Influences of Dis-tuned Tip Clearance on the Discrete Aerodynamic Noise in Centrifugal Compressor

Mingxu Qi*, Meijie Zhang, Chaochen Ma

ISROMAC 2016

International Symposium on Transport Phenomena and Dynamics of Rotating Machinery

Hawaii, Honolulu

April 10-15, 2016

Abstract

With the purpose of decreasing the discrete aerodynamic noise in turbocharger compressor, the influences of dis-tuned impeller tip clearances on the aerodynamic noise are numerically and experimentally investigated. The dis-tuned tip clearance configuration is realized by ensuring either a bigger tip clearance size for main blades while a smaller one for splitter blades, or a smaller tip clearance size for main blades while a bigger one for splitter blades. The influences of dis-tuned tip clearance on centrifugal compressor performances are validated and results indicate that the dis-tuned tip clearance method used in current study has minor influences on compressor performances. Based on full unsteady CFD results, near field discrete aerodynamic noise analysis is performed on the tip clearance dis-tuned compressors, while far field aerodynamic noise is experimentally investigated. Both numerical and experimental results show that the dis-tuned tip clearance configurations have significant influences on compressor discrete aerodynamic noise, with maximum noise SPL decreased by 8 dB under the compressor operation points investigated in current research..

Keywords

Centrifugal Compressor —Dis-tuned Tip Clearance —Discrete Aerodynamic Noise

School of Mechanical Engineering, Beijing Institute of Technology, Beijing, China

***Corresponding author**: qimx@bit.edu.cn

INTRODUCTION

With the emission restriction standard being more and more strict, turbocharging technology has been widely used in boosted engines for passenger and commercial vehicles and has the trend to be a standard technology to match the even more strict emission restrictions. However, due to the high rotating speed and highly strong unsteady flow in compressor and turbine, the turbocharger acts as one of the major noise source in the engine system and introduces discrete noises and turbulence noises. The aerodynamic noise generated by turbocharger compressor consists of broadband noise and discrete noise. As a major noise source, the discrete noise level is related with impeller rotating speed and blade number. Liu investigated the aerodynamic noise in centrifugal compressor with numerical methods and concludes that the maximum sound power is generated by the impeller[1]. Wen studied the aerodynamic noise of centrifugal compressor by experimental method and results show that there is narrow band tip clearance noise at 1/2 blade passing frequency, and its amplitude increases with rotating speed increasing[2]. Reference [3] carried out experimental investigation on centrifugal compressor noise, and results show that dominant noise generation mechanisms depend on the tip speed Mach number at the compressor inlet. Meanwhile, rotor-stator interaction is also considered to be a major noise source in turbomachinery. Dipole and monopole tonal noise in the frequency domain is predicted in [4] by an aero-acoustic model based on the FW-H equation, and the results show that the interaction between the impeller and the diffuser plays an important role in centrifugal fan noise and is deemed to be the origin of the tonal noise. As another similar evidence, interaction between impeller and volute tongue is also experimentally proved to be a strong noise source in low speed centrifugal fan[5]. In order to reduce or control the level of aerodynamic noise, a lot of treatments or approaches are investigated. An optimization model of the blade circumferential angle was developed to minimize the A-weighted sound level and good results were achieved in reference [6]; Active control on the tip clearance flow with air injection was used to reduce the noise level and improve the performances on an axial flow compressor[7], which was proved to be successful; Wake generator was used in reference [8] to adjust the phase of the impeller blade passing frequency in a fan and the sound pressure level was reduced by 8dB. Such approaches are effective on reducing compressor noise level, but the corresponding structures are normally applied to large sized axial compressors or fans and are not practical for small size centrifugal compressors with high rotating speeds. Considering the fact that the discrete noise in centrifugal compressor is with frequency of BPF and with the purpose of dis-tuning the tip clearance flow, dis-tuned tip clearance method is utilized in current research to investigate its influences and potential effects on centrifugal compressor noise reduction. The dis-tuned tip clearance is realized by un-equivalent clearance size setting for the main blades and splitter blades, and both numerical end experimental investigations are thus carried out.

1. CASE DESCRIPTION

The tip clearance dis-tuned methods is utilized on a turbocharger centrifugal compressor in current study. The centrifugal compressor op erates from rotating speed of 40000 r/min to 90000 r/min, with impeller diameter being 90mm. The impeller consists of 7 main blades and 7 splitter blades, with designed tip clearance size being

0.5mm. Two tip clearance dis-tuned models are developed by adjusting the actual tip clearance of main blades and splitter blades. Investigated models are summrized as follows: MODEL 1, the tip clearance is set to be 0.4mm for main blades and 0.6mm for splitter blades; MODEL 2, the prototype of the compressor, with tip clearance being 0.5mm both for main blades and splitter blades; MODEL 3, the tip clearance is set to 0.6mm for main blades and 0.4mm for splitters. Numerical siumulations are carreid out on the compressor stage (including impeller, vaneless diffuser and volute) for all the different settings. For experimental study, the impeller with these 3 kinds of tip clearance settings are machined with 5-axial milling method, and during the test, only the impeller is replaced while other components of the compressor are retained.

Meridional dimensions of the investigated models are illustrated in figure 1 and described in table 1.

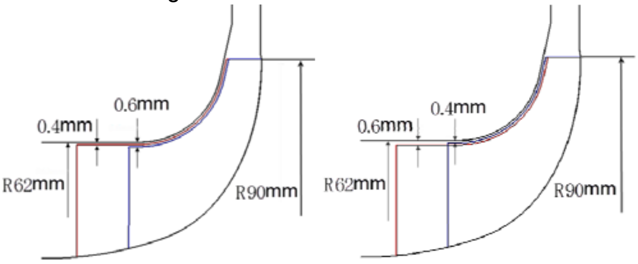


Figure 1. Meridional dimensions of the centrifugal compressor (left: MODEL 1, right: MODEL 3)

Table 1. Tip clearance settings

Model	Tip clearance/mm	
	Main blades	Splitter blades
1	0.4	0.6
2-Prototype	0.5	0.5
3	0.6	0.4

2. NUMERICAL AND EXPERIMENTAL SETUP

2.1 Numerical Method

Flow solver EURANUS, integrated in the NUMECA FINE/Turbo CFD package, is used in current research. The solver is based on a structured multi-block, multi-grid approach, and solves the time-dependent Reynolds averaged Navier-Stokes equations using finite volume method, with either the algebraic turbulence model of Baldwin-Lomax, one equation model of Spalart-Allmaras or two-equation models of k-ε models used for closure. For the present calculations, the Spalart-Allmaras turbulence model is selected and the second-order central scheme with addition of second and fourth order artificial dissipation is used. V-cycle multi-grid technology, coupled with local time stepping and implicit residual smoothing method is utilized for time marching acceleration. Computational domain consists of impeller, vaneless diffuser and volute. Mixing plane procedure is used at rotor-stator interface for compressor performances prediction and experimental validation. Non-Linear Harmonic (NLH) methods is used for unsteady flow simulations, which solves steady state N-S equations with derterministic stress prediction in time domain, and represents the flow into unsteady flow in frequency domain. In all the NLH simulations, third order harmonics and second order purtabations are modeled.

Multi-block structured mesh is used for the computational domain. Totally about 2.2 million grid cells are modelled for the compressor stage, with non-dimensional first layer cell size (y^+) being about 1-5. Corresponding solid surface grid of the computational domain is shown in figure 2.

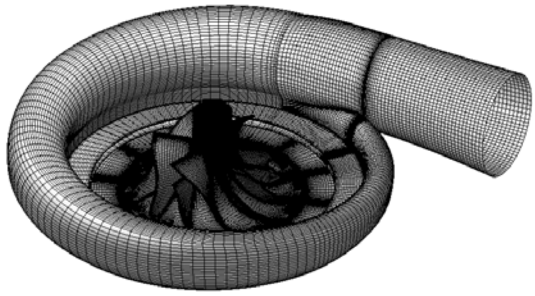


Figure 2. Solid surface grid of the computational domain

The near field aerodynamic of the compressor is calculated at the upstream of impeller inlet and Formulation 1A raised by Farassat [9] for FW-H equation is used for the calculation. To enable the caluclation of the near field noise at this planar surface, the pressure fluctuations are resolved by the unsteady flow simulations.

2.2 Experiment Setup

Both the compressor performance tests and the far field noise measurement are conducted on the compressor performance test rig developed in Beijing Institute of Technology. Total pressure and temperature are measured at the inlet and outlet of the centrifugal compressor for efficiency and pressure ratio testing. The air mass flow rate is measured by a double folium curve flow meter.

The far field aerodynamic noise is directly measured by using the SPL meter. During the tests, the background noise level is always much lower than that of the compressor, which fits the requirement of noise measuring standard. Two SPL meters are placed on the same planar of the compressor shaft. One of the SPL meter locates at upstream and 1 meter away from compressor inlet, another SPL meter locates at perpendicular direction and 1 meter away to the compressor shaft. Figure 3 shows the measuring positions of the measuring positions of compressor far field noise.

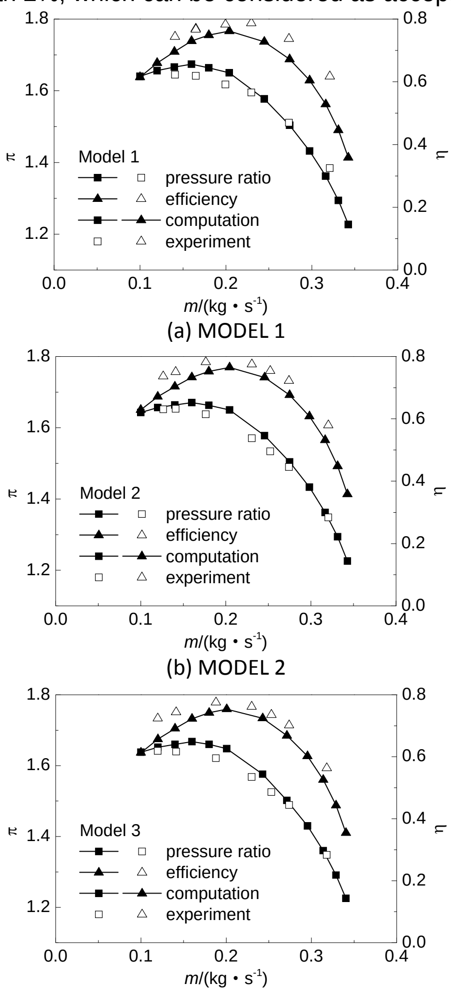
Figure 3. Far field noise measurement setup

3. RESULTS AND DISCUSSION

3.1 Experimental Validation on Numerical Methods

The predicted compressor total pressure ratio and total-to-total efficiency are validated with experimental data. The comparisons between numerical and experimental data of the three dis-tuned models under rotating speed of 60000 r/min are presented in figure 4.

As can see from the results, with the numerical methods described in above sections, the predicted pressure ratio and efficiency for MODEL 1, 2 and 3 match the experimental result reasonablly. The location of the peak efficiency point on the speed line and the trends of the curves (pressure ratio against massflow rate and efficiency against massflow rate) are correctly captured. The maximum error of efficiency between the predicted and tested data is within 4% and that of the pressure is less than 2%, which can be considered as acceptable.



(a) MODEL 1

(b) MODEL 2

(c) MODEL 3

Figure 4. Experimental validation on compressor performances

The calculated aerodynamic noise at passage passing frequency under peak efficiency point of 60000 r/min for the three models are compared with experimental results, as shown in figure 5. For all the models investigated, the numerical methods under predict the SPL of PPF noise by 13dB to 18dB. which However, the influences of the dis-tuned tip clearance on aerodynamic noise are proved to be in the same trend between numerical simulation and experiment. MODEL 1 is proved to decrease the compressor aerodynamic noise level while MODEL 3 tends to increase the noise level.

Considering the facts that the measurement is conducted in a non-anechoic space and the similar SPL vairation trends predicted by numerical and experimental approaches, the results can be considered to be reasonable.

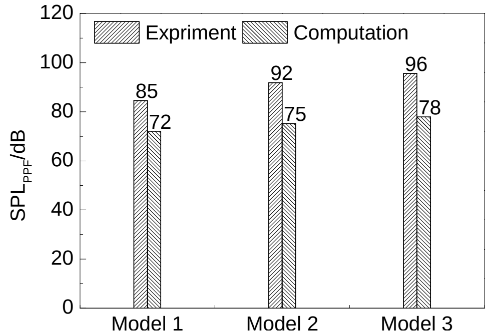
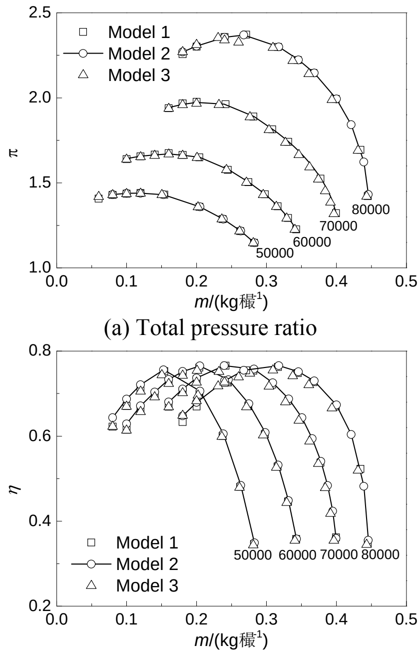


Figure 5. PPF SPL validation

3.2 Influences of Dis-tuned Tip Clearance On Compressor Performances

Even though the dis-tuned models can decrease the aerodynamic noise level in centrifugal compressor, the performances are still the most important things should be considered, i.e., reduction of noise should not lead penalties on compressor performances. Thus, the performances of the centrifugal compressor with 3 different dis-tuned tip clearance model under 4 different speed lines are experimentally tested and the pressure ratio as well as efficiency are compared in figure 6. The tested compressor rotating speeds increases from 50000 r/min to 80000 r/min.



(a) Total pressure ratio

(b) Total- total efficiency

Figure 6. Influence of dis-tuned tip clearance on compressor performance

As can be seen from the results, for all the operating speed lines being tested, MODEL 1 has almost the same performances with MODEL 2 (prototype) both on efficiency and pressure ratio. While slightly decreased pressure ratio and efficiency are observed for MODEL 3. The difference between MODEL 3 and prototype increases as compressor rotating speed increases. Anyway, the maximum decrease on pressure ratio and efficiency is still within 1%. Since the tip clearance size of main blades in MODEL 1 is decreased while in MODEL 3 is increased, it can be inferred that the size increase of main blade tip clearance has a greater impact on compressor performances than the increase of splitter blades tip clearance. Hence, to consider the influences of dis-tuned tip clearance on compressor performances, one can say that dis-tuned tip clearance method used in current research has minor influences on compressor performances, which can even be kept unchanged by decreasing main blades tip clearance while increasing that of splitter blades. With this conclusion, potential reduction of compressor discrete noise with the dis-tuned model mentioned above can thus be deemed as practical in engineering viewpoint.

3.3 Near Field Aerodynamic Noise Analysis

The near field discrete aerodynamic noise is numerically calculated at the monitoring surface upstream of the impeller inlet, and the distribution of the monitoring points on the surface is a 7X4 matix, which has 7 stations along circumferential direction and 4 points along spanwise direction on each station, as shown in figure 7. The peak efficiency and near choke operation points under 60000 r/min are investigated in this part. For each operation point, the data collection time is set to 0.001 second and resolution frequency is iset to 1000Hz.

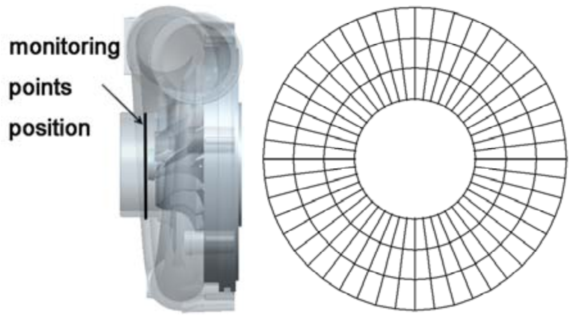


Figure 7. Near field discrete noise monitoring points

Figure 8-a and 8-b illustrate the specturm characteristics of the discrete aerodynamic noise for three different models at one monitoring point locates at 75% span. One can see that for all the models investigated, the discrete aerodynamic noise mainly apears at 7000Hz (the compressor passage passing frequency, PPF) and 14000Hz (the compressor blade passing frequency, BPF). Meanwhile, the SPL at PPF of 7000Hz tends to be much higher than that at BPF of 14000Hz. Comparisons between the dis-tuned tip clearance models and prototype indicate that for MODEL 1, the SPL at 7000Hz is decresed by 3.2dB at peak efficiency point and 2dB at near choke point, while for MODEL 2, the SPL at 7000Hz is increased by 2.5dB and 3dB respectively.

The SPL contour of near field discrete aerodynamic noise of 3 different models at the monitoring surface are shown in figure 9-a and 9-b, with results for peak efficiency and near choke operation points included. As a comparison between different operation points for the same model, it can be seen that for all the models investigated, the SPL level on the monitoring surface at near choke point is higher than that at peak efficiency point.

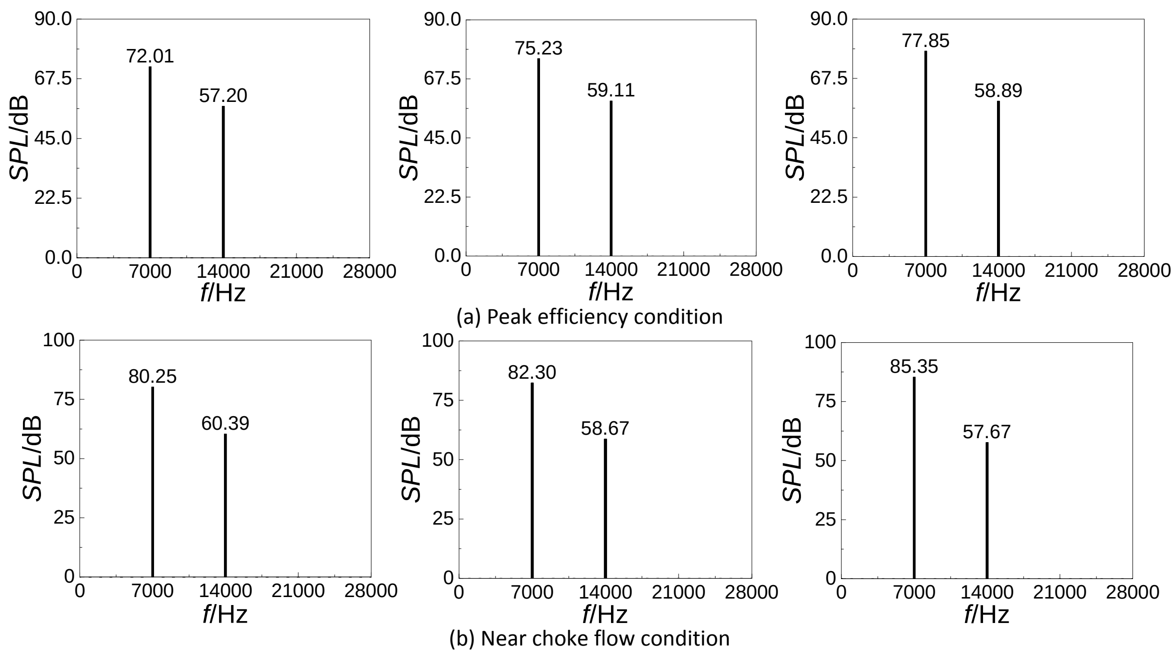
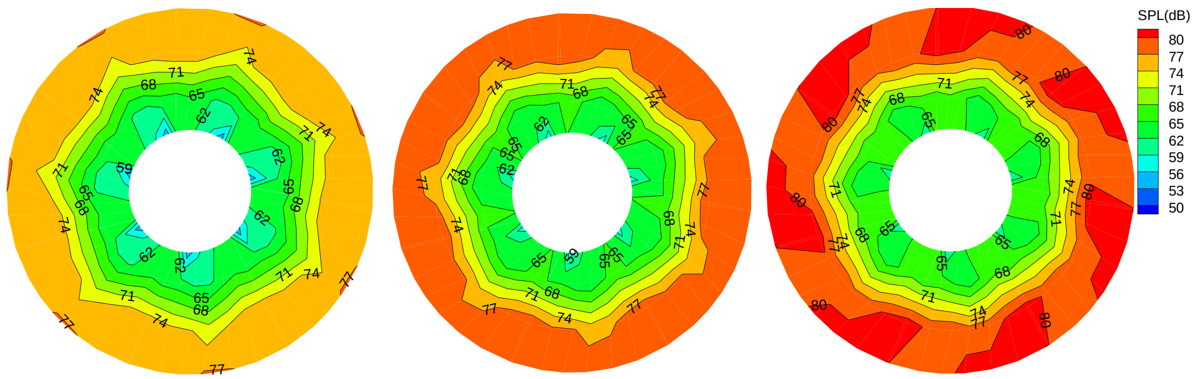
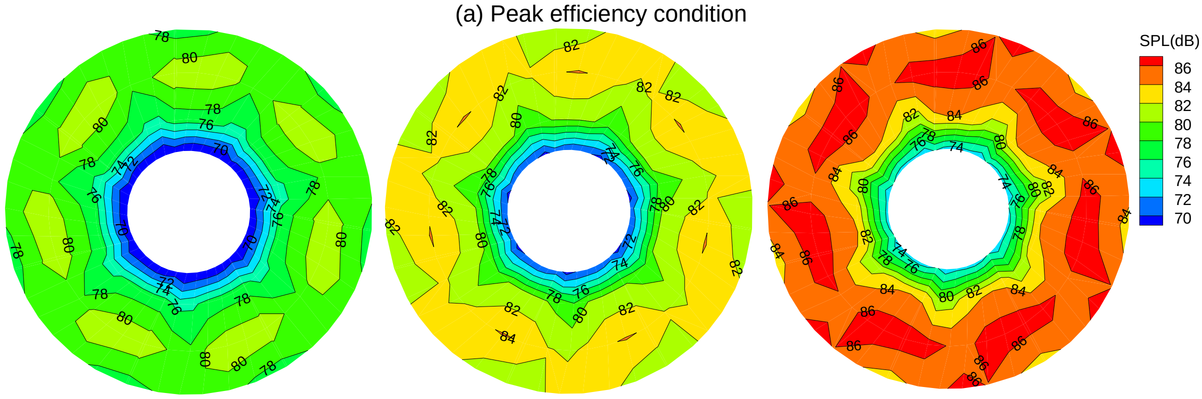


Figure 8. Spectrum characteristic of discrete aerodynamic noise

(Left: model 1 Middle: model 2 Right: model 3)



(a) Peak efficiency condition



(b) Near choke flow condition

Figure 9. Discrete aerodynamic noise SPL at centrifugal compressor inlet
(Left: model 1 Middle: model 2 Right: model 3)

Meanwhile, the SPL tends to increase along spanwise direction and gets to its maximum level in the tip region. When comparing the SPL among different dis-tuned models under same operation point, one can also find that the dis-tuned tip clearance has significant influence on the near field discrete aerodynamic noise. Under peak efficiency operation point, though similar contour distribution can be found for all the three models, the maximum value of SPL differs a lot. By taking MODEL 2 as reference, the averaged SPL value on the monitoring surface for MODEL 1 is obviously lower than MODEL 2 by difference being about 3 dB, while the SPL of MODEL 3 is higher than MODEL 2, especially in the region near the tip. Similar results can be found for the 3 models under near choke operation point: SPL of MODEL 1 is about 2 to 3 dB lower than MODEL 2 in a global view on the monitoring surface, while SPL of MODEL 3 is about 3-4 higher than MODEL 2.

3.4 Far field aerodynamic noise analysis

The far field aerodynamic noise of the centrifugal compressor with three dis-tuned tip clearance settings are measured at peak efficiency point of rotating speed of 50000 r/min, 60000 r/min and 70000 r/min. Two SPL meters locate at 1 meter away from the compressor are installed for noise measuring, as shown in figure 3. During the tests, the compressor is driven by a radial turbine, and the far field aerodynamic noise measured also includes the noise of the turbine. Since the blade number of the turbine impeller equals 10 and is identical to the blade number of the compressor, the noise source can thus be identified by frequency. Under rotating speed of 5000 r/min, 60000 r/min and 70000 r/min, the compressor passage passing frequency (PPF) equals to 5833Hz, 7000Hz and 8166Hz accordingly, while the turbine blade passing frequency (BPF) equals to 8333Hz, 10000Hz and 11666Hz. The PPF sound pressure amplitudes of the three models at 1# monitoring point is listed in table 2, in which, Δp represents the relative value upon MODEL 2. A positive value means the sound pressure amplitude of the model is higher than that of the prototype, while a negative value means the sound pressure amplitude of the model is lower than that of the prototype.

Table 2. Comparison of PPF amplitudes of three models at 1# monitoring point

n(r/min)	Model	PPF(Hz)	Sound pressure(Pa)	ΔP(Pa)
50000	1	5833	0.22	-0.02
	2	5833	0.24	0
	3	5833	0.69	0.45
60000	1	7000	0.34	-0.42
	2	7000	0.76	0
	3	7000	1.15	0.39
70000	1	8166	0.60	0
	2	8166	0.60	0
	3	8166	0.86	0.26

With the given data of sound pressure amalitude in the table, one can see that for MODEL 1, the dis-tuned tip clearance decreases the PPF sound pressure under rotating speed of 50000 r/min and 60000 r/min, comparing to that of the prototype (MODEL 2). While for MODEL 3, the dis-tuned tip clearance increases the sound pressure under all the rotating speeds.

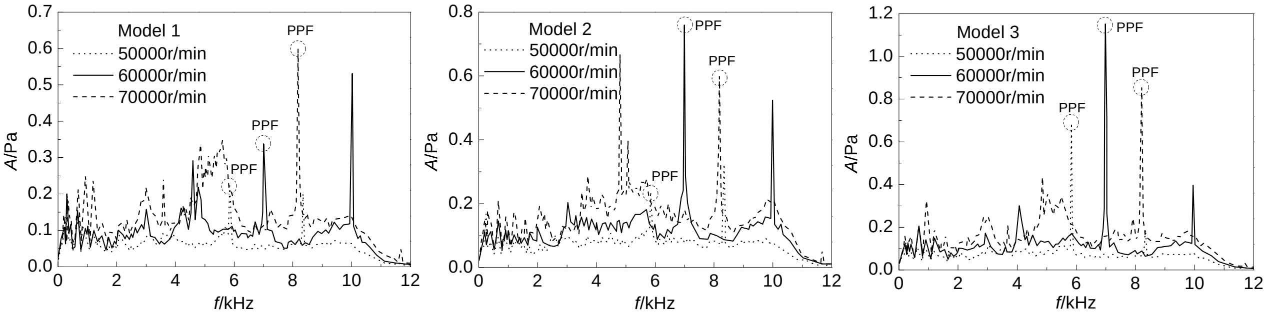


Figure 10. Noise sound pressure spectrum at 1# monitoring point
(Left: MODEL 1, Middle: MODEL 2, Right: MODEL 3)

The sound pressure spectrum at 1# monitoring point for the 3 models are presented in figure 10. From the spectrum characteristics, both broadband noise and discrete noise can be observed. The sound pressure amplitude of broadband noise for all the models at any rotating speed is much lower than that of discrete noise, and with the increase of rotating speed, the broadband noise also increases.

The discrete noise consists of compressor passage passing frequency (PPF) noise and turbine blade passing frequency (BPF) noise. Comparisons among the three models indicate that the PPF amplitude of MODEL 1 is significantly decreased at rotating speed of 60000r/min and slighly decreased at rotating speed of 50000r/min and 70000r/min, as compared to the prototype (MODEL 2). While for MODEL 3, the PPF ammplitude is always higher than the prototype. These results again confirm that the dis-tuned tip clearnance model for MODEL 1 is able reduce the discrete aerodynamic noise of the investigated compressor, while the one for MODEL 3 worsen the discrete noise level.

The measured SPL under peak efficiency point of speed line 50000r/min, 60000r/min and 70000r/min are shown in figure 11 for easy comparisons on discrete noise under PPF. With the chart, the influences of the dis-tuned tip clearance model on the discrete noise can be clearly pointed out. With the dis-tuned model of decreasing the tip clearance of main blades while increasing that of splitter blades, MODEL 1 can decrease the discrete noise, and the maximum reduction on SPL is up to 7dB at 60000 r/min. While with the dis-tuned model of increasing the tip clearance of main blades and decreaseing that of splitter blades, MODEL 3 leads to discrete noise enhancements, with the increased SPL being up to 8dB under 50000 r/min.

With above discussion, it can be concluded that the dis-tuned tip clearance control has significant influences on compressor discrete noise, either in near field noise analysis or in far field noise measurements. Similar conclusions are given by both numerical simulations and experimental measurements. However, dis-tuned tip clearance method does not necessarily reduce the discrete noise. Its influences on the discrete noise reduction depend upon the detailed dis-tuning method. In current study, the prefered dis-tuned method is to decrease the tip clearance of the main blades while increase the tip clearance of the splitter blades.

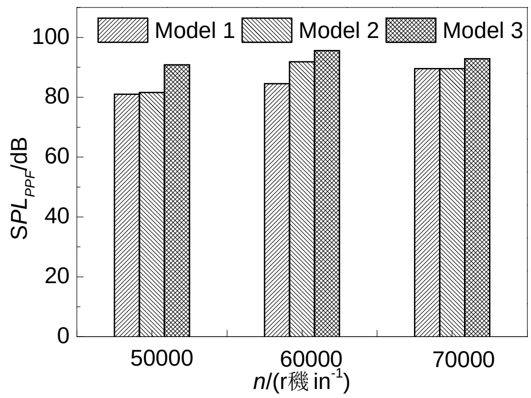


Figure 11. PPF noise comparison

CONCLUSION

In order to decrease discrete aerodynamic noise in small-size high-speed centrifugal compressor, two dis-tuned tip clearance models are developed and their influences on the performances and discrete aerodynamic noise of compressor are investigated with both numerical and experimental methods. With above discussion, following conclusions can be given on the compressor investigated in current research:

1) The dis-tuned tip clearance method utilized in this paper has minor influences on the compressor performances. With the method of increasing the main blades tip clearance size while decreasing the splitters', the compressor pressure ratio and efficiency can be kept unchanged.

2) The dis-tuned tip clearance method has influences on the compressor discrete aerodynamic noise. Both the near field and far field discrete noise can be reduced by dis-tuning the tip clearance size of main blades and splitter blades. For the compressor investigated in current research, the proposed dis-tuning method for noise reduction is to decrease the tip clearance size of main blades while increase the tip clearance size of splitter blades.

ACKNOWLEDGEMENT

The authors would like to thank for the support from National

Science Foundation of China (No. 51006011, 51276018).

REFERENCES

[1] Liu Yang, Zhang Wenzheng, Du Bingxin, et al. Numerical analysis of aerodynamic noise for turbocharger centrifugal compressor[J]. Journal of Vehicular Engine, 2013(2):31-35.
[2] Wen Huangbing, Xu Wenjiang, Bao Suning, et al. Experimental research on noise characteristics and mechanism of marine diesel engine turbocharger [J]. Chinese Internal Combustion Engine Engineering, 2013,34(1):76-80.
[3] Raitor T, Neise W. Sound generation in centrifugal compressors[J]. Journal of Sound and Vibration, 2008, 314(3): 738-756.
[4] Khelladi S, Kouidri S, Bakir F, et al. Predicting tonal noise from a high rotational speed centrifugal fan[J]. Journal of Sound and Vibration, 2008, 313(1): 113-133.
[5] Velarde-Suárez S, Ballesteros-Tajadura R, Pablo Hurtado-Cruz J, et al. Experimental determination of the tonal noise sources in a centrifugal fan[J]. Journal of sound and vibration, 2006, 295(3): 781-796.
[6] Wu Xianjun, Zhu Shijian, Li Zhiming. An optimization model by using unequal spacing blades for turbine noise reduction [J]. Applied Acoustics, 2004, 23(6):40-44.
[7] Neuhaus L, Neise W. Active control to improve the aerodynamic performance and reduce the tip clearance noise of axial turbomachines[C]//11th AIAA/CEAS Aeroacoustics Conference (26th AIAA Aeroacoustics Conference), Monterey, CA, May. 2005: 23-25.
[8] Polacsek C, Desbois-Lavergne F. Fan interaction noise reduction using a wake generator: experiments and computational aeroacoustics[J]. Journal of Sound and Vibration, 2003, 265(4): 725-743.
[9] Farassat F. Derivation of Formulations 1 and 1A of Farassat[J]. NASA TM, 2007, 214853: 2007.