Study on aerodynamic noise of axial flow fan in ship signal cabinet

Zenghui Zhu¹ and Shu Li
No.724 Research Institute of CSIC, Nanjing, China
¹ E-mail: zhuzenghui1120@163.com

Abstract. In order to study the aerodynamic noise characteristics of axial flow fan in the ship signal cabinet, the aerodynamic noise theory analysis and experimental research were carried out. By comparing and analyzing the vibration and noise spectrum of cabinet, it was determined that the noise of cabinet mainly came from the aerodynamic noise of axial flow fan in the cooling unit of rotating cage. The noise control of cabinet was studied by using the optimal design of the fan installation structure and the application of sound-absorbing cotton. The results show that the noise of fan was discrete noise. The noise increased 2.91 dB(A) when the net was installed in the fan inlet, meanwhile, the energy ratio of high-frequency noise (6000 to 14000 Hz) increased significantly. The sound was sharp. The noise of fan became smaller with the decrease of speed. The peak frequencies of fan noise spectrum were the multiple frequency of fundamental frequency and shaft frequency at different speeds. The sound pressure level was the largest at the fundamental frequency. That is, the noise of fan came from its blades. The noise reduction was 4.24 dB(A) by applying the sound-absorbing cotton on the inner wall of cabinet. The effects of noise reduction were obvious in the 200 to 2000 Hz range. The research results can provide theoretical and experimental support for the optimal design of ship signal cabinet.

1. Introduction

Due to the requirement of integration, the number of electronic modules in the ship signal cabinet increases. The electronic modules are sensitive to temperature, and high temperature inside the cabinet will reduce the reliability of the electronic modules. Therefore, the design of cabinet should consider cooling design. At present, the commonly used cooling mode in the cabinet is air cooling, that is, the cooling unit is installed in the rotating cage, and the axial flow fan inside the cooling unit rotates through the fin in the electronic modules. The number, distribution and speed of fan in the cooling unit are determined by fluid finite element software. Because of the influence of fan blade shape, blade numbers, speed, air duct and resonance, the cabinet is noisy [1].

Hamakawa et al. [2] analyzed the relationship between aerodynamic noise of axial flow fan under maximum pressure and the fluctuation of leakage eddy current velocity of rotating blade top. Zhang et al. [3] obtained the unsteady parameters of the internal flow field of the small axial flow fan by using Large Eddy Simulation (LES), and the sound pressure level (SPL) of different regions of fan were calculated based on the Flowcs Williams-Hawkings (FW-H) model. Based on LES, Li et al. [4] calculated the pulsating pressure on the centrifugal fan blade, and predicted the noise of fan pipeline by using the acoustic finite element. The researchers discussed the influence of fan design parameters on its aerodynamic noise based on acoustic simulation, Limited research had been conducted on noise test of axial flow fan in the ship signal cabinet. As the noise requirement of ship equipment is high, the
cabinet in the cabin should meet the noise limit requirement of GJB 763.3-89. Therefore, noise control and analysis of cabinet should be carried out.

In this paper, aerodynamic noise of axial flow fan is analyzed theoretically. Vibration and noise analysis platform of ship signal cabinet is built by using microphones, accelerometers and data collector. Based on noise and vibration acceleration spectrum, the noise source characteristics of signal cabinet are analyzed. The noise control of cabinet is studied by using the optimal design of the fan installation structure and the application of sound-absorbing cotton. The influence of net and speed on the noise of fan are discussed, and the influence of sound-absorbing material on the noise spectrum characteristics of cabinet is analyzed. It provides theoretical and experimental support for the optimization design of ship signal cabinet.

2. Theoretical analysis for aerodynamic noise of axial flow fan

The unsteady flow of fluid in the axial flow fan is the main cause of aerodynamic noise. According to the spectral characteristics of noise, the noise includes discrete noise and broadband noise.

Discrete noise is aerodynamic noise with discrete peaks in the noise spectrum, also known as rotational noise, which has periodicity [5, 6]. In 1970, based on the motion point source equation and the Fourier series expansion, Lowson calculated the noise generated by the fan rotating blade. The m harmonic of sound pressure \( p_m \) generated by the fan blade is defined as [7]

\[
p_m = \frac{imZ^2\omega}{2\pi c_0r_1} \sum_{n=-\infty}^{\infty} (-i)^{n-m} \times \left( \frac{xT_u}{n} - \frac{mZ-nA}{mZM} \right) J_{mZ-nA} \left( \frac{mZMy}{n} \right) \]

where \( m \) and \( n \) are the integer, \( i \) the imaginary unit, \( \omega \) the angular velocity of fan blade, \( c_0 \) the sound velocity, \( r_1 \) the distance between observation point and origin of coordinate system, \( x \) and \( y \) the coordinate, \( M \) the Mach number, \( Z \) the number of rotating blades, \( A \) the number of static blades, \( T_u \) the \( n \) order pulsating component of thrust, \( D_n \) the \( n \) order pulsating component of drag, \( J_{mZ-nA} \) the \( mZ-nA \) order Bessel function.

According to Equation (1), the sound pressure becomes larger with the increase of speed. Discrete noise is the largest at the fundamental frequency of fan, and its noise attenuates with the increase of harmonics.

The fundamental frequency of fan \( f_b \) is defined as

\[
f_b = \frac{NZ}{60} \tag{2}
\]

where \( N \) is the speed of fan.

Equation (2) shows that the fundamental frequency of fan is linear with its speed and the number of blades, and the fundamental frequency of fan increases with the increase of speed and the number of blades [8].

Broadband noise, also known as vortex noise or turbulent noise, has a certain randomness, which is shown as continuous broadband spectrum. After the internal fluid of fan is separated, the blade pulsating pressure is formed, which produces broadband noise. Broadband noise is affected by vortex sound and turbulent flow [9, 10]. The SPL of broadband noise of fan \( L_p \) is defined as [11]

\[
L_p = 20\log_2 \frac{Z\rho_0\omega}{c_0} \sum_{u=2}^L \frac{D_u \cos \beta_u}{r_u} + 80 \tag{3}
\]

where \( L \) is the number of blade units, \( \rho_0 \) the air density, \( D_u \) the thickness of blade unit, \( u \), \( r_u \) the distance between observation point and sound source point, \( \beta_u \) the angle of attack of blade unit.

According to the Equations (1) and (3), discrete noise and broadband noise of fan are related to the speed, the number of blades and the distance of noise measuring point. Discrete noise and broadband noise increase with the increase of speed and the number of blades. The farther the noise measuring point is from the fan, the smaller the SPL at the measuring point is. Measuring point is at 1 meter away from the test object specified by GJB 4058-2000. If the fan is in a large flow state, the noise spectrum of fan is mainly discrete noise. If the fan is in a small flow state, the fan noise spectrum is mainly broadband noise.
3. Test equipment and methods
The test equipment included signal cabinet, microphones, accelerometers and data collector. The 40-channels data collector can select input modes such as voltage, ICP and charge. It can collect sound pressure and acceleration signals. The maximum sampling rate is 204.8 kHz with 24-bit module conversion. Figure 1(a) shows the structure of cabinet. Figure 1(b) shows the structure of cooling unit in the cabinet. The cabinet adopted the form of cast aluminum closed structure, and the electronic modules were air-cooled through the cooling unit of rotating cage in the cabinet. Four fan 1 and one fan 2 of the cooling unit were axial flow fans with a working voltage of 48 V. Fan outlet was equipped with net whose material was steel. Table 1 shows the parameters of fan. $f_i$ is the shaft frequency of fan.

Table 1. Parameters of fan.

| Fan No | Z  | Size/mm  | $N$/r·mm$^{-1}$ | $f_i$/Hz | $f_b$/Hz |
|--------|----|----------|----------------|----------|----------|
| 1      | 9  | 120×120×26 | 5080          | 84.7     | 762      |
| 2      | 11 | 90×90×26  | 4220          | 70.3     | 773.7    |

The aerodynamic noise test of cabinet was carried out by noise source location and noise control. Figure 2(a) shows the layout of measuring points of cabinet. 3 microphones were arranged at 1 meter away from the cabinet respectively. To analyze the influence of front door of cabinet on its noise, the
noise was tested under two operating conditions: closing the front door (condition 1) and opening the front door (condition 2).

Figure 2(b) and 2(c) show the layout of measuring points of rotating cage. 2 microphones were arranged at 1 meter away from the rotating cage respectively. Accelerometer 3 was applied to the front panel of cooling unit to measure the acceleration of rotating cage along Z-direction. Accelerometer 4 was applied to the end of rotating cage shaft to measure the acceleration along Y-direction. Accelerometer 5 was applied to the side of rotating cage to measure the acceleration along X-direction.

Figure 2(d) shows the layout of measuring points of fan 1 which is installed on the support. Adjusting DC power supply voltage could change the speed. The working voltages (V) of condition 3 and 4 were 48. The net was installed in the fan inlet under condition 3. The net was installed in the fan outlet under condition 4. The working voltages (V) of condition 5, 6, 7, 8, 9 and 10 were 48, 42, 36, 32, 28 and 24 in order, under which the fan had no net. To study the influence of net and speed on the noise of fan. The noise test and analysis of fan were carried out under the above operating conditions [12].

For analyzing the influence of sound-absorbing material on the noise of cabinet, melamine sound-absorbing cotton was coated on the inner wall of cabinet with thickness of 10 mm. Figure 2(a) shows the layout of measuring points. The noise test and analysis of cabinet were carried out under two operating conditions of closing the front door (condition 11) and opening the front door (condition 12).

4. Noise analysis for ship signal cabinet

4.1. Noise spectrum of cabinet

Figure 3(a) shows the noise spectrum of cabinet with the front door closed. The SPL was 57.26 dB(A). The noise spectrum distribution of measuring points 1, 2 and 3 were discrete. The SPL were concentrated in the 0 to 2500 Hz range. When the frequency was more than 2500 Hz, the SPL of cabinet gradually attenuated with the increase of frequency. The SPL of cabinet was the largest at 762 Hz which was \( f_b \) of fan 1. Peak frequencies (Hz) were 255, 1352, 1524, 762 and 1524 were multiple frequency of \( f_b \) of fan 1, 255, 1352 and 1524 were multiple frequency of \( f_s \) of fan 1. The SPL of \( f_b \) and \( f_s \) of fan 2 were small.

Figure 3(b) shows the noise spectrum of cabinet with the front door opened. The SPL was 69.91 dB(A). Compared with condition 1, the SPL were smaller in the 0 to 800 Hz range, but the SPL in the 2000 to 2500 Hz range became larger. Peak frequencies (Hz) were 762, 1352, 1524 and 2287. 762, 1524 and 2287 were multiple frequency of \( f_b \) of fan 1, 1352 was multiple frequency of \( f_s \) of fan 1.

4.2. Spectrum of vibration and noise of rotating cage

Figure 4 shows the spectrum of vibration and noise of rotating cage. The SPL was 73.1 dB(A). The SPL were small in the 0 to 800 Hz range. Peak frequencies of vibration and noise were multiple frequency of \( f_b \) and \( f_s \) of fan 1. The acceleration was the largest at 16 times \( f_b \) of fan 1.

4.3. Finite element model of cooling unit

Figure 5 shows the finite element model of cooling unit. Tetrahedral and hexahedral elements were used for meshing. The number of finite element units was 16376. In the finite element model of cooling unit, the fans were replaced by equivalent mass points. Figure 6 shows the first six-orders modal deformation cloud of cooling unit. Four holes of screw on the front panel and two pins on the
back were fixed. The first 15-orders modal frequencies (Hz) were 94.774, 187.52, 272.58, 319.84, 395.97, 459.44, 515.91, 544.16, 693.17, 713.41, 736.66, 772.03, 788.32, 845.67 and 850.72.

![Finite element model of cooling unit.](image1)

**Figure 5.** Finite element model of cooling unit.

![First six-orders modal deformation cloud of cooling unit.](image2)

**Figure 6.** The first six-orders modal deformation cloud of cooling unit.

![Sweep vibration transmissibility curve of cooling unit.](image3)

**Figure 7.** Sweep vibration transmissibility curve of cooling unit.

To verify the reliability of modal calculation data of cooling unit, sinusoidal sweep vibration tests of cooling unit in X-direction, Y-direction and Z-direction were carried out on the electrodynamics vibration generator, and sweep vibration transmissibility curve of cooling unit was analyzed to obtain its resonant frequency. Z-direction was the normal direction of fan installation surface, Y-direction was the normal direction of cooling unit front panel. The fixing mode of cooling unit was the same as simulation calculation. Accelerometers were installed on the electrodynamics vibration generator and the cooling unit to achieve input and output acceleration. Dividing output acceleration by input acceleration could generate vibration transmissibility curve which was shown in Figure 7.
Table 2 shows the comparison between test and calculated modal frequency. \( f_c \) is the calculated frequency. \( f_t \) is the test frequency. \( R_a \) is the modal mass ratio. Because the cooling unit was simplified in modal calculation, the fans were replaced by equivalent mass points, and the influence of damping on the modal parameters was not considered, there were deviations between test and calculated modal frequencies. For the first, the 9th and the 10th order whose modal mass ratio were large, the errors were less than 1%.

Table 2. Comparison between test and calculated modal frequencies.

| No | \( f_c \) /Hz | Z-direction \( f_t \) /Hz | \( R_a \)% | Y-direction \( f_t \) /Hz | \( R_a \)% | X-direction \( f_t \) /Hz | \( R_a \)% | Error% |
|----|--------------|-----------------|---------|-----------------|---------|-----------------|---------|--------|
| 1  | 94.774       | 95.5            | 48.5    | 0.8             |         |                 |         |        |
| 2  | 187.52       | 177.5           | 2.2     | 5.3             |         |                 |         |        |
| 3  | 272.58       | 252             | 4.5     | 7.6             |         |                 |         |        |
| 4  | 319.84       | 307             | 4.6     | 4               |         |                 |         |        |
| 5  | 395.97       | 369.5           | 1.7     | 6.7             |         |                 |         |        |
| 6  | 459.44       | 481.5           | 0.5     | 4.8             |         |                 |         |        |
| 7  | 515.91       | 488.5           | 6       | 5.3             |         |                 |         |        |
| 8  | 544.16       |                 |         |                 |         |                 |         |        |
| 9  | 693.17       | 692             | 12.6    | 0.2             |         |                 |         |        |
| 10 | 713.41       | 717             | 42.3    | 0.5             |         |                 |         |        |
| 11 | 736.66       |                 |         |                 |         |                 |         |        |
| 12 | 772.03       |                 |         |                 |         | 778.5          | 6.4     | 0.8    |
| 13 | 788.32       |                 |         |                 |         | 801.5          | 2       | 1.7    |
| 14 | 845.67       |                 |         |                 |         | 848.5          | 3.2     | 0.3    |
| 15 | 850.72       |                 |         |                 |         | 852            | 0.4     | 0.2    |

The errors were less than 7.6%. The first-order calculation and test modal frequency were in good agreement. Figure 3, Figure 5 and Figure 7 show that noise and vibration energy corresponding to the modal frequency of cooling unit were small, that is, cooling unit did not generate resonance which leads to the larger noise of cabinet.

According to the above results, the peak frequencies of vibration and noise spectrum of cabinet and rotating cage were multiple frequency of \( f_b \) and \( f_s \) of fan 1, and the noise spectrum distribution was discrete, and concentrated in the 0 to 2500 Hz range. When closing the front door, the cabinet was airtight and could isolate sound energy emitted by the fan. When opening the front door, the sealing property of cabinet became worse, but sound energy could still be shielded through the left and right sides of cabinet. Therefore, when the front door was closed, its SPL was the smallest and 15.84 dB(A) smaller than that of rotating cage. When the front door was opened, its SPL increased and 3.19 dB(A) smaller than that of rotating cage. The noise of cabinet was mainly related to aerodynamic noise of fan 1. The following research on noise control of cabinet was carried out from the optimal design of fan installation structure and the application of sound-absorbing cotton.

5. Noise reduction analysis for ship signal cabinet

5.1. Optimal design of axial flow fan installation structure

As can be seen from Figure 8 and Table 3, compared with condition 5, the SPL of condition 3 increased 2.91 dB(A), the energy ratio of high-frequency noise (6000 to 14000 Hz) increased significantly, and the sound was sharp. Peak frequencies in high-frequency noise were multiple frequency of \( f_b \). The SPL of condition 4 was close to that of condition 5. The SPL of conditions 6 to 10 were concentrated in the 0 to 4000 Hz range, and peak frequencies were the multiple frequency of the corresponding \( f_b \) and \( f_s \). The SPL of background noise mainly came from the cooling fan of DC power supply. The SPL at peak frequencies of background noise were smaller in the noise spectrum of
conditions 3 to 10, which indicates that the noise of fan was less disturbed by the background noise of cooling fan of DC power supply.

Figure 8. Noise spectrum of fan under different operating conditions.

Table 3. SPL and peak frequencies of fan under different operating conditions.

| Operating condition | SPL / dB(A) | Peak frequencies / Hz | \( N / \text{rpm} \) | \( f_s / \text{Hz} \) | \( f_b / \text{Hz} \) |
|---------------------|-------------|-----------------------|-----------------|----------------|----------------|
| 3                   | 67.82       | 6092.6854,7616,8378,9140,9902,10664,11426,12188,12950,13712 | 5080            | 84.7           | 762            |
| 4                   | 64.17       | 676,762,1352,1524,2030,2286,3050 | 5080            | 84.7           | 762            |
| 5                   | 64.91       | 676,762,1352,1524,2030,2286,3050 | 5080            | 84.7           | 762            |
| 6                   | 64.22       | 674,758,1348,1515,2022,2273,3030,4546 | 5049            | 84.2           | 757.4          |
| 7                   | 64.01       | 660,743,1486,1980,2228,2970,3712 | 4954            | 82.6           | 743.1          |
| 8                   | 61.08       | 607,682,1213,1364,1819,2047,2728,3411,3640 | 4546            | 75.8           | 681.9          |
| 9                   | 58.97       | 551,619,1101,1239,1858,2477 | 4131            | 68.9           | 619.7          |
| 10                  | 57.81       | 483,543,1087,1448,1631,1933,2170 | 3621            | 60.4           | 543.2          |
| Background noise    | 43.31       | 367,734,1052,1262,1468 | -               | -              | -              |
According to the above results, when the net was installed in the fan inlet, the net changed the flow field in the fan inlet, which reduced the fan inlet air volume and increased the noise obviously, and the SPL of 6000 to 14000 Hz increased significantly. When the net was installed in the fan outlet, the net isolated sound energy of fan, so the SPL was slightly reduced. According to Equation (1), the SPL became smaller with the decrease of speed, which was consistent with the change trend of noise test data in Figure 8. Peak frequencies at different speeds were multiple frequency of \( f_b \) and \( f_s \), the SPL became smaller with the increase of frequency, and the SPL was the largest at \( f_s \). On the premise of meeting the cooling demand of electronic modules of cabinet, the SPL of cabinet could be reduced when the speed of fan decreased and the net was not installed in the fan inlet.

5.2. Analysis for noise reduction material of signal cabinet

Melamine sound-absorbing cotton belongs to porous sound-absorbing material. Compared with the low-frequency noise, sound-absorbing cotton has better sound absorption effect in the high-frequency noise. With the increase of sound-absorbing cotton thickness, sound-absorbing coefficient becomes larger but gradually tends to be stable. Due to the limitation of internal installation space of cabinet, 10-mm-thick melamine sound-absorbing cotton is selected to be coated on the inner wall of cabinet for the analysis of noise reduction materials.

![Figure 9. Noise spectrum of cabinet with sound-absorbing cotton.](image)

Figure 9 shows the noise spectrum of cabinet with sound-absorbing cotton. Compared with condition 1, the SPL of condition 11 was 53.02 dB(A) and decreased 4.24 dB(A). Condition 11 and 1 had the same peak frequencies, and the SPL of condition 11 was significantly smaller than that of condition 1 in the 200 to 2000 Hz range. Compared with condition 2, the SPL of condition 12 was 67.79 dB(A) and decreased 2.12 dB(A), the noise spectrum distribution of condition 12 and 2 were the same, and the SPL of condition 12 was smaller than that of condition 2 in the 700 to 1600 Hz range.

It can be seen from the above that sound-absorbing cotton attached to the inner wall of cabinet could reduce its noise, and sound-absorbing cotton had no effect on the peak frequencies in the noise spectrum of cabinet. When the front door was opened, sound-absorbing cotton on the left and right inner walls of cabinet absorbed sound energy of fan, but sound energy at the front door was not reduced, so the SPL of microphone 1 was obviously larger than that of microphone 2 and 3. When the front door was closed, sound-absorbing cotton on the inner wall of the front door, left side and right side absorbed sound energy of fan, so the SPL of microphone 1, 2 and 3 were close, and the effects of noise reduction were obvious in the 200 to 2000 Hz range when the front door was closed.

6. Conclusions

According to the noise test data of ship signal cabinet, the SPL of opening the front door of cabinet was 12.65 dB(A) larger than that of closing the front door. The vibration and noise spectrum of three operating conditions (opening the front door, closing the front door and rotating cage) were discrete and concentrated in the 0 to 2500 Hz range. The peak frequencies of vibration and noise spectrum
were multiple frequency of fundamental frequency and shaft frequency of axial flow fan. The errors of calculated and test modal frequencies of cooling unit in the rotating cage were less than 7.6%, and calculated modal data were reliable. Noise and vibration energy corresponding to the modal frequencies of cooling unit were small, that is, the cooling unit did not generate resonance, which leads to the increase of cabinet noise. The noise of cabinet was mainly related to the aerodynamic noise of axial flow fan.

When the net was installed in the fan inlet, the SPL increased 2.91 dB(A), and the high-frequency energy component was more in the noise spectrum, and the noise was sharp. The SPL became smaller with the decrease of speed. The peak frequencies at different speeds were multiple frequency of fundamental frequency and shaft frequency of fan. The SPL was the largest at fundamental frequency and attenuated with the increase of frequency. That is, the noise of fan came from its rotating blades. On the premise of meeting the cooling demand of electronic modules of cabinet, the SPL of cabinet could be reduced when the speed of fan decreased and the net was not installed in the fan inlet. When melamine sound-absorbing cotton was coated on the inner wall of cabinet, the SPL of cabinet decreased, and the effects of noise reduction were obvious in the 200 to 2000 Hz range when the front door was closed.

References
[1] Liu G, Wang L and Liu X M 2020 Numerical investigation on effects of blade tip winglet on aerodynamic and aeroacoustic performances of axial flow fan Journal of Xi'an Jiaotong University 54(7) pp 104-112
[2] Hamakawa H, Shiotzuki M, Adachi T, et al. 2012 Correlation between aerodynamic noise and velocity fluctuation of tip leakage flow of axial flow fan Open Journal of Fluid Dynamics (2) pp 228-234
[3] Zhang L and Jin Y Z 2012 Numerical investigation on vortex structure and aerodynamic noise performance of small axial flow fan Open Journal of Fluid Dynamics (2) pp 359-367
[4] Li Y, Liu Z Z, Xu Y B, et al. 2019 Large eddy simulation of the flow field in centrifugal fan and numerical prediction of noise in pipe Chinese Journal of Ship Research 14(4) pp 91-97+154
[5] Sun Y Z, Xiao S D and Xu X K 2016 Analysis of aerodynamic noise induced by rotating blades of an axial fan Noise and Vibration Control 36(4) pp 124-128
[6] Ye X M, Zhang J K and Li C X 2017 Aerodynamic acoustic characteristics of an axial flow fan with different blade tips Journal of Chinese Society of Power Engineering 37(7) pp 558-568
[7] Mao Y J and Qi D T 2009 Review of aerodynamic noise in turbomachinery Advances in Mechanics 39(2) pp 189-202
[8] Zhong Y H, Li Y N and Gao F 2019 Study on discrete noises and blade distribution characteristics of vehicle axial fans China Mechanical Engineering 30(9) pp 1072-1080+1089
[9] Ouyang H, Tian J, Wu Y D, et al. 2009 Research of aerodynamic noise source of low speed axial fans based on vortex-sound theory Journal of Engineering Thermophysics 30(5) pp 765-768
[10] Li X X, Gao G J, Kou Z M, et al. 2019 Impact of blade tip clearance on aerodynamic noise of counter-rotating axial fan Chinese Hydraulics & Pneumatics (7) pp 44-49
[11] Li L L, Huang Q B and Qiao Y F 2006 Research on model of the vortex noise of axial fan blade and its characteristics China Mechanical Engineering 17(10) pp 1056-1059
[12] Ito T, Minorikawa G and Fan Q Y 2009 Experimental research for performance and noise of small axial fan International Journal of Fluid Machinery and Systems 2(2) pp 136-146