AN INTAKE SILENCER FOR THE CONTROL OF MARINE DIESEL TURBOCHARGER COMPRESSOR NOISE

Liu Chen, Cao Yipeng, Zhang Wenping and Zhao Xiaochen

Harbin Engineering University, College of Power and Energy Engineering, Harbin, Heilongjiang, P.R. China
email: yipengcao@hrbeu.edu.cn

Shen Hua

Chongqing Jiangjin Shipbuilding Industry Co. Ltd, Chongqing, P.R. China

Turbocharger noise is one of the main noise sources in turbocharged diesel engine, in which the compressor intake noise plays a leading role in the turbocharger noise. Therefore, it is of vital necessity to install an intake silencer in the inlet air line of compressor. This paper describes the design of intake silencer, which is a radial flow silencer. The noise attenuation empirical formula of the radial flow silencer is used to preliminary predict the acoustic performance of the silencer. Besides, the acoustic performance is predicted by Finite Element Method in order to get noise attenuation. Finally, a series of experiments are designed and performed to verify the accuracy of numerical simulation and the effectiveness of silencer, including compressor open inlet, silencer installed and replacement tube installed. Through comparing the results of numerical simulation with the experiment values, it is shown that the silencer can reduce noise effectively.

1. Introduction

Turbochargers have been extensively used in marine diesel engines to improve their performances, a turbocharger can increase engine power and torque by 20%~30% against a naturally aspirated engine of the same size, therefore turbocharging techniques are more and more widely used [1]. However, though engine efficiency and power output is increasing, the turbocharger has a lot of negative impacts such as compressor noise. The excessive noise has been one of the most important present-day problems that a number of experimental and analytical studies have been conducted to understand the characteristics of centrifugal compressor noise. Raitor studied compressor noise with a series of experiments and the results showed that that Blade Passing Frequency (BPF) noise occupies the dominant position of the spectral components governing the overall noise level of centrifugal compressor [2]. The numerical prediction of centrifugal compressor noise was done by Sun and the same conclusions was achieved [3]. In both cases a possible noise problem can be a strong BPF typically in the KHz range.

In order to address these noise issues, various silencers are designed and mounted on compressor. Raimo Kabral developed a novel type of compact dissipative silencer that is based on a combination of a micro-perforated tube backed by a locally reacting cavity, due to the high damping achieved at the Cremer optimum it is easy to create a compact silencer with a significant damping (say > 30 dB) in a range than an octave [4,5]. Eric introduced a silencer design based on a combination of Herschel-Quincke tube and two quarter-wave resonators, this device has a transmission loss of over 15 dB from 1600 to 3400Hz when it is applied on the automotive engines [6]. Lee designed a multi-chamber silencer that three Helmholtz resonators are implemented in series and each resonator consists of a chamber and a number of slots, the experimental and analytical results have provided a satisfactory control of the whoosh noise [7]. Tian Sheng designed a reactive silencer to reduce turbocharger synchronous noise generated from compressor pressure pulsations, the effectiveness of the silencer is tested on a vehicle and the result showed that the silencer can reduce noise effectively in the frequency range of 1000-5000Hz, and the transmission loss reaches 15dB for turbo speed between 150 and 190k rpm [8]. The above researches obtained very good results in the turbocharger noise control, but they are mainly concentrated in automotive turbocharger, in this paper the intake silencer used in middle and low speed diesel engine turbocharger is studied.

2. Research object

Under the self-cycling test condition, the noise variation between 0 and 5k Hz was measured at different speeds. In Fig. 1, the experimental results showed that compressor noise is mainly high frequency noise, and BPF noise occupies the dominant position of the spectral components, which could be seen in Fig. 2.

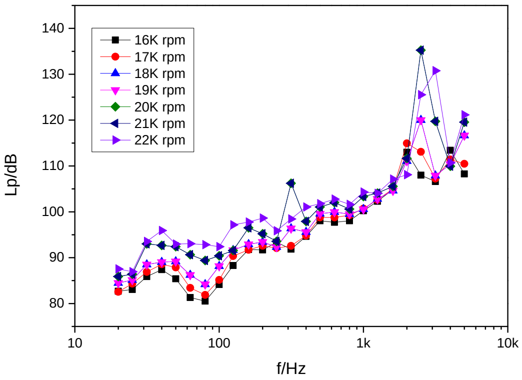


Figure 1: noise variation at different speeds when open inlet

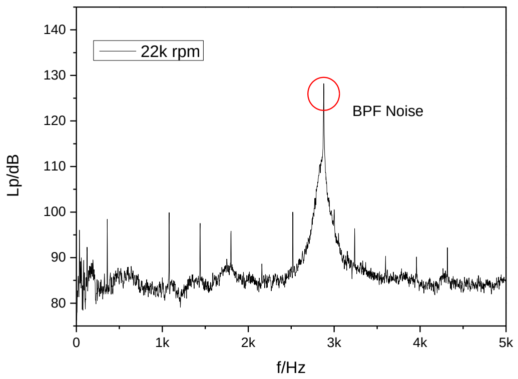


Figure 2: noise spectrum of compressor at 22k rpm

The research object in this paper is an intake silencer mounted in the inlet air line of turbocharger compressor, the turbocharger is a high flow rate and high pressure ratio marine diesel engine turbocharger that consists of centrifugal compressor and axial turbine. The intake silencer was designed as a dissipative silencer that is shown in Fig. 3, the interior absorbing material can absorb compressor aerodynamic noise energy to suppress noise radiation. There are 37 sound absorber blades that are composed of external micro perforated panel and internal aluminium-silicate rock wool, the whole structure indicated the silencer a typical runoff silencer. Though the silencer absolute structure size is relatively large but it is quite compact compared with the turbocharger that makes it adapts to the ship engine cabin.

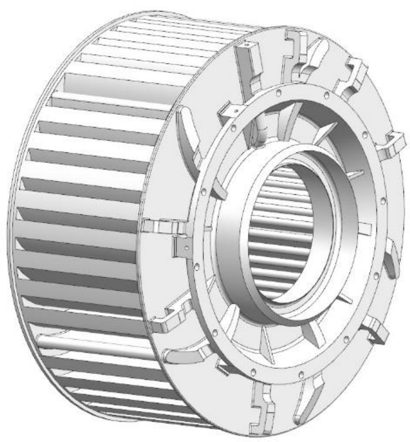


Figure 3: research object

3. Empirical formula prediction

The commonly used empirical formula of acoustic attenuation for the dissipative silencer is:

$$L_{NR1} = K \cdot \frac{P}{S} \cdot l \tag{1}$$

Where K is attenuation coefficient that mainly depends on the sound absorption coefficient, different empirical formulas have different K values and table 1 lists the K values of the 4 empirical formulas used in this paper [9]; P is the equivalent perimeter of the silencer flow passage; S is the equivalent area of the silencer flow passage; l is the length of the silencer absorption flow passage. α_0 is the sound absorption coefficient.

Table 1: K values of the empirical formulas

empirical formula	Zellen	R. Rogers	H. J. Sabine	A. N. BaLoB
K	$1.5\alpha_0$	$4.34\times\frac{1-\sqrt{1-\alpha_0}}{1+\sqrt{1-\alpha_0}}$	$1.5\alpha_0^{1.4}$	$1.1\alpha_0$

In the silencer, the flow passage area of the sound propagation direction increases with the increase of the radius and the acoustic attenuation from the acoustic inlet to the acoustic outlet caused by the sound diffusion is:

$$L_{NR2} = 10 \cdot \lg \frac{S_2}{S_1} \tag{2}$$

Where S_1 is the area of acoustic inlet plane; S_2 is the area of acoustic outlet plane.

The sound absorption coefficient of sound absorption material used in this silencer and the noise reduction (NR) calculated by empirical formulas were shown in Fig. 4-5. The simulation results of every empirical formula show the same trend of noise reduction.

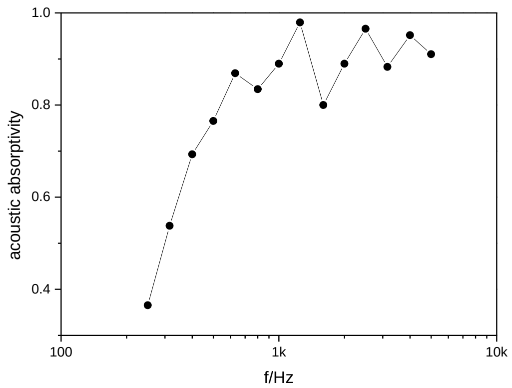


Figure 4: sound absorption coefficient

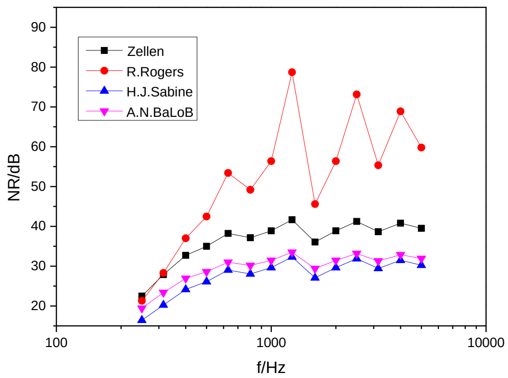


Figure 5: noise reduction

4. Numerical simulation

Different numerical techniques, e.g. FEM, single and multi-domain BEM, Green’s function method, Munjal’s approach and the transfer matrix method, Fourier boundary element method have been applied to analysis and design of complicate acoustic system [10]. In this paper, the silencer structure is complicated and thus FEM was chosen to calculate noise attenuation. A finite element model is set up to calculate the noise reduction (NR) of the silencer, which is shown in Fig. 6, the direction of air flow is opposite to the noise radiation in real work. The fluid material and absorbing material were defined through the density and sound speed, the fields were split into hybrid grid elements and the maximum mesh size is 10mm in order to ensure the upper limit frequency of calculation accuracy beyond 5000Hz, which is determined by Nyquist's Theorem. A unit particle velocity boundary was used in acoustic inlet and the acoustic outlet set as sound absorbing boundary.

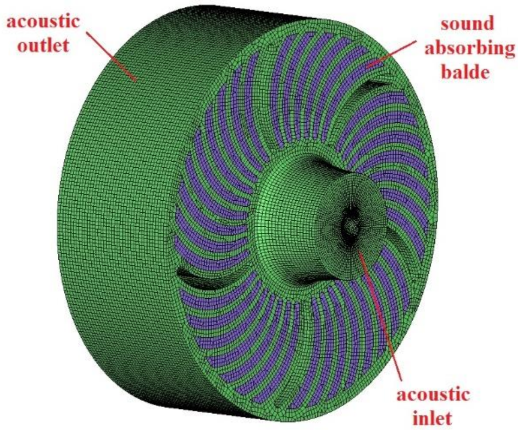


Figure 6: finite element model

The absorbing material used here is aluminium-silicate rock wool and its density is 89g/L, the empirical formula of aluminium-silicate acoustic characteristics under different frequencies are as Eq. 3 and Eq. 4, including the complex impedance and complex wave number.

$$\tilde{z}/z_0 = 1 + 42.169 f^{-0.7295} - i \cdot 20.66 f^{-0.5429} \tag{3}$$

$$\tilde{\mathrm{k}}/\mathrm{k}_0 = 1 + 53.991 f^{-0.6663} - i \cdot 61.85 f^{-0.6465} \tag{4}$$

Where $\tilde{z}$ and z_0 were the complex impedance of air and sound-absorbing material respectively, $\tilde{\mathrm{k}}$ and k_0 were the complex wave number of air and sound-absorbing material respectively, f was frequency.

The numerical prediction was based on the FEM method, the particle velocity was set in the acoustic inlet and the acoustic outlet was set as non-reflecting boundary. The aluminium-silicate acoustic characteristics under different frequencies were attached to the sound absorbing blades. Finally NR of the silencer was obtained and shown in Fig. 7 with 1/3 octave form. It could be seen that there is a big noise attenuation in high frequency.

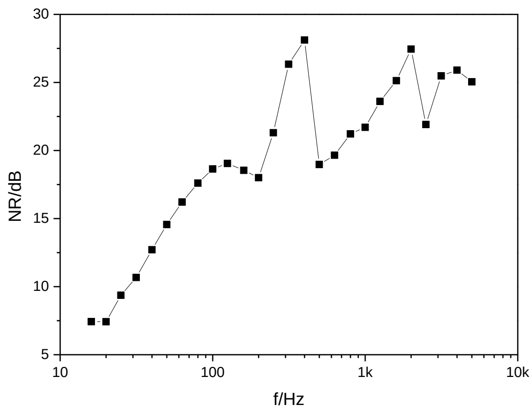


Figure 7: calculated NR of the silencer

5. Experimental study

5.1 Brief introduction of test bench

For testing the effectiveness of the silencer and the validity of numerical simulation presented in this paper, the compressor noise experiment facilities was designed and shown in Fig. 8. This test bench is a professional turbocharger performance test bench [11-12].

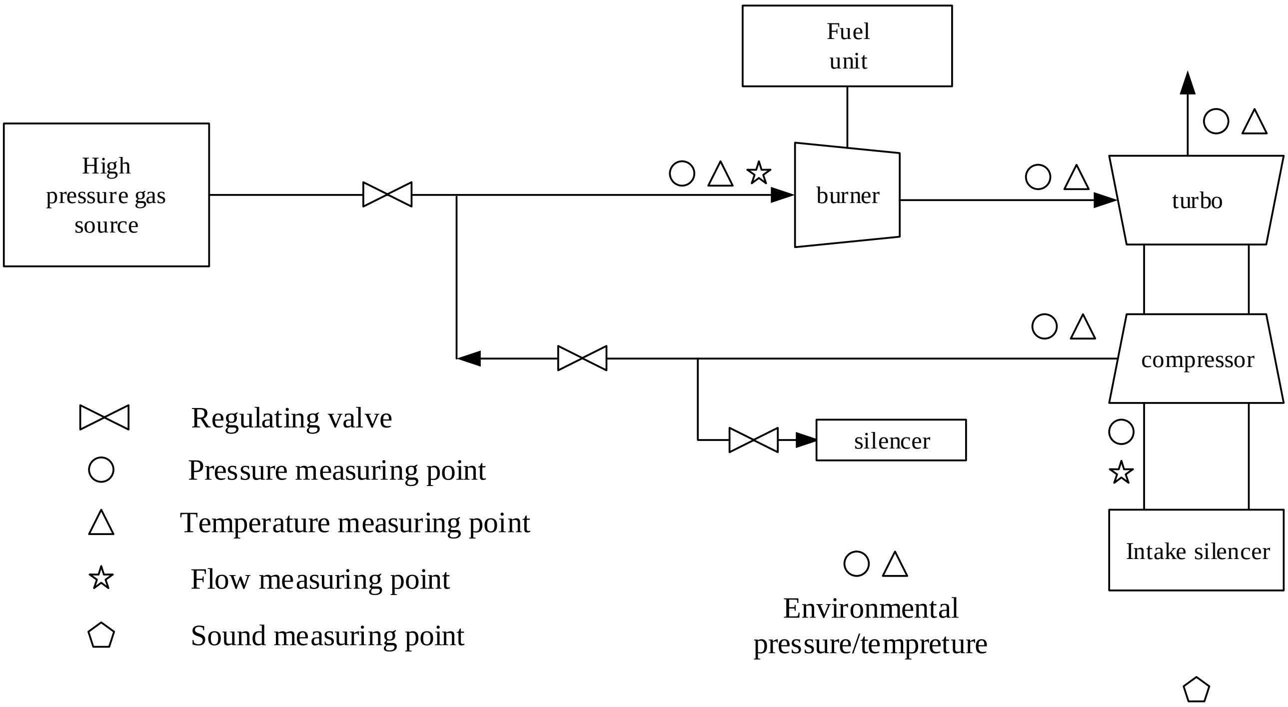


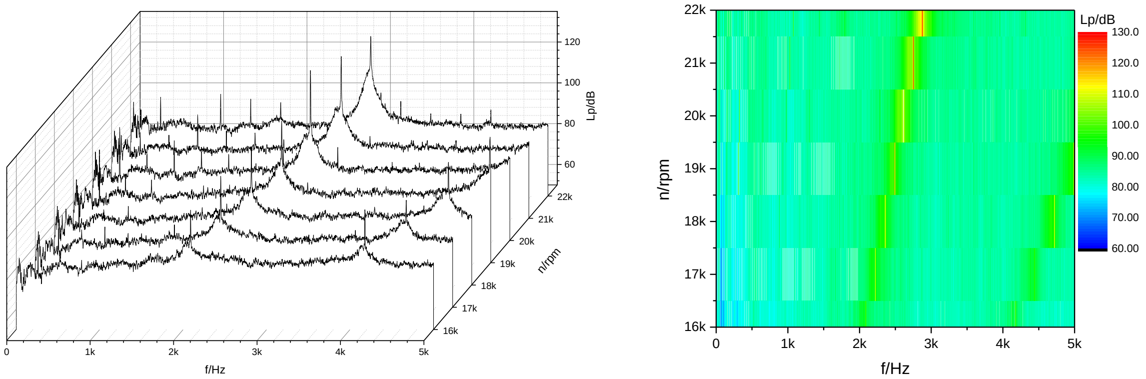
Figure 8: test bench

5.2 Measuring point arrangement

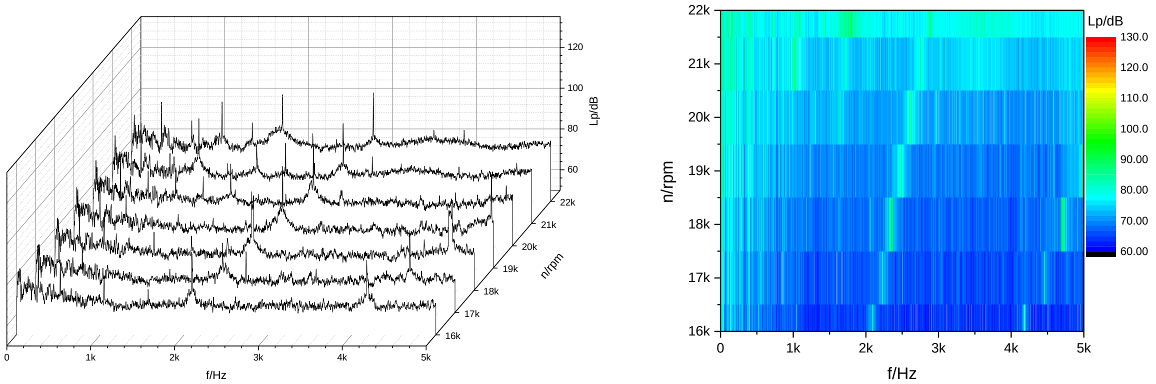
The sound measuring point is set on compressor axis and the distance is 1m, there were three different test conditions including compressor open inlet, silencer installed and replacement tube installed. Under each test condition, the noise was measured at various speeds (from 16k rpm to 22k rpm, interval 1k rpm) with turbocharger self-circulation operation. When compressor open inlet, the compressor noise can be radiated directly into the environment and thus the ideal compressor air radiation noise is obtained at the measuring point. The second test condition is silencer installed that used for comparison with the measurement results of the compressor open inlet in order to illustrate the effectiveness of the intake silencer. Although the silencer is a typical dissipative silencer, the silencer structure still has a certain noise reduction effect, the third test condition is replacement tube installed and the replacement tube is the residual part of the silencer that the sound absorption structure removed.

5.3 Experimental results

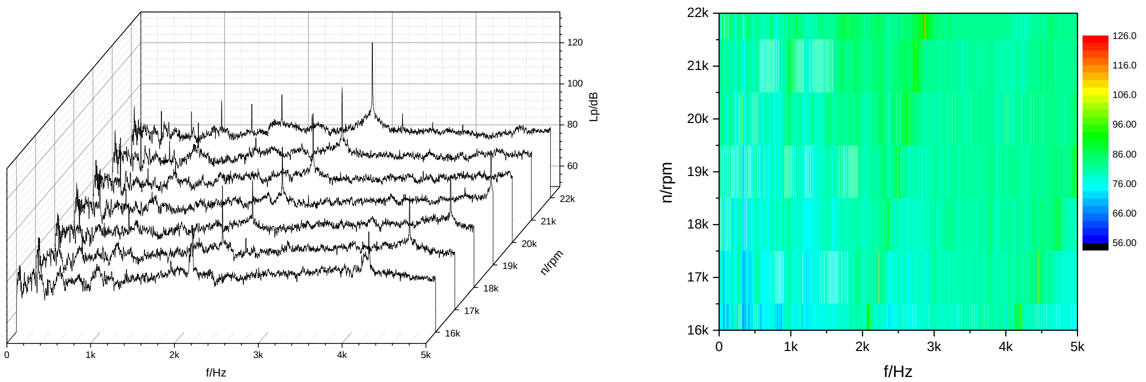
With the experimental facilities, the effectiveness of silencer was tested, there were three different test conditions including compressor open inlet, silencer installed and replacement tube installed. In Fig. 10, under the three test conditions, the compressor noise at different speeds were measured and the results were shown as waterfall and contour maps in order to compare with each other. The former can clearly reveal the change of noise with frequency and speed increase and the latter can be more clearly when compared different test conditions.



(a) compressor open inlet



(b) silencer installed

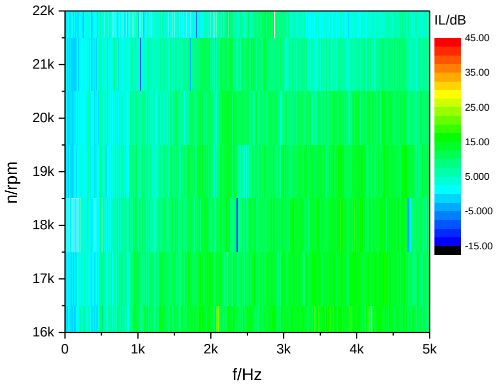


(c) replacement tube installed

Figure 9: sound spectrum at different speeds under three test conditions

In Fig. 9, it can be seen from the three graphs on the left that BPF noise was effectively suppressed that the noise peak decreased significantly. It could also be seen from the graphs on the right, there is a clear peak band in Fig. 9 (a) that it moves to high frequency gradually as the speed increases, in the second graph, the peak band basically disappeared with the silencer installed. The last figure is the result of the replacement tube, although the noise is reduced when compared with the compressor open, the effect is far better with the silencer installed.

In order to get a more intuitive explanation of the silencer effect, using insertion loss (IL) for the evaluation of noise attenuation, the insertion loss is the sound pressure level difference measured before and after the installation of the silencer. At the same time, the sound pressure level reduction measured when compressor open inlet and silencer installed was shown in Fig.11.



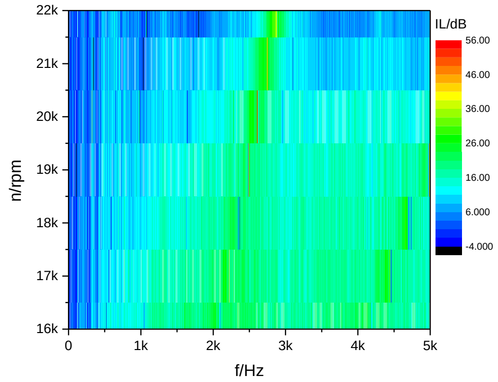
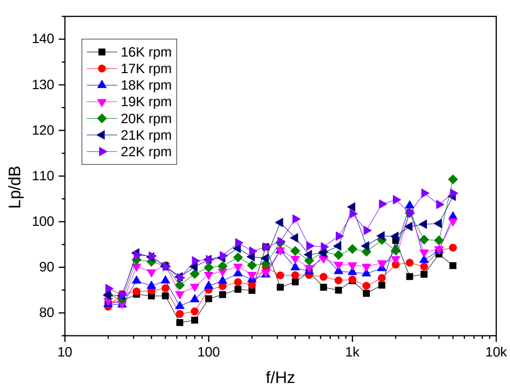


Figure 10: IL

Figure 11: NR between compressor open inlet and silencer installed

Finally, the experimental results measured when compressor open inlet and silencer installed were compared with the original noise distribution mentioned in the introduction in order to illustrate the effect of silencer on high frequency noise, the results of the latter was shown in Fig. 12 and Fig. 13. Compared these results, it could be seen that the high frequency noise in supressed effectively and the amplitude of high frequency noise decreased obviously.



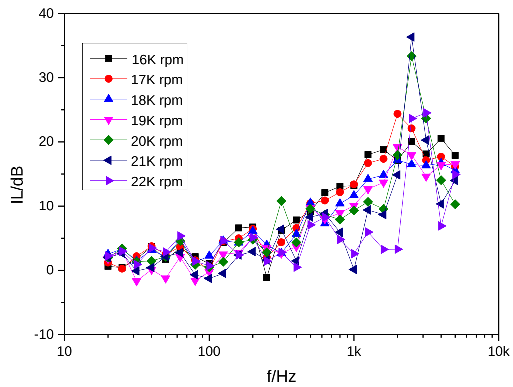


Figure 12: noise variation at different speeds when silencer installed

Figure 13: NR between silencer installed and replacement tube installed

Several results were chosen for a comparison between the results obtained by formula, FEM and experiments that is shown in Fig.14, all simulation results is higher than experiment because all methods simplify the actual structure, experimental results show a big reduction in bands 2500Hz and 3150Hz, these bands contains BPF. It can also be seen that FEM method simulation result has a good agreement with experiment in these bands.

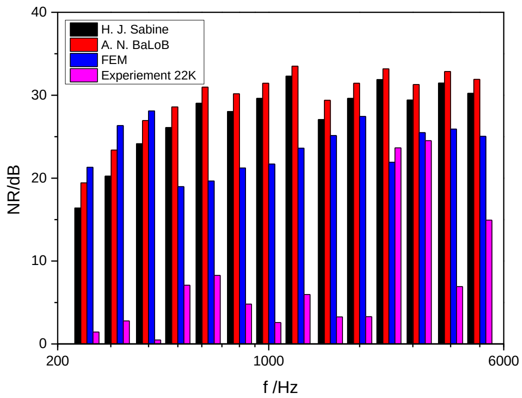


Figure 14. Comparison between the results obtained by formula, FEM and experiments

6. Conclusions

An intake silencer for the control of marine diesel turbocharger compressor noise was researched in this paper. Empirical formula prediction, numerical simulation and experimental study were finished and all results were discussed:

(1) Empirical formula prediction and numerical simulation could be used to predict the noise reduction effect in the early stage of silencer design.

(2) The dissipative silencer has a significant supress effect on the compressor noise, especially the high frequency noise.

(3) Silencer structure has a good noise reduction ability, but the sound absorbing material can better suppress the high-frequency noise.

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