each subsystem. Mean square value of vibration speed of each subsystem is as follows:

$$\langle V_i^2 \rangle = \frac{E_i}{M_i}$$

where: E_i — energy of i^{th} subsystem;

M_i — mass of i^{th} subsystem.

(3) Mean square value of sound pressure of subsystems in space sound field is as follows:

$$\langle P_i^2 \rangle = \frac{E_i Z_c^2}{M_i}$$

where: Z_c —sound impedance of space sound field;

M_i — mass of i^{th} subsystem, i.e. air mass of space sound field.

3.3.4.2 Structure modeling is to be carried out in accordance with the following requirements:

(1) In general, structure modeling is to be carried out to the whole ship, and model for calculation is to represent the closed compartment subsystem and its connection with other subsystems. In general, all enclosed spaces on board the ship are to be established as independent acoustic cavity subsystems and connected with hull structure.

(2) Relevant subsystem types are to be determined according to such factors as boundary conditions and material medium characteristics of actual structure. Flat plate subsystem, bent plate subsystem and acoustic cavity subsystem are mainly adopted to establish SEA model of hull structure.

(3) Such structures as decks, bulkheads, floors, web beams and girders are mainly simulated by flat plate subsystems. Hull plate may be simulated by bent plate subsystem or approximately simulated by flat plate subsystem; internal finishing material characteristics may be parametrically simulated.

(4) According to drawings of full-scale structure, flat plate and bent plate subsystems are of stiffener structure type, and stiffening condition of main direction is at least to be considered.

(5) For division of structural subsystems, the size of subsystems is to be as large as possible to ensure that mode number within subsystem analysis bandwidth $\triangle\omega$ is more than 5.

(6) Ship side is divided into 2 subsystems with waterline as benchmark. Subsystems under water need to be connected with external flow field. Tanks are simulated by acoustic cavity (liquid medium).

(7) Windows, doors and other openings are to be replaced by metal plates connected with them. Such equipment as seats and luggage racks need not be simulated.

3.3.4.3 Property of material is to comply with the following requirements:

(1) The property of material mainly means characteristics of hull structural materials, and characteristic parameters of relevant material are to be selected as subsystem property.

(2) Characteristics of hull structural materials mainly include modulus of elasticity, Poisson's ratio, density and loss factor (in the form of octave band).

(3) The property of structural subsystems mainly includes material characteristics and density of hull structure, stiffener condition (area, moment of inertia, spacing, etc.) and internal

finishing (material characteristics, thickness, etc.).

(4) The property of acoustic cavity subsystem mainly includes medium characteristics (sound speed and density) and sound absorption coefficient (in the form of octave band), etc.

(5) Material properties are provided by the manufacturer or a relevant material manual is to be referred to, and structural parameters are mainly obtained from design drawings.

(6) The effect of internal finishing material on compartment noise is to be considered in combination with internal arrangement plan and relevant material property is to be set (thickness, loss factor, sound absorption coefficient, etc.).

3.3.4.4 Noise source excitation input is to comply with the following requirements:

(1) The standard input parameter for excitation source is input power (in the form of octave band). Input power of equipment to hull structure mainly includes equipment vibration which is transferred to foundation and hull structure through foot and airborne noise is radiated directly to air medium.

(2) Measured data of equipment vibration noise are usually to be taken as input data. When the measured result is not available, calculation may be carried out according to the empirical formula in 2.4 of Chapter 2.

(3) If the noise level of equipment near main engine is less than the noise level of main engine by more than 20dB, they may not be treated as excitation source.

(4) Equipment vibration (structure-borne noise) and airborne noise are to be imposed to subsystems corresponding to SEA model.

(5) Input power P of equipment vibration to hull structure is to be calculated according to foundation vibration v and foundation impedance Z obtained by bench test:

$$P = Z \cdot v^2$$

where: v^2 — mean square value of foundation vibration speed, in m^2/s^2;

Z — effective mechanical input impedance of foundation.

(6) If bench foundation structure and equipment installation method (including type of vibration isolator) are similar to those of a real ship, bench foundation vibration data may be adopted directly on foundation structure of SEA model; if not similar, it may be converted by impedance into input power P of equipment vibration to hull structure, and then be loaded to foundation structure of SEA model by impedance conversion;

(7) Input power of equipment noise is directly shown as radiation sound power of the equipment and is normally obtained directly by test.

3.3.5 Compartment noise calculation

3.3.5.1 Octave band or 1/3 octave band is to be selected as analysis type to calculate A-weighted sound pressure level;

3.3.5.2 The calculated results in above 3.3.5.1 and the calculated ventilation noise are superimposed to obtain the noise prediction results for each compartment or space on the cruise ship, which should not exceed the corresponding compartment noise control index.

3.4 Prediction of ventilation noise in air conditioning ventilation system

3.4.1 The prediction of air conditioning ventilation system mainly includes the calculation of sound source noise of air conditioning system, calculation of natural noise attenuation of piping system and calculation of noise regenerated by air flow in piping system, etc. Figure 3.4.1 shows the noise prediction process of air conditioning ventilation system.

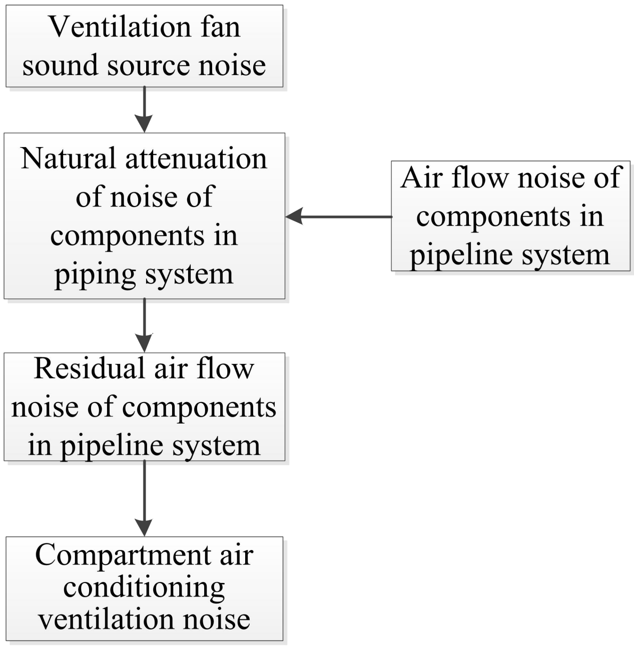


Figure 3.4.1 Noise prediction process of air conditioning ventilation system

3.4.2 The measured data of air conditioning system sound source noise are generally to be provided by the equipment manufacturer. If measured results are not available, calculation may be carried out according to the empirical formula in 2.4.4 of Chapter 2.

3.4.3 Natural attenuation of noise in duct system

3.4.3.1 The natural noise attenuation of air conditioning piping system is mainly from the sound attenuation of plenum chamber, straight ducts, elbows, T-junction and tapered ducts, the sound attenuation at the end of the ventilation outlet and the sound attenuation of noise transmitting from ventilation outlet into the room.

3.4.3.2 The natural noise attenuation of plenum chamber noise may be calculated as follows:

When $50\text{Hz} < f < f_{co}$, $\Delta L = 10.76 A_f S + W_e + L_{oae}$

When $f_{co} < f < 5000Hz$, $\Delta L = 3.505[\frac{S_{outlet}Q}{4\pi r^2} + \frac{S_{outlet}(1-\alpha)}{S\alpha}]^{-0.359} + L_{oae}$

Where: f_{co}— cutoff frequency, to be calculated as follows:

$f_{co} = \frac{c}{2a}$ (for rectangular duct);

$f_{co} = 0.586\frac{c}{d}$ (for round duct).

c ——speed of sound in air, in m/s;
a ——the long side of rectangular duct, in m;
d ——diameter of round duct, in m;
A_f——area correction factor, in db/m^2, see Table 3.4.3.2(1).
S ——total area inside the plenum excluding inlet/outlet, in m^2.
W_e——wall correction, in dB; see Table 3.4.3.2(1).
L_{oae}——correction of offset angle of inlet and outlet. See Table 3.4.3.2(2) to Table 3.4.3.2(3).

S_{oulet}——area of outlet, in m^2.

Q——directivity factor; taken as 2 for opening near center of wall, or 4 for opening near corner of wall.

r——distance between centers of inlet and outlet, in m.

α——average sound absorption coefficient of plenum lining; the unlined sound absorption coefficient is shown in Table 3.4.3.2 (4).

Area correction coefficient A_f and correction coefficient W_e　　Table 3.4.3.2(1)

Area correction coefficient A_f, in dB/m^2			Wall correction coefficient W_e, in dB				
Frequency	Plenum volume		25mm 40kg/m^3 fiber lining	50mm 40kg/m^3 fiber lining	100mm 40kg/m^3 fiber lining	200mm 40kg/m^3 fiber lining	Unlined
	<1.5m^3	>1.5 m^3					
50	1.4	0.3	1	1	0	1	0
63	1	0.3	1	2	3	7	1
80	1.1	0.3	2	2	3	9	2
100	2.3	0.3	2	2	4	12	1
125	2.4	0.4	2	3	6	12	1
160	2	0.4	3	4	11	11	0
200	1	0.3	4	10	16	15	4
250	2.2	0.4	5	9	13	12	1
315	0.7	0.3	6	12	14	14	5
400	0.7	0.2	8	13	13	14	7
500	1.1	0.2	9	13	12	13	8

Correction coefficient of offset angle of inlet and outlet L_{oae} ($f<f_{co}$)　　Table 3.4.3.2(2)

Frequency(Hz)	Offset angle θ					
	0	15	22.5	30	37.5	45
50	0	0	0	0	0	0
63	0	0	0	0	0	0
80	0	0	–1	–3	–4	–6
100	0	1	0	–2	–3	–6
125	0	1	0	–2	–4	–6
160	0	0	–1	–2	–3	–4
200	0	0	–1	–2	–3	–5
250	0	1	2	3	5	7
315	0	4	6	8	10	14
400	0	2	4	6	9	13
500	0	1	3	6	10	15
≥630	N/A	N/A	N/A	N/A	N/A	N/A

Correction coefficient of offset angle of inlet and outlet L_{oae} ($f \geq f_{co}$)　　Table 3.4.3.2(3)

Frequency	Offset angle θ					
	0	15	22.5	30	37.5	45
≤160	N/A	N/A	N/A	N/A	N/A	N/A
200	0	1	4	9	14	20

250	0	2	4	8	13	19
315	0	1	2	3	4	5
400	0	1	2	3	4	6
500	0	0	1	2	4	5
630	0	1	2	3	5	7
800	0	1	2	2	3	3
1000	0	1	2	4	6	9
1250	0	0	2	4	6	9
1600	0	0	1	1	2	3
2000	0	1	2	4	7	10
2500	0	1	2	3	5	8
3150	0	0	2	4	6	9
4000	0	0	2	5	8	12
5000	0	0	3	6	10	15

Where the central axes of inlet and outlet is not in-line, as shown in Figure 3.4.3.2, the offset angle is to be calculated according to the following formula:

$$\cos\theta = \frac{l}{r} = \frac{l}{\sqrt{l^2 + r_v^2 + r_h^2}}$$

where: θ——offset angle of inlet and outlet, in º;

l——length of plenum, in m;

r_v——vertical offset between axes of inlet and outlet, in m;

r_h——horizontal offset between axes of inlet and outlet, in m.

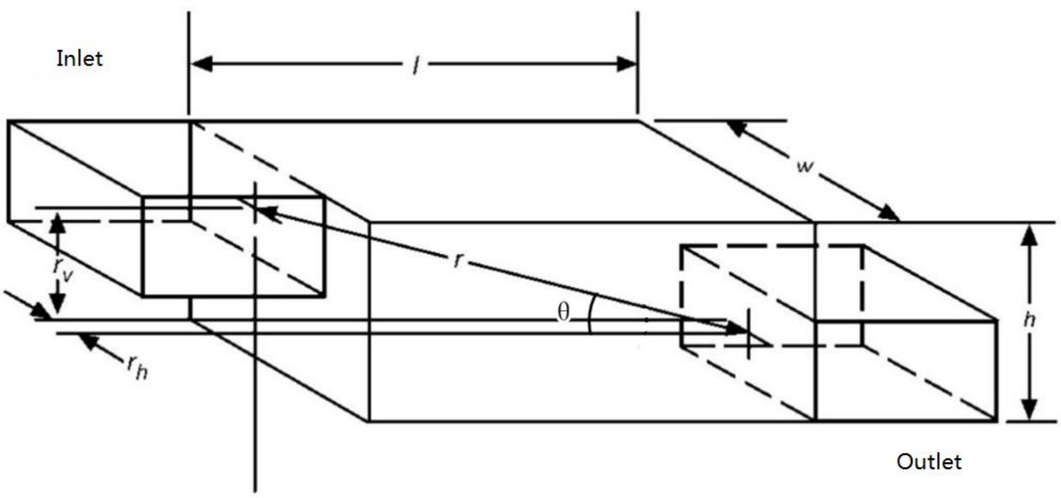


Figure 3.4.3.2 Schematic of plenum chamber

Unlined sound absorption coefficient in plenum chamber **Table 3.4.3.2(4)**

Unlined sound absorption coefficient in plenum chamber	63	125	250	500	1000	2000	4000

Metal sheet	0.04	0.04	0.04	0.05	0.05	0.05	0.07

3.4.3.3 The natural attenuation of rectangular duct and round duct in straight ducts can be obtained from Table 3.4.3.3.

Sound attenuation in metal straight ducts（dB/m） **Table 3.4.3.3**

Duct dimension, in *De*/m		Octave band center frequency /Hz				
		63	125	250	500	≥1000
Rectangular duct	0.075~0.2	0.6	0.6	0.45	0.3	0.3
	0.2~0.4	0.6	0.6	0.45	0.3	0.2
	0.4~0.8	0.6	0.6	0.3	0.15	0.15
	0.8~1.6	0.45	0.3	0.15	0.1	0.06
Round duct	0.075~0.2	0.1	0.1	0.15	0.15	0.3
	0.2~0.4	0.06	0.1	0.1	0.15	0.2
	0.4~0.8	0.03	0.06	0.06	0.1	0.15
	0.8~1.6	0.03	0.03	0.03	0.06	0.06
Note: Piping size D_e, for round piping is the diameter, for rectangular piping D_e=2*ab*/(*a*+*b*), where *a* and *b* are side lengths of rectangular piping respectively.						

3.4.3.4 The natural attenuation of elbows plays a certain role in natural attenuation of piping system, especially in the lined elbow and the medium and high frequency range. Table 3.4.3.4(1) shows the natural attenuation of square elbows without turning vanes, Table 3.4.3.4(2) shows the natural attenuation of square elbows with turning vanes, and Table 3.4.3.4(3) shows the natural acoustic attenuation of round elbows, where *f* is the octave band center frequency (kHz) and *w* is the width of square elbows (mm), as shown in Figure 3.4.3.4.

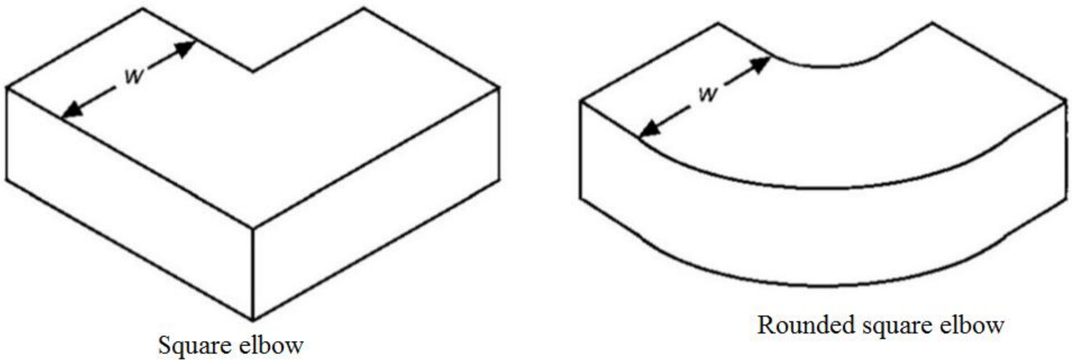


Figure 3.4.3.4 Square elbows

Natural attenuation of square elbows without turning vanes **Table 3.4.3.4(1)**

Natural attenuation of square elbows without turning vanes (dB)		
Dimension parameter	Unlined elbows	Lined elbows
fw<48	0	0
48<*fw*<96	1	1
96<*fw*<190	5	6
190<*fw*<380	8	11

380<*fw*<760	4	10
fw>760	3	10

Natural attenuation of square elbows with turning vanes Table 3.4.3.4(2)

Natural attenuation of square elbows with turning vanes (dB)		
Dimension parameter	Unlined elbows	Lined elbows
fw<48	0	0
48<*fw*<96	1	1
96<*fw*<190	4	4
190<*fw*<380	6	7
fw>380	4	7

Natural attenuation of round elbows Table 3.4.3.4(3)

Natural attenuation of round elbows (dB)	
Dimension parameter	Round elbows
fw<48	0
48<*fw*<96	1
96<*fw*<190	2
fw>190	3

3.4.3.5 For natural attenuation of T-junction, the noise energy may be distributed according to the proportion of branch cross-sectional area (or air volume distribution ratio), then the natural noise attenuation from the main duct to any branch duct may be calculated as follows:

$$\Delta L = 10\lg\left(S_1/S\right)$$

Where: S_1— cross-sectional area of the branch duct, in m^2;

S— cross-sectional area of all branch ducts at the bifurcation, in m^2.

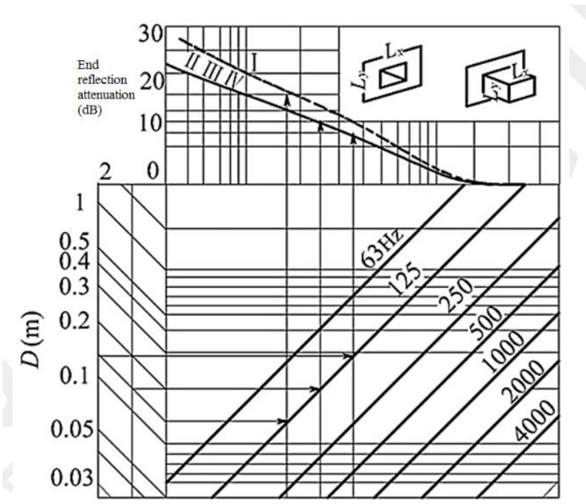
3.4.3.6 Natural attenuation of tapered ducts and natural acoustic attenuation caused by the sudden change of piping cross section may be calculated as follows:

$$\Delta L = 10\lg\frac{\left(\frac{S_2}{S_1}+1\right)^2}{4}\times\frac{S_2}{S_1}$$

Where: S_1——cross-sectional area of piping before diameter reduction, in m^2;

S_2——cross-sectional area of piping after diameter reduction, in m^2.

3.4.3.7 The size of the reflection loss at the end of outlet is related to the outlet area, outlet location and noise frequency, which may be obtained from Figure 3.4.3.7.



Category of opening position: Ⅰ Center of room, projecting part (free space) Ⅲ Corner of wall

Ⅱ Center of wall (ceiling) Ⅳ angle of intersection of three sides

Figure 3.4.3.7 Nomogram showing loss at the end of outlet

3.4.3.8 The value of the indoor sound pressure level L_p at the distance r from the outlet may be calculated by the following formula:

$$L_p = L_w + 10\lg\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right)$$

Where: L_w——sound power level from the outlet into the room, in dB;

Q——outlet directivity factor; when the sound source is located in the geometric center of the room, Q =1; when the sound source is located in the center of the indoor ground or in a wall, Q =2; when the sound source is located at the midpoint of an indoor sideline, Q =4; When the sound source is located in a corner of the room, Q =8;

R——room constant, in m^2, $R = \frac{S\overline{\alpha}}{1-\overline{\alpha}}$;

$\overline{\alpha}$——average sound absorption coefficient in a room.

3.4.4 Regenerated noise of air flow in duct system

3.4.4.1 The duct components with air flow regenerated noise in the duct system mainly include: straight ducts, elbows, valves, tapered ducts, T-junction and outlets, etc. Airflow noise in piping components is generally obtained by laboratory testing; when measured data is not available, it may be calculated by using the following method.

3.4.4.2 Airflow noise L_w of straight ducts is related to flow velocity and duct cross-sectional area, to be calculated according to the following formula:

$$L_w = L_{wc} + 50lgv + 10\lg S + \Delta b$$

where: L_{wc}——specific sound power level of straight ducts, in dB, generally taken as 10;

v ——flow velocity, in m/s;

S ——Cross-sectional area of ducts, in m^2.

Δb ——See Table 3.4.4.2 for the corrected values of sound power levels of air flow noise at various octaves.

Octave band corrected value of sound power level of air flow regenerated noise in straight duct Δb Table 3.4.4.2

Straight duct	Octave band center frequency(Hz)							
	63	125	250	500	1000	2000	4000	8000
Corrected value (dB)	-5	-6	-7	-8	-9	-10	-13	-20

3.4.4.3 The airflow noise L_w of the tapered duct is mainly determined by the flow velocity and cone angle, to be calculated as follows:

$$L_w = A + B\lg v - 3K$$

Where: v —— flow velocity, in m/s;

A, B——coefficient, obtained from Table 3.4.4.3(1);

K—— corrected values related to the tapered angle, obtained from Table 3.4.4.3(2).

Coefficients A, B Table 3.4.4.3(1)

Coefficient	Octave band center frequency (Hz)					
	63	125	250	500	1000	2000
A(dB)	47.2	48.6	52.8	52.8	54.2	57.2
B(dB)	27.3	22.9	15.2	13.0	9.8	5.3

Correction K Table 3.4.4.3(2)

Correction	Tapered angle of tapered duct (°)								
	0~20	21~40	41~50	51~58	59~63	64~68	69~73	74~80	81~90
K(dB)	9	8	7	6	5	4	3	2	1

3.4.4.4 Elbow air flow noise is to be calculated according to the following formula:

$$L_w = L_{wc} + 10\lg f_{lower} + 30lgd_e + 50\lg v$$

where: L_{wc}—— the specific sound power level of the elbow (dB) is a function of N_{str} , $N_{str} = fd_e / v$. For square and rectangular right-angle elbows, it can be found from Figure 3.4.4.4(1); for round elbows, it can be found from the curve of $v / v_a = 1$ in Figure 3.4.4.4(2).

$10\lg f_{\text{lower}}$——to be found in Table 3.4.4.4;

d_e——elbow diameter, for rectangular elbow, $d_e = \frac{2ab}{a+b}$.

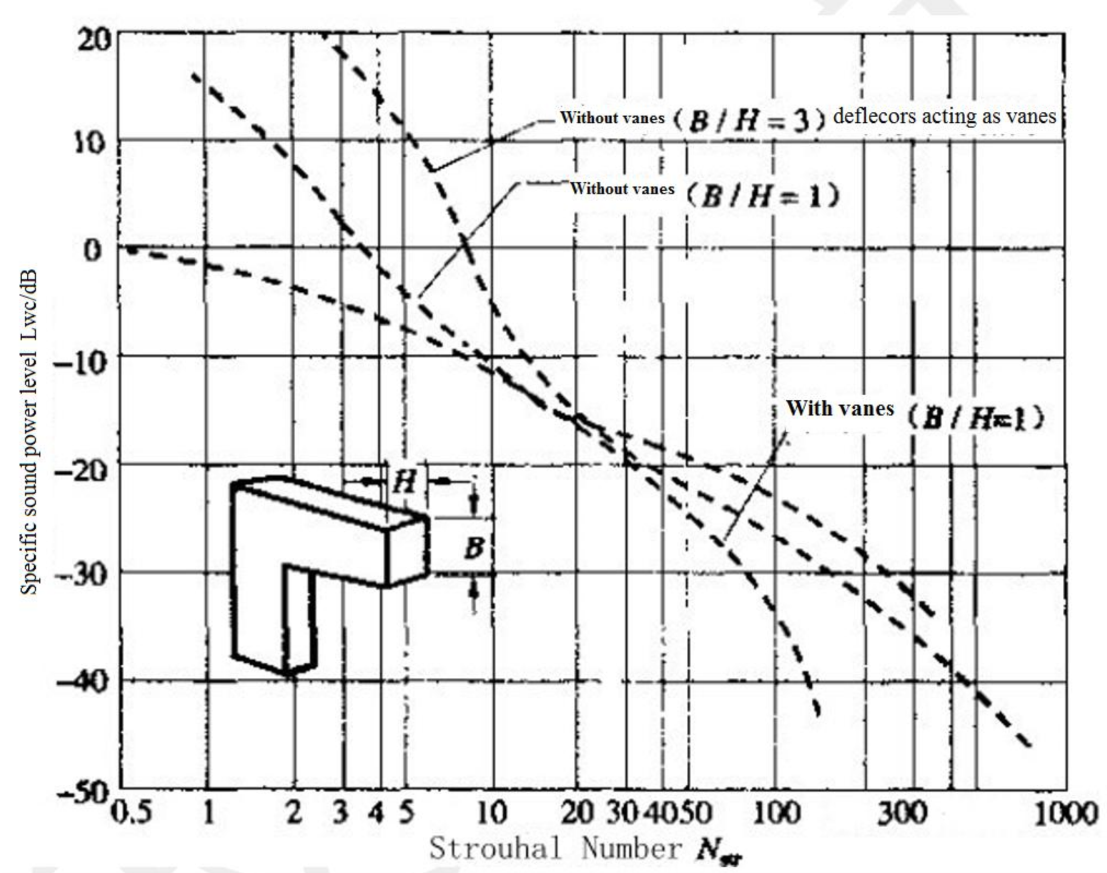


Figure 3.4.4.4(1) Functional relationship between the specific sound power levels of square and rectangular elbows and N_{str} (Strouhal Number)

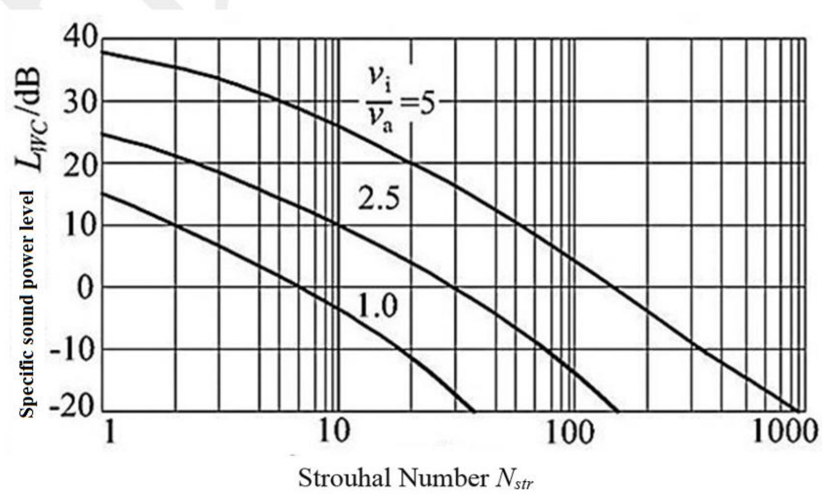


Figure 3.4.4.4(2) Functional relationship between the specific sound power level of the round elbow (T-junction), N_{str} **and the inlet/outlet flow rates of the T-junction**

Corrected values of $10\lg_{f\,lower}$ **Table 3.4.4.4**

Elbows	Octave band center frequency (Hz)							
	63	125	250	500	1000	2000	4000	8000

Correction of $10\lg f_{\text{lower}}$	16	19	22	25	28	32	35	38

3.4.4.5 The sound power level of valve air flow noise is to be calculated according to the following formula:

$$L_w = L_\theta + 10\lg S + 55\lg v + \Delta b$$

where: L_θ ——Constant determined by valve vane angle;

when θ= 0°, $L_\theta = 30$ dB;

when θ=45°, $L_\theta = 42$dB;

when θ= 65°, $L_\theta = 51$dB;

S ——duct cross-sectional area, in m^2;

v ——velocity in duct, in m/s.

Δb ——correction of frequency band sound power level, as shown in Table 3.4.4.5:

Correction of frequency band sound power level of valve air flow regenerated noise Δb

Table 3.4.4.5

Valve vane angle θ	Octave band center frequency (Hz)							
	63	125	250	500	1000	2000	4000	8000
0°	-4	-5	-5	-9	(-14)	(-19)	(-24)	(-29)
45°	-7	-5	-6	-9	-13	-12	-7	-13
65°	-10	-7	-4	-5	-9	0	-3	-10

3.4.4.6 Calculation of air flow noise in T-junction is similar to that of elbows, as follows:

$$L_w = L_{wc} + 10\lg f_{\text{lower}} + 30lgd_e + 50\lg v + \Delta b$$

Where: L_{WC}——the specific sound power level of T-junction may be obtained according to v / v_a (the flow rate of the T-junction inlet and outlet respectively) and N_{str} values from Figure 3.4.4.4(2). Figure 3.4.4.4(2) only applies to the condition of $r/d_e = 0.15$ (r is the radius of curvature of the elbow, in m). For different r / d_e values, the corrected values may be obtained from Figure 3.4.4.6.

Δb ——correction of T-junction air flow regenerated noise due to eddy current may be obtained in Table 3.4.4.6.

Correction of T-junction air flow regenerated noise due to eddy current Table 3.4.4.6

T-junction eddy current	v / v_a (Ratio of T-junction inlet to outlet flow rates)								
	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0
Correction	0	0.7	1.2	1.8	2.4	2.9	3.3	3.6	4.0

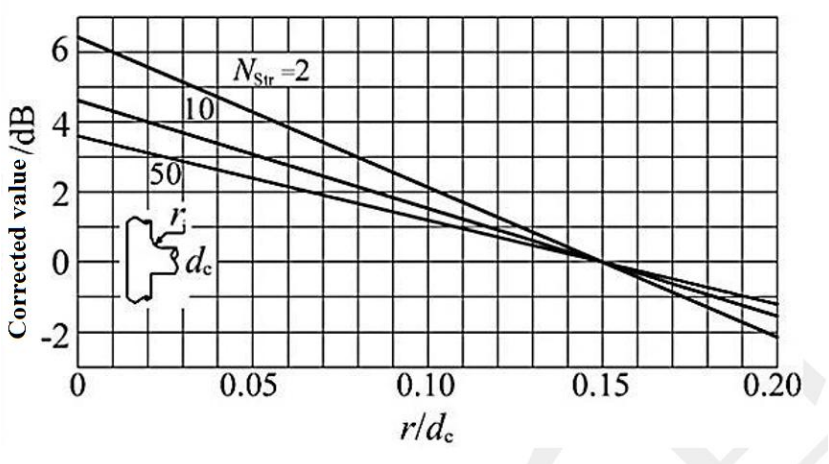


Table 3.4.4.6 Correction of the specific sound power level of T-junction (round elbow) according to different r/de

3.4.4.7 The sound power level of air flow noise in full frequency band of various air supply/return outlets is to be calculated by the following formula, and the correction of each frequency band is shown in Table 3.4.4.7(2) and Table 3.4.4.7(3).

$$L_w = 10\lg S + a\lg V + b$$

where: L_W——full frequency band sound power levels of air flow noise, in dB;

S——cross-sectional area of the air supply outlet or the throat area, in m^2;

V——flow velocity at the cross-section of the air supply outlet or throat, in m/s;

a,b ——Empirical constant determined by outlet type, see Table 3.4.4.7(1).

Values of a, b　　　　Table 3.4.4.7(1)

Outlet type	*a*	*b*
Nozzle	83	-38
Louver air supply outlet	48	15
Grille air supply outlet	50	30
Slot air supply outlet	40	54
Diffuser	50	35
Disk air supply outlet	50	42
Wheeled air supply outlet	50	32
Grille air return outlet	50	38
Disk air return outlet	67	21
Diffused air return outlet	63	31

Correction of air flow noise frequency of each type of air supply outlet　　Table 3.4.4.7(2)

Type of air outlet	Octave band center frequency (Hz)							Remarks
	63	125	250	500	1000	2000	4000	
Nozzle	-2	-7	-7	-11	-16	-18	-19	Throat speed ≤15m/s

Adjustable louver air outlet	-3	-7	-9	-14	-14	-17	-22	Throat speed≤15m/s
Grille air outlet	-6	-5	-6	-9	-11	-18	-26	Cross-section speed≤5m/s
Seam air outlet	-8	-7	-6	-6	-9	-14	-24	Cross-section speed≤5m/s
Round diffuser	-2	-5	-8	-12	-16	-23	-29	Throat speed≤7m/s
Directional diffuser	-3	-6	-7	-8	-8	-11	-18	Throat speed≤7m/s
Disk type air outlet	-6	-5	-6	-9	-11	-16	-24	Throat speed≤7m/s
Wheeled air outlet	-	-5	-4	-7	-9	-14	-24	Throat speed≤6m/s

Correction of sound power level of air flow noise frequency band of each type of air return outlet

Table 3.4.4.7(3)

Type of air return outlet	Octave band center frequency (Hz)							Remarks
	63	125	250	500	1000	2000	4000	
Grille air return outlet	-8	-12	-10	-6	-6	-14	-23	Cross-section speed≤3m/s, valve fully open
Disk type air return outlet	-9	-7	-10	-10	-12	-16	-29	Throat speed≤5m/s
Diffused air return outlet	-3	-9	-11	-14	-11	-10	-18	Throat speed≤5m/s

3.4.4.8 The sound power level of air flow regenerated noise of orifice plate air supply outlet may be calculated as follows (all frequencies are the same):

$$L_W = 15 + 60\lg v + 30\lg\xi + 10\lg S$$

where: v ——flow velocity before orifice plate, in m/s;

ξ ——Orifice plate resistance coefficient, see table 3.4.4.8;

S ——Total surface area of orifice plate, in m^2.

Resistance coefficient of orifice plate (or grille) ξ **Table 3.4.4.8**

Opening area (%) / Flow speed(m/s)	20	30	40	50	60	70	80
0.5	30.0	12.0	6.0	3.6	2.3	1.8	1.4
1.0	33.0	13.0	6.8	4.1	2.7	2.1	1.6
1.5	36.0	14.5	7.4	4.6	3.0	2.3	1.8
2.0	39.0	15.5	7.8	4.9	3.2	2.5	1.9
2.5	40.0	16.5	8.3	5.2	3.4	2.6	2.0
3.0	41.0	17.5	8.6	5.5	3.7	2.8	2.1

3.4.5 In addition to using the method provided in this section to forecast the ventilation noise of the air conditioning ventilation system, CFD numerical method may also be used for calculation.

Chapter 4 VENTILATION NOISE CONTROL OF AIR CONDITIONING VENTILATION SYSTEM

4.1 Duct arrangement of air conditioning ventilation system

4.1.1 The noise level of the air conditioning system transmitted to the use room and the surrounding environment is to comply with the requirements of the noise level of the cruise ships.

4.1.2 The installation of refrigeration units, water pumps, air conditioning units and ventilation ducts in the air conditioning system is to be equipped with effective vibration isolation, so that the noise generated by them transmitted to the air conditioning space and the surrounding space can comply with the requirements of the noise level of the cruise ships.

4.1.3 When setting the air conditioning system ducts, the air duct after sound muffling treatment is not to pass through the room with high noise. The air duct with high noise is not to pass through the room with low noise requirements. When it is necessary to pass through, sound insulation treatment is to be taken.

4.1.4 For air conditioning systems with sound muffling requirements, it is recommended to select the air flow velocity in the air duct according to Table 4.1.4.

Recommended air flow velocity in air duct (m/s) Table 4.1.4

Permissible indoor noise level dB(A)	Wind speed of main duct	Wind speed of branch duct
25～35	3～4	≤2
35～50	4～7	2～3
50～65	6～9	3～5
65～85	8～12	5～8
Note: The wind speed of the air duct between the fan and the muffling device is recommended to be 8 ~10m/s.		

4.1.5 Fans, air conditioning devices and refrigeration rooms are not to be close to rooms with high acoustic environment requirements. Where it is required to be near these rooms, sound and vibration isolation measures are to be taken.

4.1.6 Where the noise of the equipment exposed outdoors does not meet the requirements of the environmental noise standard, noise reduction measures are to be taken. For example, the muffler is to be provided in the inlet and exhaust vents, or the noise reduction barrier is to be provided around the vents.

4.1.7 When the natural attenuation of the noise from the air conditioning system cannot reach the applicable noise standards, the muffler is to be provided or other sound muffling measures are to be taken.

4.1.8 The influence of air flow in the duct on the muffler capability is to be taken into consideration for the arrangement of mufflers. The air duct between the mufflers and the partition wall of the machinery room is to be equipped with sound insulation capability.

4.1.9 Where the duct passes through the enclosure of the machinery room, the gap between the duct and the enclosure is to be filled with elastic materials with fire and sound insulation capabilities.

4.2 Determination of compartment noise limit

4.2.1 Before the noise control of the air conditioning ventilation system, the specified noise limit of the target compartment is to be defined first, based on which the noise reduction design and evaluation of the air conditioning ventilation system is to be carried out.

4.3 Selection of air conditioning system equipment

4.3.1 The relevant equipment of the air conditioning system is to be reasonably selected in the noise reduction design of the air conditioning ventilation system. The fan with low noise is to be selected as far as possible and is to run at the highest power condition point, and its air volume and air pressure are to match the design requirements. Adopting distributed small system air conditioning design, improving the connection method between fans and the ducts, and taking necessary vibration isolation measures are very beneficial to reduce system noise. Figure 4.3.1(1) shows the connection method of fans and the ducts, Figure 4.3.2(2) shows the comparison of connection methods of fans and ducts, and Table 4.3.1 shows the influence of noise of air conditioning equipment such as fan and common noise reduction measures.

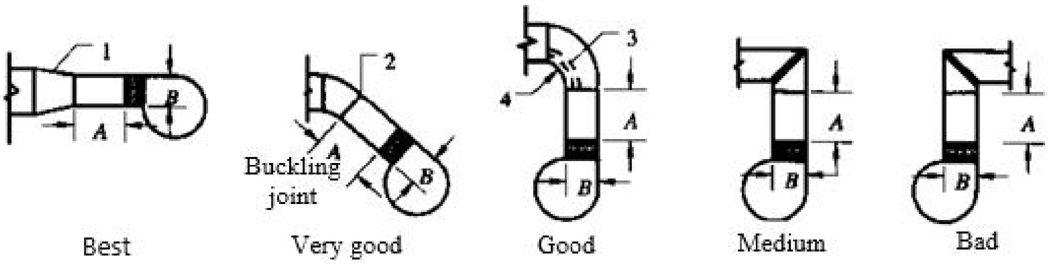


Figure 4.3.1(1) Connection methods of fans and ducts

Note: 1-Tapered duct 2-Outlet duct section A≥1.5B 3-Wind guide vane 4-Elbow radius ≥150mm

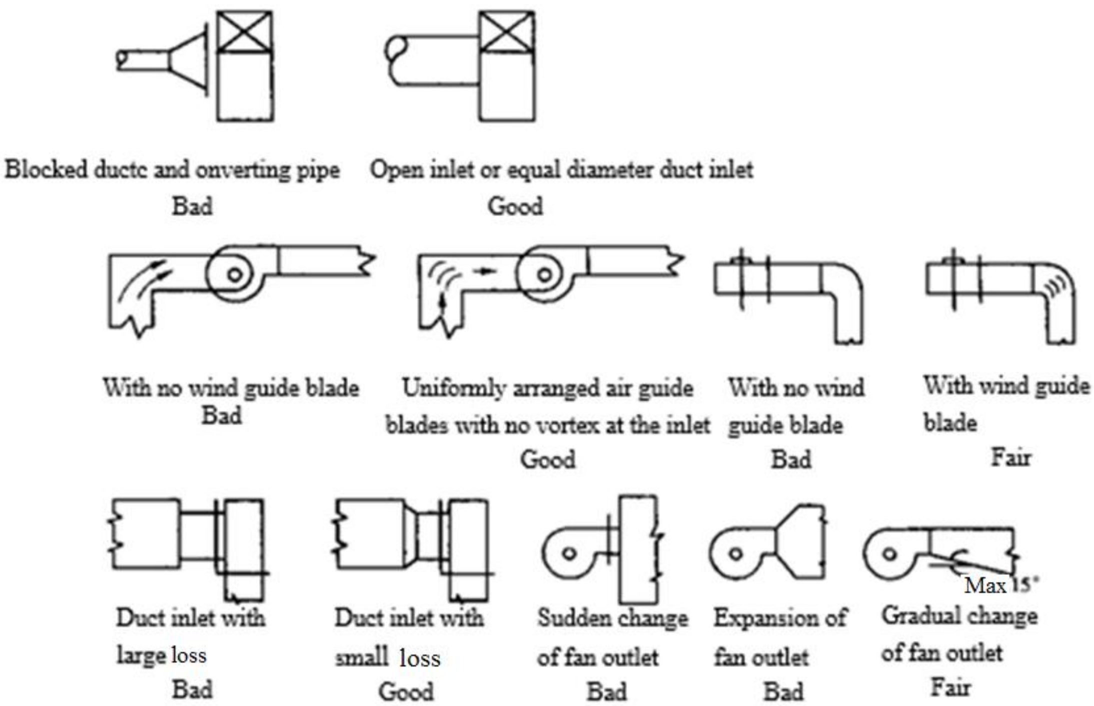


Figure 4.3.1(2) Comparison of connection methods of fans and ducts

Influence of noise of air conditioning equipment such as fan and common noise reduction measures **Table 4.3.1**

Air conditioning equipment	Noise transmission and influence	Common noise reduction measure

Fan	Transmitted into the indoor or affect the outdoor and adjacent cabins through the air duct	Heat insulation and sound absorption for the machine room, vibration isolation for unit, soft joint, elastic hook and muffler for duct
Air conditioner	Transmitted into the indoor or affect the outdoor and adjacent cabins through the air duct	Sound insulation and sound absorption for machine enclosure, and the rest is the same as the fan
Outdoor heat pump and air cooled condenser	Noise affects the outdoor environment	Muffler for exhaust shaft fan, sound insulation, sound absorption or sound barrier for machine enclosure, and vibration reduction for foundation
Refrigerator and water chilling unit	Noise and vibration affect the whole cabin	Sound insulation and sound absorption for the machine room, vibration isolation for unit, vibration reduction and elastic hook for duct
Water pump	Noise and vibration affect the whole cabin	Sound insulation and sound absorption for the machine room, vibration isolation for unit, vibration reduction and elastic hook for duct
Fan coil unit	The noise is transmitted into the room directly	Sound muffling treatment for air outlet and sound muffling static pressure box treatment for air inlet

4.4 Selection and arrangement of mufflers

4.4.1 Main principles of muffler selection

4.4.1.1 The required sound muffling capacity is determined in accordance with the noise and spectrum characteristics of the fan and the allowable noise standard of the target compartment, that is, the sound muffling performance of the muffler is to adapt to the required sound muffling capacity.

4.4.1.2 The pressure loss of the selected muffler is to adapt to the allowable pressure loss of the piping system.

4.4.1.3 The regenerated air noise of muffler is to adapt to the sound source and muffler performance so that the muffler performance can be fully developed.

4.4.1.4 The external dimension and length of the muffler are to be suitable for the actual installation position.

4.4.1.5 The selected muffler is to be of strong structure, corrosion resistance, fire resistance, dust prevention, long service life and small volume, and its sound-absorbing material is to meet marine environmental conditions (damp-proof, flame-retardant, oil resistant and free from dust or harmful gas release), being easy for installation and maintenance.

4.4.2 Main principles for determining the installation position of mufflers

4.4.2.1 The muffler is to be provided as far as possible in the duct section where the airflow is stable.

4.4.2.2 The muffler is to be provided in the air duct section behind the fan room as far as possible to avoid the noise in the machine room into the duct behind the muffler.

4.4.2.3 Where the main flow rate is high, the muffler is to be installed in the branch section.
4.4.2.4 For systems requiring high sound muffling requirements and a lot of mufflers, mufflers can be provided sectionally rather than arranged concentratedly.
4.4.2.5 For the air conditioning system with limited installation length and space, the effect of muffler elbow and straight duct muffler can be used.
4.4.2.6 Where the installation position of the muffler is limited, the sound muffling plenum can be designed and installed by using the building space and the air outlet position of the air conditioning box.
4.4.2.7 When the air outlet and inlet of adjacent sound insulation rooms come from the same air duct, anti-crosstalk muffling devices must be provided.
4.4.2.8 The return air system is also to be equipped with enough mufflers, and attention is to be paid to the patency of the return air to avoid excessive airflow regenerated noise at the air outlet.

4.5 Design procedure for mufflers

4.5.1 Steps for design of dissipative mufflers
4.5.1.1 Determining structural form of muffler. Based on the flow passing through the muffler and intended average flow velocity, it is to calculate required flow-through cross section for selecting the structural form of the muffler (straight-through or sheet type).
4.5.1.2 Adopting suitable sound absorbing material. The sound absorbing material is to meet environmental conditions of cruise ships; its engineering application is to be taken into account for its sound absorption coefficient under consideration so as to determine its thickness and allowable weight.
4.5.1.3 Determining muffler length. Muffler length (including sound muffling bend) is to be determined according to the sound pressure level of the noise source and noise control requirements, taking into account the allowable space size on board.
4.5.1.4 The surface protection layer of sound absorbing material is to be selected according to airflow velocity.
4.5.1.5 Resistance loss is to be estimated according to muffler length and structure.
4.5.1.6 Sound muffling capacity is to be corrected according to “high frequency failure” and airflow-regenerated noise.
4.5.2 Steps for design of reactive mufflers
4.5.2.1 Based on required frequencies for sound absorption, the expansion room and its insertion duct length are to be designed rationally to determine frequencies for maximum sound muffling.
4.5.2.2 Expansion ratio is to be determined and sectional size of each part of the expansion room designed according to required sound muffling capacity.
4.5.2.3 To check that upper and lower limiting frequencies of the muffler in the expansion room is within the required scope of sound absorption frequencies.
4.5.2.4 Resistance loss is to be calculated.