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EXPERIMENTAL STUDY OF CIRCUMFERENTIAL PROPAGATION FEATURES INDUCED BY UNSTEADY TIP LEAKAGE FLOW IN AXIAL COMPRESSOR

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ABSTRACT

This paper presents an experimental study of the propagation features induced by periodic oscillation of tip leakage flow under different operating modes. The influences of rotating speed, tip clearance size, inlet condition have been explored. For uniform inlet condition, the compressor throttling process is studied at three shaft speeds, which are 1800 rpm, 2100 rpm and 2400 rpm, and at each shaft speed the effects of three tip clearance sizes are compared by changing casings, i.e. 0.7 mm, 1.1 mm and 1.7 mm. For inlet hub distortion condition, the effects of three tip clearance sizes respectively at 2100 rpm and 2400 rpm are discussed. To collect the dynamic static pressure at casing over blade tip, fifteen Kulite dynamic pressure sensors are circumferentially arranged at 20% axial chord downstream blade leading edge, which cover about thirty degrees circumferentially. Also three rows of dynamic sensors aligned with blade chord are placed, and there are five sensors in each row, which are from 20% axial chord upstream of blade leading edge to 60% axial chord downstream of blade leading edge. To test the section and level of the distortion, the radial distributions of the inlet total pressure are measured with a five hole probe. The frequency spectrums for every operating condition at different blade chord positions are analysed by Power Spectrum Density results, and the circumferential propagation characteristics of propagation speed and mode orders are studied by the time-space pressure contours and spatial Discrete Fourier Transform results. The evolution trends of these three features during compressor throttling process are summarized. And the key parameters that mainly affect the propagation features are found. The experimental results validate previous numerical results.

INTRODUCTION

Tip leakage flow (TLF) has profound effects on axial compressor performance. It has been an important issue in turbomachinery research field since the 1950s. Some experimental and numerical researches have found and confirmed the periodic oscillation of TLF in last two decades. Remarkable oscillation occurs in compressors with relative large tip clearance size at near stall point. Relevant researches mainly focused on three aspects. First is noise problem caused by unsteady TLF [1-3]. Second is blade vibration induced by unsteady TLF [4, 5]. Last is the role of unsteady TLF in stall inception and its value to monitor and control stall [6-8], this is also the main topic of this article. These three aspects are not independent, but closely related to each other. The common key parameters are the frequency spectrum characteristics of TLF unsteadiness and the circumferential propagation features of the pressure wave induced by unsteady TLF, such as propagation velocity and mode orders.

Kameier [3] experimentally studied the development of frequency and propagation velocity of TLF unsteadiness during compressor throttling process both in rotor relative and absolute stationary reference frames. They named this phenomena as rotating instability (RI). The results show that the development in both frames has different trends, and tip clearance size has obvious effect on the characteristics of TLF and its noise level. Mailach [9] and März [10] found that unsteady TLF shows a frequency lower than blade passing frequency (BPF) and induces a pressure wave propagating at about 50% rotor shaft speed (Ns). Mailach [9] also found the mode orders are about half of blade number. Young [6] set an eccentric rotor to get a variation of tip clearance size. The experiment detected the propagation of

TLF unsteadiness with large tip clearance size, which shows similarity to the results of Mailach [9] and März [10]. By changing compressor shaft speed, tip clearance size and blade number, Pardowitz [11] experimentally studied the features of RI, including frequency, mode orders and propagation velocity, and put forward a new hypothesis about RI mechanism that instability waves of different wavelengths are generated stochastically in a shear layer resulting from a backflow in the tip clearance region. In the research about unsteady TLF and its effect on nonsynchronous vibration, Drolet [12] pointed out that tip instability convection coefficient is of great importance to predict nonsynchronous vibration. Zhou [1] provided a relationship between rotor sound source and sound field parameters such as frequency and mode, and they also experimentally verified it. Fu [13] numerically studied RI in a transonic axial compressor and found the main mode occupied about 2-3 blade pitches.

The above results show that the periodic oscillation of TLF will induce a pressure wave that rotates circumferentially with particular wavelengths. The parameters of this pressure wave are determined by many factors, including compressor geometrical and aerodynamic parameters. Different types of rotating waves cause different kinds of stall inception. It is necessary to deeply and in a detailed matter study the circumferential propagation features induced by unsteady TLF.

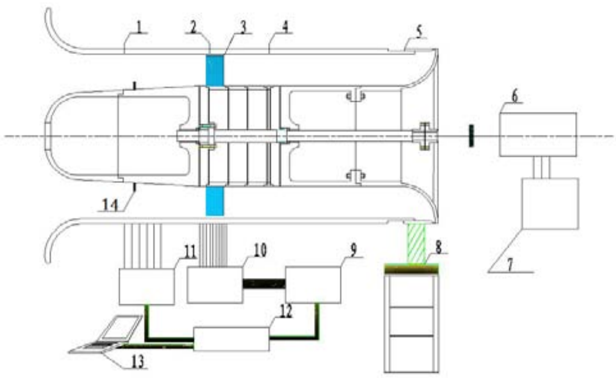
In the authors' research group, the periodic oscillation of TLF, its propagation characteristics and relation with stall inception have been performed. Deng [14] numerically studied the oscillation characteristics of TLF and the effect factors such as flow rate, tip clearance size, rotating speed and so on. And they explained the triggering mechanism of TLF unsteadiness. Furthermore, Tong [7] experimentally validated the numerical results of Deng [14] and put forward a three-stage development mode about TLF. Geng [15] numerically studied the frequency and propagation characteristics in both absolute and rotor relative reference frames, and concluded their developing trends during compressor throttling. Li [16] experimentally studied the propagation features induced by unsteady TLF and got the same conclusions as Geng [15]. It has been shown that once the periodic unsteadiness of TLF occurs in compressor, it will always propagate in absolute reference frame at the speed less than Ns, thus the autocorrelation coefficient based on pressure signals on casing will decrease. This technique can be used to predict stall inception.

This article will experimentally investigate the propagation characteristics of TLF under the effects of tip clearance size, shaft speed and inlet distortion. The development trends of propagation features during throttling process will be analysed. The results will provide the basis for the next research about the transition from unsteady oscillation of TLF to stall precursors.

EXPERIMENT SETUP AND MEASUREMENT

The experiments are conducted on a low speed single rotor axial compressor. The compressor has 60 blades, whose tip chord length is 36.3 mm and setting angle is 39.2° (angle from tangential direction). The design speed is 2400 rpm, and a DC motor is used to drive the compressor. By adjusting the DC motor, three speeds are selected, i.e. 1800 rpm, 2100 rpm and 2400 rpm. The rotor tip diameter is 497.8 mm with a hub-to-tip ratio of 0.75. By changing the compressor casing, three tip clearance sizes are obtained, i.e. 0.7 mm, 1.1 mm and 1.7 mm. A throttle driven by a stepping motor is set at the compressor exit to adjust the flow rate. This will keep the throttling speed and the flow field stable to ensure repeatability. A metal ring piece is placed in front of the rotor to generate inlet distortion. The height of the ring is 8 mm and the thickness is 3 mm. By changing the diameter of the ring, the inlet hub, mid and tip distortion are generated. The schematic of the compressor test rig is shown in Fig.1.

The overall performance map of the compressor is measured by the inlet and outlet static pressure taps on the casing, as shown in Fig.1. To collect the dynamic pressure signals, a series of Kulite XT-140 dynamic pressure sensors are mounted on the casing over the blade tip. The pressure range is 35 kPa. The detailed arrangement of these sensors is depicted in Fig.2, including three rows in blade chord direction from 20% axial chord upstream of blade leading edge to 60% axial chord downstream of blade leading edge and one circumferential row at 20% axial chord downstream of blade leading edge. For each row, the sensors have equidistant spacing. To make the measurement has enough space resolution and also limited by the size of dynamic pressure sensor, there are five sensors in each blade chord row and four circumferential sensors in each blade pitch direction. The sampling frequency of the sensors is 100 kHz, it is much higher than BPF.



1. Inlet static pressure 2. Dynamic static pressure 3. Rotor
4. Outlet static pressure 5. Throttle 6. Motor
7. Frequency converter 8. Linear unite 9. Tape recorder
10. Amplifier 11. Pressure transducer
12. PXI acquisition module 13. Computer 14. Inlet distortion

Fig.1 Schematic of compressor test rig and measurement system

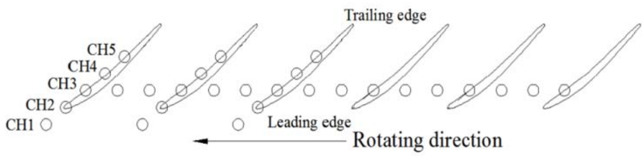


Fig.2 Arrangement of dynamic pressure sensors

OPERATING CONDITION AND TEST PROCEDURE

The experiments are conducted at different rotation speeds and with different tip clearance sizes and inlet conditions. Table 1 gives the summary of the combinations of these operating conditions. When with the inlet mid and tip distortions, the compressor will go directly into stall once it is started. So only the effects of inlet hub distortion are studied in this paper, and two larger rotation speeds are selected.

Table 1 Summary of operating conditions

		Rotation speed/rpm (uniform inlet)			Rotation speed/rpm (inlet hub distortion)	
		1800	2100	2400	2100	2400
Tip clearance size/mm	0.7	√	√	√	√	√
	1.1	√	√	√	√	√
	1.7	√	√	√	√	√

The test procedure is as following. Step (1): Select one tip clearance size. For one rotation speed, first check the repeatability of the compressor performance, then collect the dynamic pressure signals at series of operating points during throttling process. Step (2): Change the rotation speed, repeat step (1). Step (3): Install inlet distortion ring, repeat steps (1) and (2). Step (4): Change the compressor casing, repeat steps (1), (2) and (3).

Taking the combination of 2400 rpm and 1.1mm tip clearance size as example, the compressor characteristic lines are shown in Fig.3. It is obvious that the four results are very close and show good repeatability.

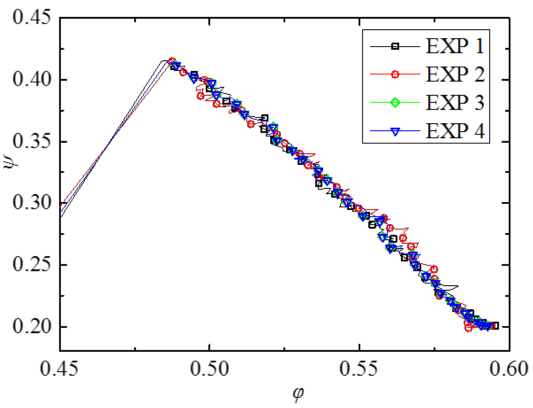


Fig.3 Compressor characteristic lines (2400 rpm, 1.1 mm)

ANALYSIS METHOD

The circumferential propagation features induced by unsteady TLF are studied from three aspects. One is the PSD results based on FFT, which is used to analyse the frequency features of unsteady TLF, and the pressure signals along blade chord are mainly studied. The other two are the propagation speed and mode orders. First, the original dynamic pressure signals along circumferential direction are extracted. Second, the band-pass filter based on PSD result is performed for these signals. Third, the time-space pressure contours are plotted. By calculating the slope of the pressure isoline, the propagation speed can be got. For each time instant, the mode orders are obtained by conducting the spatial DFT.

The combination of 2400 rpm and 1.1mm tip clearance size still is taken as example. Fig.4 shows the PSD results at three typical flow points. With compressor throttling, the unsteadiness of TLF appears and becomes more obvious. At the same time, the position of dominant unsteadiness with largest amplitude moves upstream, which is about 20% axial chord at near stall point. So in the following sections, the results at 20% axial chord are mainly focused on. Fig.5 further depicts the time-space pressure contours at 20% axial chord for two typical flow points. At larger flow point of φ =0.57, there is almost no TLF unsteadiness, and the propagation speed is 1.0 Ns. At near stall point of φ=0.49, there is obvious unsteadiness of TLF, and the propagation speed is 0.46 Ns. Fig.6 is the spatial DFT results at near stall point of φ=0.49, the mode orders are about 50.

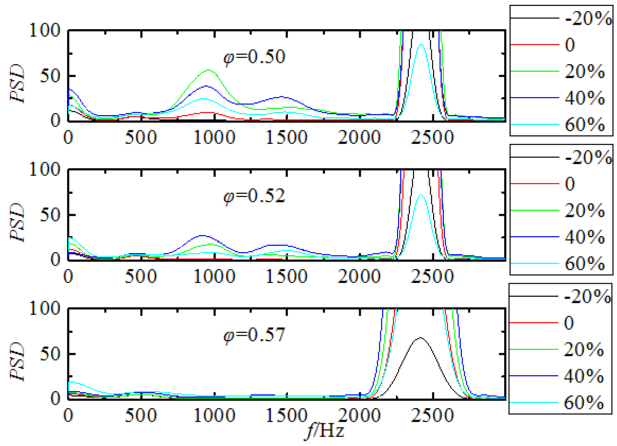


Fig.4 PSD results (2400 rpm, 1.1 mm)

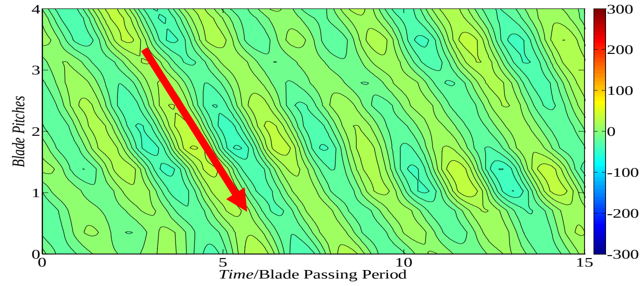
RESULTS AND DISCUSSION

In the following three sections, the evolution trends of dominant frequency, propagation speed and mode orders during throttling process are analysed. In section one and two, the effects of tip clearance size and rotation speed are analysed, and the experiments are conducted with uniform inlet condition. In section three, the effects of inlet distortion are discussed.

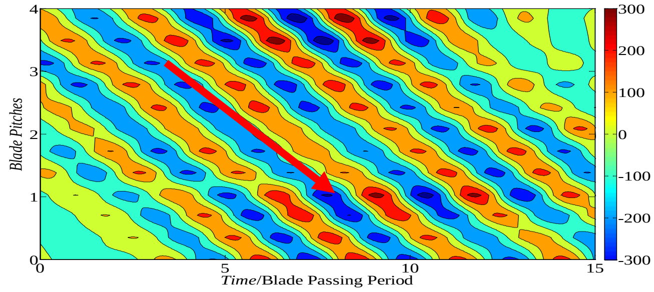
Effects of tip clearance size

The effects of three tip clearance sizes at three rotation speeds are compared. The compressor performance map at 2400 rpm with three tip clearance sizes are shown in Fig.7. With the increase of tip clearance size, the stall limit moves to larger mass flow rate, and pressure rise coefficient at the same flow point decreases. The effects of tip clearance sizes on compressor performance at other two rotation speeds are similar.

The evolutions of dominant frequency component of largest amplitude with three tip clearance sizes at three compressor rotation speeds are summarized in Fig.8. For each rotation speed, when the tip clearance sizes decrease, the unsteadiness of TLF appears at much smaller mass flow rate, and the frequency value at the same flow point increases. The overall trend of dominant frequency during compressor throttling process first increases and then decreases. There is a critical mass flow rate, at which the frequency is largest. This critical flow point moves to smaller mass flow rate with the decrease of tip clearance size. The slopes on both sides become more flat. These results are the same as those by numerical simulation [15] shown in Fig.8.



(a) φ =0.57



(b) φ =0.49

Fig.5 Time-space pressure contours (2400 rpm, 1.1 mm)

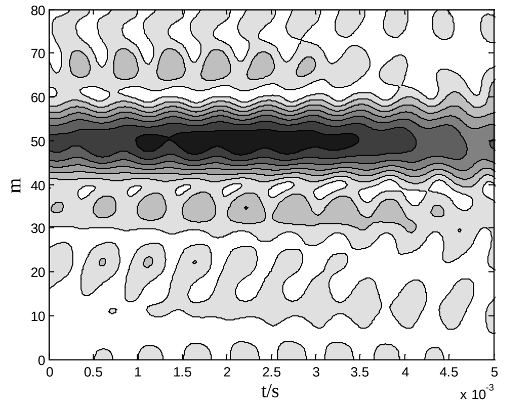


Fig.6 Mode orders at φ =0.49 (2400 rpm, 1.1 mm)

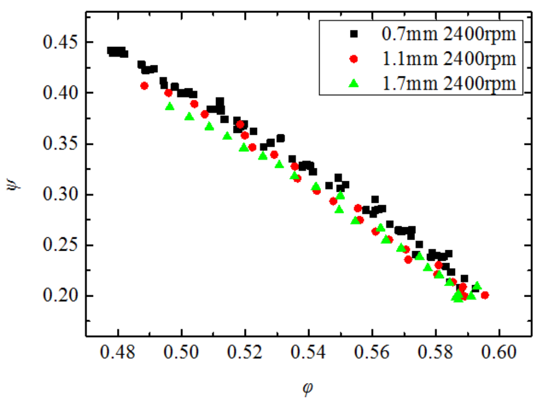
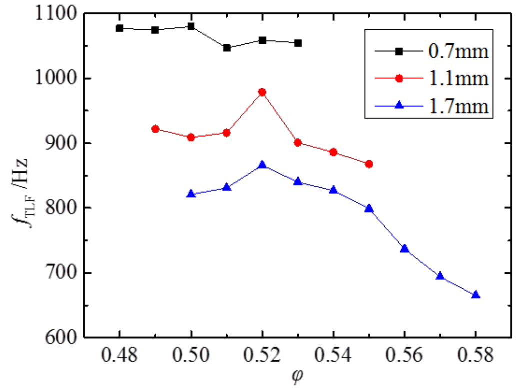
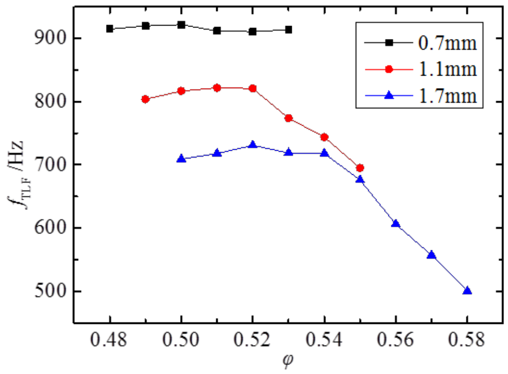


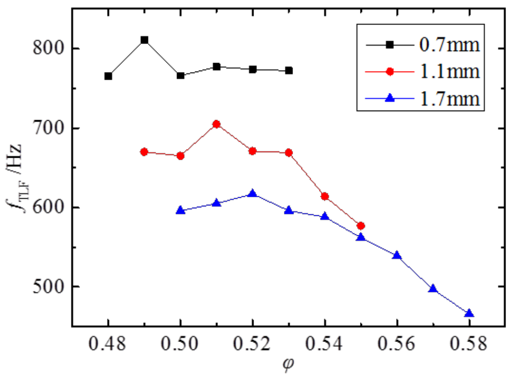
Fig.7 Compressor performance map at 2400 rpm



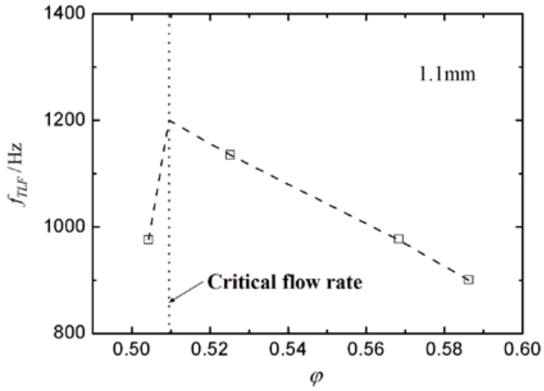
(a) 2400 rpm



(b) 2100 rpm



(c) 1800 rpm

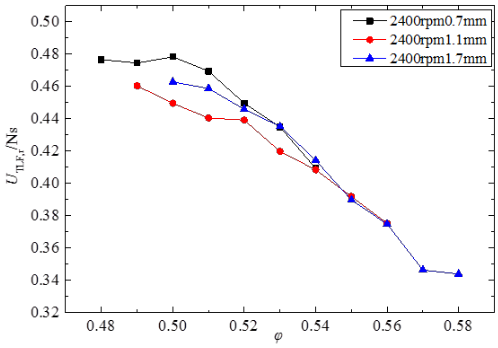


(d) numerical results at 2400 rpm [15]

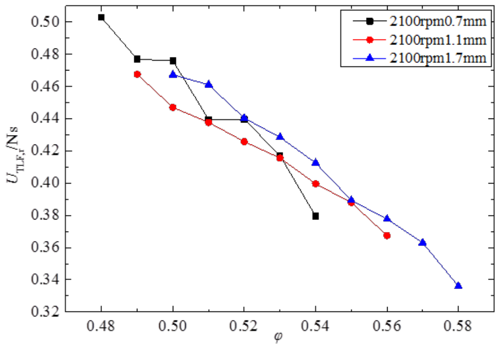
Fig.8 Evolutions of dominant frequency

Fig.9 shows the evolution trends of propagation speed of pressure wave induced by unsteady TLF. For each rotation speed and tip clearance size, the propagation speed increases with compressor throttling. For the same compressor rotation speed, there is no much difference between different tip clearance sizes at the same mass flow point. Fig.10 further shows the evolution trends of mode orders. For each compressor rotation speed, when the tip clearance sizes are relatively small, such as 0.7 mm and 1.1 mm, the mode orders at the beginning of unsteady TLF is the same as compressor blade number, then decrease with compressor throttling. When the tip clearance size is relatively large enough, the mode orders at the beginning of TLF are less than blade number and decrease with compressor throttling. At the same mass flow point, the mode orders increase with the decrease of tip clearance size. These results also agree well with the numerical ones by Geng [15].

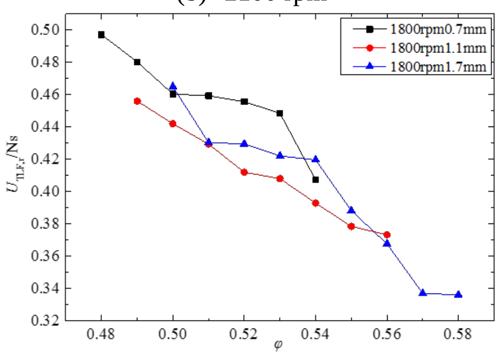
It is shown that for the low-speed axial compressor in this paper tip clearance size has much more effects on frequency and mode orders than propagation speeds.



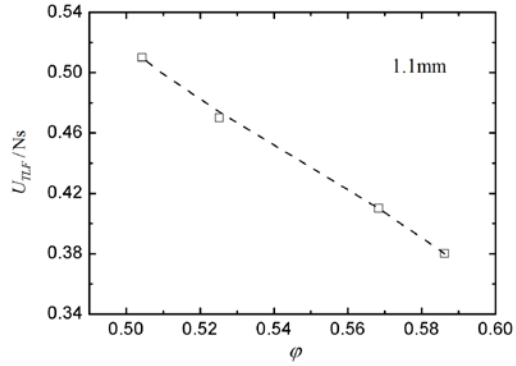
(a) 2400 rpm



(b) 2100 rpm

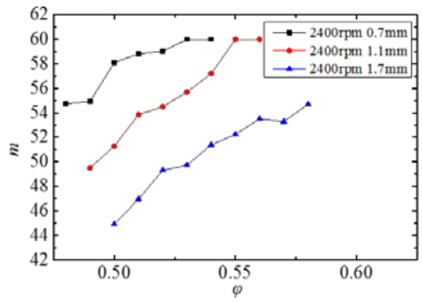


(c) 1800 rpm

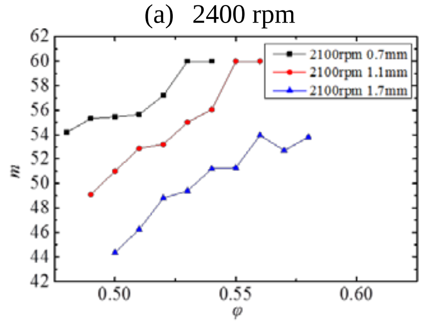


(d) numerical results at 2400 rpm [15]

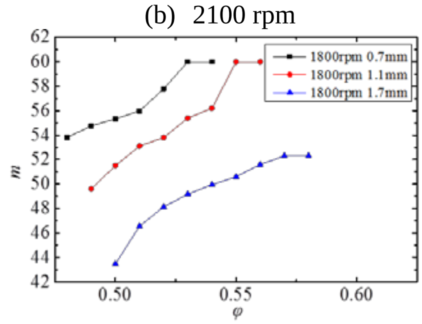
Fig.9 Evolutions of propagation speed



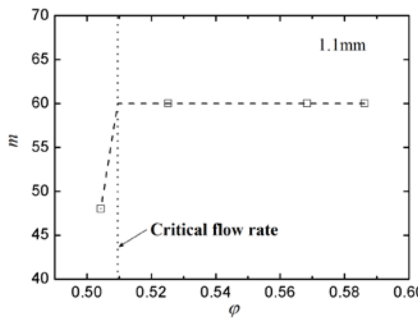
(a) 2400 rpm



(b) 2100 rpm



(c) 1800 rpm



(d) numerical results at 2400 rpm [15]

Fig.10 Evolutions of mode orders

Effects of rotation speed

Fig.11 is the compressor performance map at three rotation speeds with the same tip clearance size of 1.7 mm. There is little difference between the three curve lines, including pressure rise coefficient and stall point.

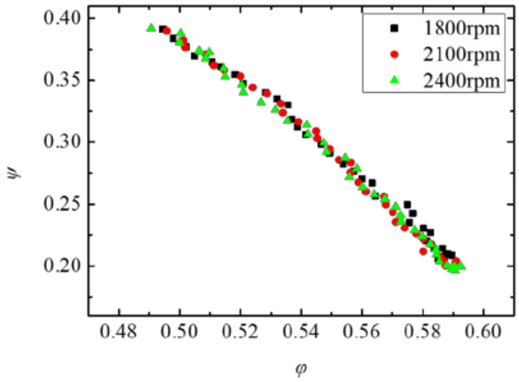


Fig.11 Compressor performance map with 1.7 mm tip clearance size

For each tip clearance size, the same flow point near stability limit is selected to compare the dominant frequency of TLF. They are φ=0.48 for 0.7 mm tip clearance size, φ=0.49 for 1.1mm tip clearance size, and φ=0.50 for 1.7 mm tip clearance size. The results are given in Table 2. For the same tip clearance size, both the absolute frequency and dimensionless frequency by BPF increase with the increase of compressor rotation speed. For the same rotation speed, both the absolute frequency and dimensionless frequency by BPF increase with the decrease of tip clearance size. This result is the same as in section one.

To make the comparison, the data in previous figure are re-arranged. The data in Fig.12 are the same as those in Fig.9, but they are plotted together in one figure. There is no much difference between different rotation speeds and tip clearance sizes at the same mass flow point. This difference may be within the precision to calculate the propagation speed. For Fig.13, the data are the same as in Fig.10. The mode orders are mainly affected by tip clearance sizes, and the compressor rotation speed has little influence on them for this low speed single rotor compressor.

Table 2 Dominant frequency of TLF at near stall point

Tip clearance size/mm	Rotation speed/rpm	f_{TLF}/Hz	BPF/Hz	f_{TLF}/BPF
1.7	1800	590	1800	0.328
1.7	2100	695	2100	0.331
1.7	2400	820	2400	0.342
1.1	1800	655	1800	0.364
1.1	2100	810	2100	0.386
1.1	2400	1045	2400	0.435
0.7	1800	805	1800	0.447
0.7	2100	960	2100	0.457
0.7	2400	1100	2400	0.458

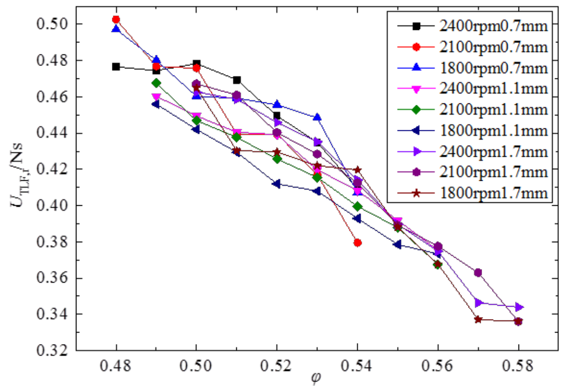


Fig.12 Evolutions of propagation speed

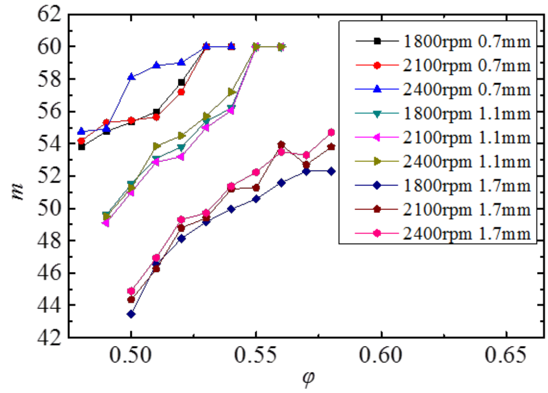


Fig.13 Evolutions of mode orders

Effects of inlet hub distortion

To test the level of distortion at compressor inlet, the five hole probe is adopted to measure the radial distribution of inlet total pressure. The radial distribution of total pressure is shown in Fig.14. Fig.15 is the compressor performance maps with three tip clearance sizes at 2400 rpm. Large tip clearance size will deteriorate compressor performance, and make it stall at larger mass flow rate.

Fig.16 is the evolutions of TLF dominant frequency. They show similar trends like those with uniform inlet condition. At each rotation speed, this frequency first increases with compressor throttled, when reaching a critical mass flow rate it begins to decrease with compressor further throttled. This critical mass flow rate depends on the compressor operating conditions. Fig.17 and Fig.18 respectively give the evolution trends of propagation speed and mode orders. They show similar change trends as in previous sections. The dimensionless propagation speed by compressor rotation speed increases with compressor throttled for each operating conditions. There is no apparent difference between these cases. For the mode orders, they decrease with compressor throttled, and mainly depend on tip clearance sizes.

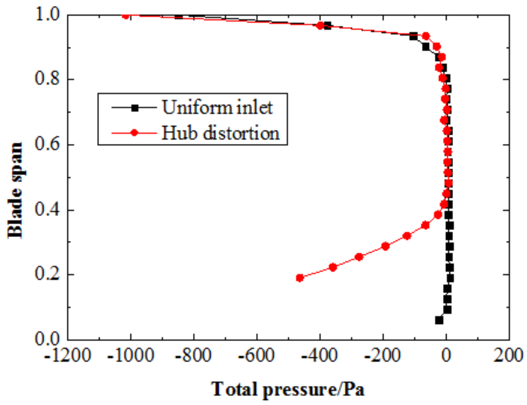


Fig.14 Radial profile of inlet total pressure

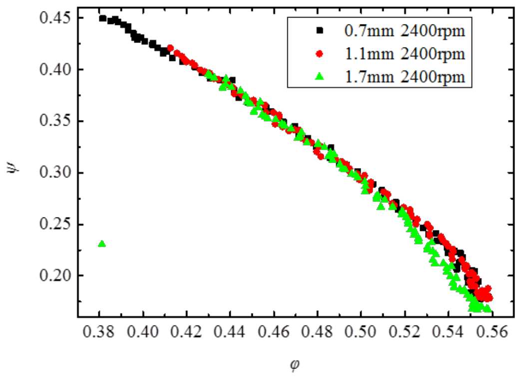
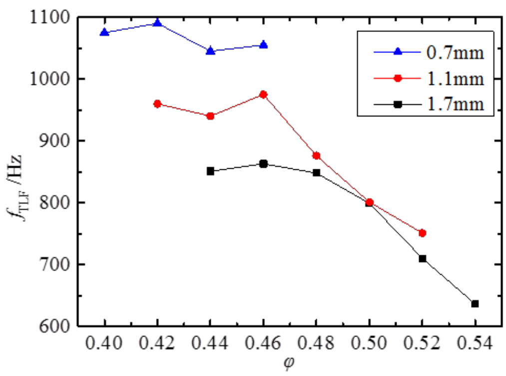
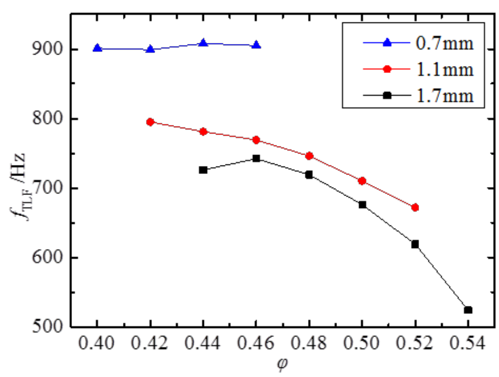


Fig.15 Compressor performance map at 2400 rpm



(a) 2400 rpm



(b) 2100 rpm

Fig.16 Evolutions of dominant frequency with inlet hub distortion

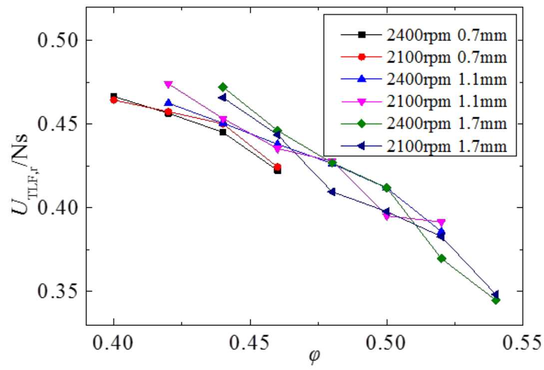


Fig.17 Evolutions of propagation speed

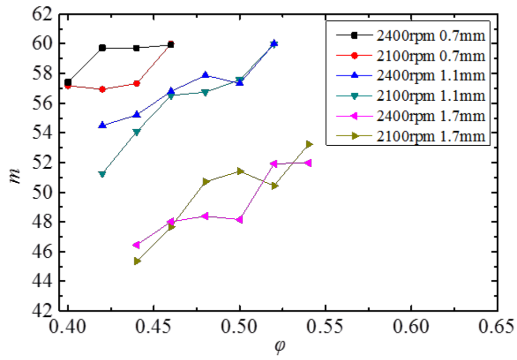


Fig.18 Evolutions of mode orders

CONCLUSIONS

By collecting and analysing the dynamic pressure signals on compressor casing over blade tip, the propagation characteristics of pressure wave induced by TLF unsteadiness are experimentally investigated. The effects of tip clearance size, rotation speed and inlet hub distortion on frequency, propagation speed and mode orders are compared.

With compressor throttling, the dominant frequency first increases and then decreases. There is a critical flow point at which the frequency value is largest. This critical flow point is determined by operating conditions. With compressor throttling, the propagation speed increases. For both uniform and hub distortion inlet conditions, there is no much difference for the evolution trends between different tip clearance size and rotation speed. With compressor throttling, the mode orders decreases. Tip clearance size has much more effects on frequency and mode orders than propagation speed.

The above results validate the results of previous CFD simulations in [15], and provide the base for the research about relation between unsteady TLF and different types of stall precursors. This will be further studied in the next research.

NOMENCLATURE

V_{axial}	average axial velocity, [m/s]
$P_{t,\,in}$	inlet stagnation pressure, [Pa]
U	tangential velocity at blade mid span, [m/s]
ρ	density, [kg/m^3]
P	static pressure, [Pa]

φ	mass flow coefficient, V_{axial}/U
ψ	pressure rise coefficient, $(P_{out}-P_{t,in}) / (0.5\rho U^2)$
TLF	tip leakage flow
RI	rotating instability
BPF	blade passing frequency
PSD	Power Spectral Density
FFT	Fast Fourier Transform
DFT	Discrete Fourier Transform
Ns	rotor shaft rotation speed

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