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A STUDY ON THE INFLUENCE OF HOLE'S DIAMETER WITH HOLED CASING TREATMENT

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ABSTRACT

The holed casing treatment is a new type of casing treatment with self-adaptability for centrifugal compressor with unshrouded impellers. It is demonstrated experimentally and numerically that both of the stall margin and the choked margin of the compressor can be expanded by the treatment. Numerical results indicate that there is a reinjected flow in the holes when the compressor works at low flowrate conditions and a bypass flow at high flowrate conditions. Hole's diameter is an important parameter for the holed casing treatment. Five cases of different diameter (1.0mm, 1.5mm, 2.0mm, 2.5mm and 3.0mm) were carried out to reveal its influence. Both the stall margin and efficiency are improving with increasing of the hole's diameter in the cases of diameter below than 2.5mm. At diameter of 2.5mm, the stall margin increment and the efficiency of the compressor are the highest among all 5 cases. However, in the case of 3.0mm, the stable working range enhancing as well as the efficiency is weakened because the velocity magnitude of the reinjected flow decreases. Therefore a key principle of choosing the diameter and the radial position of the hole is presented in the paper to get the best self-adaptive effect: enhancing stable running range as much as possible and keeping higher efficiency.

NOMENCLATURE

b	Impeller width: mm
D	impeller diameter: mm
Z	number of impeller blades
β	blade angle (from the tangential direction)
N	rotating speed: rpm
M	mass flow rate: kg/s
Q	volume flow rate: m^3/s
η	efficiency
ε	pressure ratio
d	diameter of bleeding recirculation hole: mm
H	energy head: J/kg
ψ	coefficient of head
u	circular velocity: m/s
p	static pressure: Pa
T	temperature: K
v	absolute velocity: m/s
l	length: m
λ	roughness coefficient
c	specific heat

SUBSCRIPTS

1	inlet position of impeller
2	outlet position of impeller
design	designed flow rate point
stall	predicted stall point
choke	predicted choke point
hole	related to the hole
t	stagnation point
is	adiabatic status
p	constant pressure status
original	compressor without holed casing treatment
treatment	compressor with holed casing treatment

1 INTRODUCTION

Slots and grooves in the shroud over the tip of the blade, known as casing treatment, can substantially improve the stable flow range of the compressor. The concept of casing treatment was first put forward by Hartmann et al. [1] and they found in their experiments that simpler casing treatments, without bleeding flow, might provide appreciable improvements in the stall range. Many researchers have been studying the casing treatment on the axial compressor since the early 1970s. Except for basic slot casing treatment and groove casing treatment [2-7], various other forms of casing treatments have been developed over the last 40 years [8-14].

For centrifugal compressors, Jansen et al. [15] investigated the performance of stall inhibitor concepts first. Three impeller wall treatments and a vaned diffuser treatment were assessed in their study. After that, a few other studies, which focused on casing treatment for centrifugal compressors, have been reported [16-19]. As for turbocharger compressors, a ring-groove structure is used to enhance the stable range [20-21]. It was suggested by Fisher [22] and was also called ported shroud housings as shown in Fig. 1.

Different from ported shroud housings, a new form of casing treatment, which is called holed casing treatment, is proposed in this paper for unshrouded centrifugal compressor. The sketch is depicted in Fig. 2. It is consisted of a set of bleeding recirculation passages, including bleeding recirculation holes and a channel chamber. The bleeding recirculation holes are located in the stationary shroud circumferentially with endpoints called bleed ports. The other endpoint of the bleeding recirculation passage is connected with the channel chamber which has an outlet called reinjecting port.

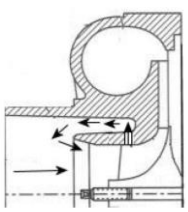


Fig. 1 Sketch of a ported shroud housing[22]

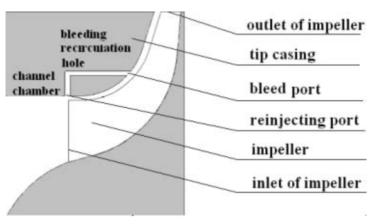


Fig. 2 Sketch of holed casing treatment

In the holed shroud housings, the flow direction in tip casing is radial, and the reinjecting port is far from the impeller. It changes the incoming flowrate directly. In reference [20], the stall margin was expanded by 12%, but the impeller efficiency was reduced by 4% at maximum flowrate. In holed casing treatment, the bleeding flow is in axial direction in the tip casing and the reinjecting port is located at the inlet of impeller. The flow field at the tip of impeller is disturbed by the reinjected flows, accompanied with the incoming flowrate change. The flow resistance in the tip casing path can be utilized to balance the static pressure difference between the bleed port and reinjecting one. At low flowrate conditions, the static pressure near the bleed port is high enough to drive a reinjected flow from the bleed port to the reinjecting one. As the compressor flowrate increases, the pressure difference and the reinjected flow amount will decrease. The flow direction will even reverse and the reinjected flow will be replaced by the bypass flow if the compressor flowrate is high enough. Thus the holed casing treatment can be a self-adaptive casing treatment. The casing treatment effect changes with the direction and flowrate in the hole. It is mainly decided by the pressure difference between the bleed port and the reinjecting port on different operating conditions. Such self-adaptive feature isn't found in references [18, 19]. In reference [20, 21], both of the positive and inverse mass flow rate in the treatment configuration appeared but the impeller efficiency at large flow rate condition was reduced. Experiments revealed that holed casing treatment expanded the stall margin and the choked margin of the compressor effectively without efficiency penalty as presented in our former work [23].

For a certain centrifugal compressor, the effect of holed casing treatment is influenced by several factors in the studies, such as the positions of the bleed port and the reinjecting port, the number and diameter of the holes, the roughness in the holes, the width of the channel chamber, the material of the casing and so on. Among the above mentioned factors, the diameter of the holes is very important. If the diameter is too small, the flow amount in the hole would be so small that the casing treatment effect may be inconspicuous. If the diameter is too big, the flow amount in the hole would be so large that the compressor efficiency may be reduced a lot.

In this paper, the performances of an unshrouded centrifugal compressor with original casing and holed casing are studied numerically and experimentally. The influence of hole's diameter is mainly studied numerically by varying the diameter.

2 MAJOR ROLE OF HOLE DIAMETER IN EFFECT OF HOLED CASING TREATMENT

As discussed in the INTRODUCTION, the self-adaptive effect of holed casing treatment depends on the direction and amount of the flow in the hole, which are mainly decided by the static pressure difference between the bleed port and the reinjecting port. The pressure difference varies with the performance of the compressor, the geometric position and shape of the hole. For a given compressor, only the radial location of the holes, the diameter and the number of holes, can influence the casing treatment effect. To pursue better self-adaptive effect, the radial location of the holes must be decided cautiously. Appropriate location of the hole would ensure that there is enough driving force to actuate the flow in the holes and improve the flow in the impeller passage effectively. Once the radial location of the holes is determined, the diameter and number of the holes become the determinants. Generally, the holes are located uniformly and the number is decided by the structure, the processing technique and the casing strength. For the small size impeller studied in this paper, 3 or 4 holes in one impeller passage are appropriate. In our study, 4 holes are selected. The hole diameter becomes very important for the holed casing treatment effect once the radial position, the number of the holes, and the roughness of the holes are determined.

The flow in a hole is treated as mean flow here. The influences of the hole parameters, such as radial position, diameter, roughness of the wall, are listed as follows.

$$H_{hole} = \psi \cdot (u_2 \cdot \frac{D_{hole}}{D_2})^2 \quad (1)$$

$$\varepsilon_{hole} = (\frac{H_{hole}}{c_p \cdot T_1} + 1)^{\frac{\kappa}{\kappa-1}} \quad (2)$$

$$p_{hole} = p_1 \cdot \varepsilon_{hole} \quad (3)$$

$$\Delta p = p_1 \cdot (\varepsilon_{hole} - 1) = f(l_{hole}, d_{hole}, v_{hole}, \rho) \quad (4)$$

$$Q_{hole} = \frac{\pi}{4} d_{hole}^2 \cdot v_{hole} \quad (5)$$

The above formulas are helpful especially during the design process. When the expected stall margin increment was determined, the maximum total reinjected flow will be determined. Then the maximum static pressure difference between the bleed port and the reinjecting port can be gotten by empirical rule summarized from the experiments and computation. Using the formulas proposed above, the static pressure difference between the inlet and different radial position can be obtained. If the static pressure difference provided can satisfy the condition, the radial position is considered as proper.

The main parameters of the impeller studied in this paper are given in Tab. 1.

Tab. 1 Parameters of the impeller

Item	Value
b_1/D_2	0.166
b_2/D_2	0.06
Z	16
β_1	30
β_2	90
N_{design}	22790
M_{design}	0.64

The different hole diameters studied in this paper are shown in Tab. 2.

Tab. 2 Different hole diameters

Case	d_{hole}(mm)
1	1.0
2	1.5
3	2.0
4	2.5
5	3.0

The geometric model of the unshrouded centrifugal compressor with holed casing treatment studied in this paper is shown in Fig. 3. The reinjecting port is located at the inlet of impeller. The holes are connected with the channel chamber and the flow direction is axial in the holes. In this study, the radial position of the holes is D_{hole}/D_2=0.686 while the number of holes is 64 and the two parameters are kept the same while the hole diameter is changed.

Fig. 3 Compressor with holed casing treatment

The flow fields are simulated in the compressors with holed casing treatment with varying diameters and the results are compared with the experiments. Analysis on the influence of the hole diameter is presented based on the results.

3 GRIDS AND NUMERICAL SIMULATION METHOD

The commercial software package NUMECA/FINE Turbo v. 8.4-1 was used to simulate the flow fields in a single impeller passage with and without holed casing treatment. The geometric model was extended to 1.68D_2 from the export of the impeller and the casing was added. The structured grids of the flow passages were generated by AUTO GRID and IGG and the multi-block patched grid technique was used. H-type grids were adopted for the main flow passage. The butterfly topology was adopted to generate the mesh in each hole. The grids are shown in Fig. 4. The number of the grid nodes in a single impeller passage for compressor with holed casing treatment is over 517,000 and the one without holed casing treatment is over 320,000. The generated grid qualities are as follows: the minimum orthogonality is 15°, the maximum aspect ratio is 1,025 and the maximum expansion ratio is 6.97. The tip clearance of the compressor studied in this paper is 0.8mm. In the model with casing treatment, the tip clearance was divided into two parts: one near the impeller tip, generated by the AUTO GRID module of NUMECA, with 12 layered grids while the other part, near the gap, by the IGG module of NUMECA, with 8 layered grids. The full non-matching connection was between the two parts of the tip clearance, and the information is transferred by a "frozen-rotor interface". 8 layers of grids are generated in the tip clearance for compressor without casing treatment. The grids near wall were refined to keep the wall y+ values lower than 10 in the flow regions after the calculation is converged.

The 3-D Reynolds averaged Navier-Stokes equations are integrated in time by a fully implicit formulation of the second-order scheme for the real gas in conjunction with the Spalart-Allmaras turbulence model. At the inlet the total pressure is 101325 Pa and the total temperature 24.5℃, the averaged static pressure is given as the outlet condition. The solid wall was set to be adiabatic and non-slippery. When the residual level decreases to a given level, the mass flowrate and pressure ratio are stable with each iteration and the relative error between inlet and outlet mass flowrate is less than 0.5%, then the calculation is regarded as converged. The numerical stall is achieved by the calculation diverges during the mass flow rate reduction. The predicted stall point is set as the last stable condition prior to the point of divergence. When the calculation condition approached the numerical stall point, the static pressure of the outlet varied in a small step of 50Pa.

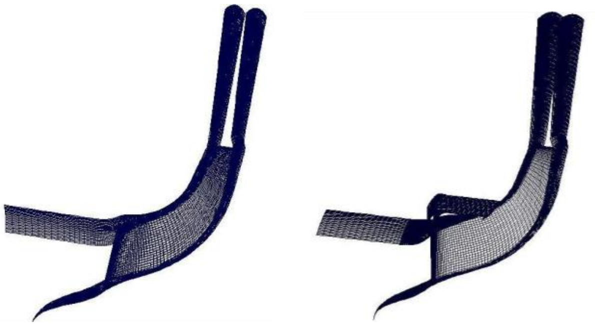
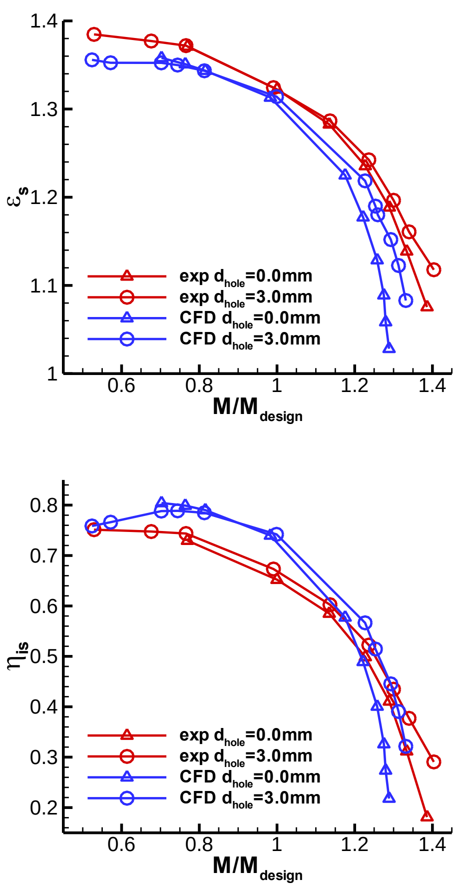


Fig. 4 Grids of single impeller passage

4 NUMERICAL RESULTS AND ANALYSIS



(b) Adiabatic efficiency

Fig. 5 Comparison of performance curves between simulations and experiments

The flow fields in the impeller passage with original casing and holed casing of different diameters were simulated separately. For a given rotating speed, the performance curve between stall and choke points was established by increasing the back pressure (outlet pressure) gradually.The performance curves for original casing (d_{hole}=0.0mm) and 3.0mm holed casing treatment (d_{hole}=3.0mm) gotten from numerical computations were compared with the experiment results as shown in Fig. 5. The relative value of the compressor inlet mass flow divided by M_{design} is used as abscissa and the ordinate ε_s is the static pressure ratio of the inlet and outlet while the ordinate η_{is} defined as

$$\eta_{is} = \frac{H_{t,2}^{is} - H_{t,1}}{H_{t,2} - H_{t,1}} \tag{6}$$

Where, H^{is}_t is adiabatic total enthalpy.

In the experiments, the holed casing treatment was implemented in the first stage of a two-stage centrifugal compressor with vaneless diffuser which is driven by an AC motor and a gear box, as shown in Fig. 6. The mass flow was measured by a flowmeter and the measurement uncertainty is 1.0%. The pressure and the temperature of the inlet and outlet were attained by the pressure sensors, the U-tubes and the temperature sensors.

(a) Two-stage compressor **(b) Holed casing**

Fig. 6 Experimental facility

The tendency of numerical results coincides with the experimental ones in terms of efficiency and pressure ratio. The experimental and numerical results both indicate that the holed casing treatment can expand the stable working range of the compressor without much efficiency loss. The differences between the experimental and numerical results are mainly due to the fact that there is a vaneless diffuser at the downstream of the impeller in the experiment while no such diffuser in the simulation.

The static pressure ratio curves and the adiabatic efficiency curves of compressors with holes of varying diameters in the casing are given in Fig. 7. The performance curves of compressor with original casing (d_{hole}=0.0mm) are also included in Fig. 7.

Stall margin (SM) is defined as

$$SM = (M_{design} - M_{stall})/M_{design} \tag{7}$$

Choked margin (CM) is defined as

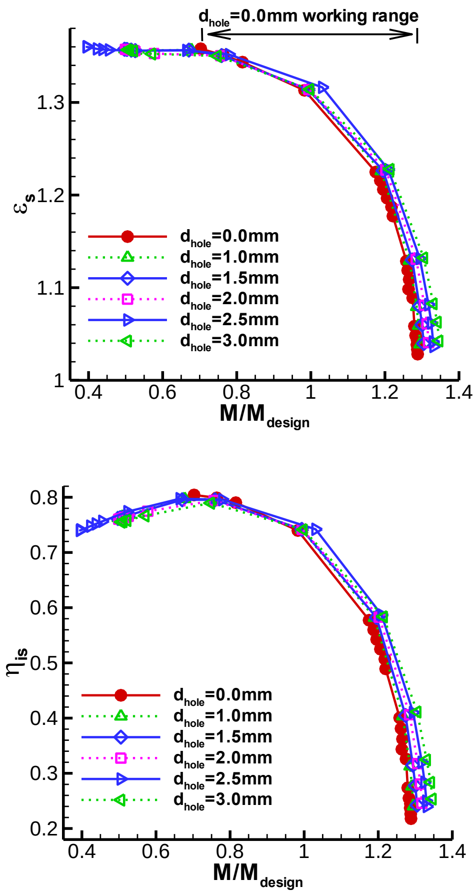
$$CM = (M_{choke} - M_{design})/M_{design} \tag{8}$$

Stall margin increment (SMI) and choked margin increment (CMI) are then defined as

$$\mathrm{SMI} = |M_{\text{stall, treatment}} - M_{\text{stall, original}}| / M_{\text{design}} \quad (9)$$

$$\mathrm{CMI} = |M_{\text{choke, treatment}} - M_{\text{choke, original}}| / M_{\text{design}} \quad (10)$$

It is shown in Fig. 7 that the stall margin(SM) and the choked margin(CM) of the compressor are both extended in all 5 cases. Besides, the pressure ratio and the efficiency change with hole diameter.



(b) Adiabatic efficiency

Fig. 7 Performance curves of different cases

If the radial position and the number of holes are given, the amount of the flow in holes would only change with hole diameter. SMI is greatly dependant by the hole diameter. With the visible SMI, the performance of the compressor changes slightly. The holed casing treatment would not influence the aerodynamic performance apparently.

4.1 INFLUENCE ON COMPRESSOR PERFORMANCE

It is shown in Fig. 7 that the static pressure ratio and the efficiency of holed casing compressors are higher than that of the original compressor when the value of relative compressor inlet mass flow M/M_{design} is larger than 1. The performance improvement increases with M/M_{design}. When M/M_{design} is close to the predicted stall point of the original casing compressor, the performance of the compressor with holed casing treatment decreases slightly and the performance drop changes with hole diameter. The maximum static pressure ratio and the top efficiency of the 6 cases are listed in Tab. 3.

Tab. 3 Maximum static pressure ratio and efficiency

case	$\varepsilon_{s,max}$	$\Delta\ \varepsilon_{s,max}$	$\eta_{is,max}$	$\Delta\ \eta_{is,max}$
d_{hole}=0.0mm	1.3580	0.0000	0.8042	0.0000
d_{hole}=1.0mm	1.3575	-0.0005	0.7971	-0.0071
d_{hole}=1.5mm	1.3574	-0.0006	0.7955	-0.0087
d_{hole}=2.0mm	1.3573	-0.0007	0.7941	-0.0101
d_{hole}=2.5mm	1.3602	0.0022	0.7974	-0.0068
d_{hole}=3.0mm	1.3568	-0.0012	0.7896	-0.0146

It is indicated that the compressor performance is improved by holed casing treatment when M/M_{design} is larger than 1.0. The performance of compressor with holed casing treatment decreases slightly at the conditions of M/M_{design}<1.0. The efficiency loss changes with hole diameter. The drop of top efficiency reaches the maximum of 1.46% in the case that the hole diameter equals to 3.0mm. When the hole diameter is 2.5mm, the compressor performance is optimal. The loss of top efficiency for 2.5mm hole is less than 0.7% and the maximum static pressure ratio doesn’t decline. In the other 4 cases, the loss of top efficiency increases with hole diameter.

4.2 INFLUENCE ON COMPRESSOR STABLE WORKING RANGE

The SMI and CMI of different diameters holed casing compressors are plotted in Fig. 8. Similar to the compressor performance, when the hole diameter equals to 2.5mm, the compressor obtains the maximum SMI. In the other 4 cases, the value of SMI is close to each other. Different from SMI, the value of

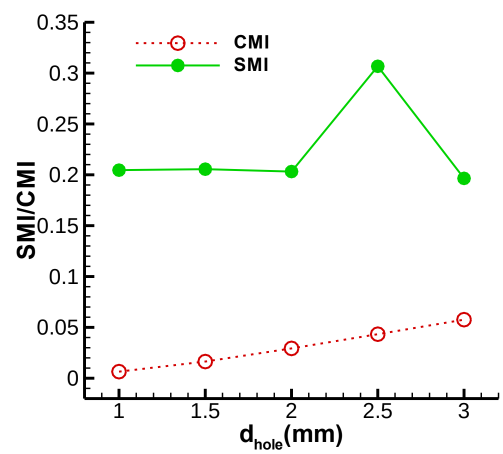


Fig. 8 SMI and CMI of different cases

Both of the stall margin and the choked margin of the compressor can be expanded with holed casing treatment. The performance of compressor with holed casing treatment can also be improved when compressor inlet mass flow is larger than M_{design}. The optimal hole diameter 2.5mm is found for the compressor, compared with the other cases. When the hole diameter equals to 2.5mm, the value of SMI is maximum and the compressor performance at low flowrate conditions is the best. The value of CMI and the performances at large flowrate conditions are moderate.

When hole diameter changes, the flow amount in the hole changes, and the flow pattern in the hole changes, too. The compressor performance is optimal while the hole diameter is 2.5mm and the reason may be attributed to the flow amount in the hole or the flow pattern in the hole. Analysis on flow amount in the hole and flow field in the hole will be given.

4.3 FLOW AMOUNT IN THE HOLE

The flow amounts in the holes are obtained at different flowrate conditions for 5 cases and the values are plotted in Fig .9. The maximum and minimum values of flow amounts are listed in

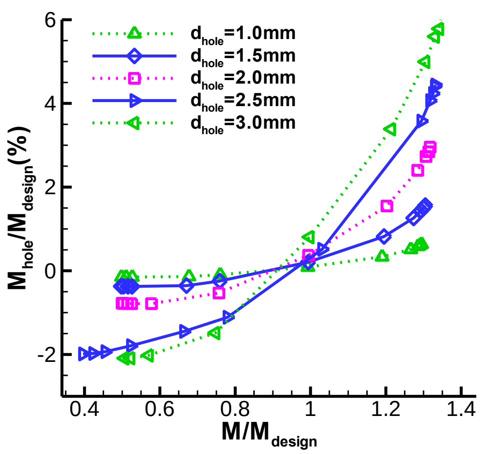


Fig. 9 Flow amounts in holes of different cases

Tab. 4 Minimum and maximum flow amounts in holes

Case	Min (%)	Max (%)
d_{hole}=1.0mm	-0.147	0.636
d_{hole}=1.5mm	-0.370	1.559
d_{hole}=2.0mm	-0.777	2.957
d_{hole}=2.5mm	-1.987	4.440
d_{hole}=3.0mm	-2.089	6.018

The ordinate in Fig. 9 is the relative value of the total flow amounts in all 64 holes divided by M_{design} and the abscissa is still M/M_{design}. The flow amount is defined as positive if the flow direction in the hole coincides with that in the impeller passage and the flow in the holes is called bypass flow. Otherwise the flow direction is inverse and it is called reinjected flow. It is shown in Fig. 9 that the tendency of the 5 curves coincides with each other. For all the five cases studied in this paper, there is a flowrate point at which the flow amount in the hole equals to zero and this point is close to M/M_{design}=1.0 which is the design point. When the compressor inlet mass flow is larger than M_{design}, there is a bypass flow in the hole. When the compressor inlet mass flow is less than M_{design}, there is a reinjected flow in the hole. In general cases, the high performance of the compressor impeller at design point is expected to be maintained. The zero flow amounts in holes at design point can meet the requirement and ensure the self-adaptive effect.

It is shown in Fig. 9 and Tab. 4 that the flow amounts in holes are influenced significantly by hole diameter. At the same flowrate conditions, the flow amount in holes increases with hole diameter. The flowrate in holes nearly doesn't change with the hole diameter at the designed point.

For the compressor studied in this paper, the choke starts at the front part of the impeller passage. At large flowrate conditions, there is a bypass flow in the hole, and the flowrate going into the impeller passage inlet decreases, so the choke is retarded, the choked margin is expanded and the working range is improved. This effect is reinforced when the hole diameter increases because the bypass flow amount increases with hole diameter, so the value of CMI and the compressor performance at large flowrate conditions increase with hole diameter.

It's different for SMI though the reinjected flow amount also increases with hole diameter. The optimum diameter is 2.5mm instead of 3.0mm. In the case of 2.5mm, the value of SMI is maximal and the compressor performance at small flowrate conditions is the best despite the non-maximal reinjected flow amount. So the flow field in the hole at small flowrate condition needs to be analyzed.

4.4 FLOW FIELD IN THE HOLE

To discuss the influence of hole diameter on SMI, five flow patterns in holes of different diameters at flowrate condition of M/M_{design}=0.52 are depicted in Fig. 10. The reinjected flow goes into the hole from the bleed port at the right hand and goes out of the channel chamber from the reinjecting port at the left hand. In the case of d_{hole}=1.0mm, the flow in the hole is nearly uniform with strong laminar flow features. With the increase of hole diamater, the flow becomes spiral movement gradually. It seems that the spiral intensity of the flow in the hole and the channel chamber increases with the rising of the hole diameter. However, in fact the intensity of the flow re-entered is lower in the case of 3.0mm than that of 2.5mm, which can also be found from Fig. 11(c). As shown in Fig. 11(c), the amplitude of the pink line which is the result of 2.5mm is larger than that of the orange line which denotes the result of 3.0mm. It is indicated that the disturbance effect of the former one (d_{hole}=2.5mm) on the main flow into the impeller is stronger than that of the latter one (d_{hole}=3.0mm). The stable working range enhancing capability of the former one is also stronger than that of the latter and it coincides with the experimental results.

Fig. 11(a) is a sketch picture near the reinjecting port and the inlet of the impeller. The left end of the red arc in Fig. 11(a) is near to the pressure side and the other end to the suction side. They are denoted by 1 and 0 respectively after normalization.

The variations of the static pressure are shown in Fig. 11(b). It is found that the number of peak values increases with hole diameter and it is especially evident in the case of 2.5mm. The variations of flow angle in Fig. 11(d) indicate that their tendencies are nearly the same for all 5 cases.

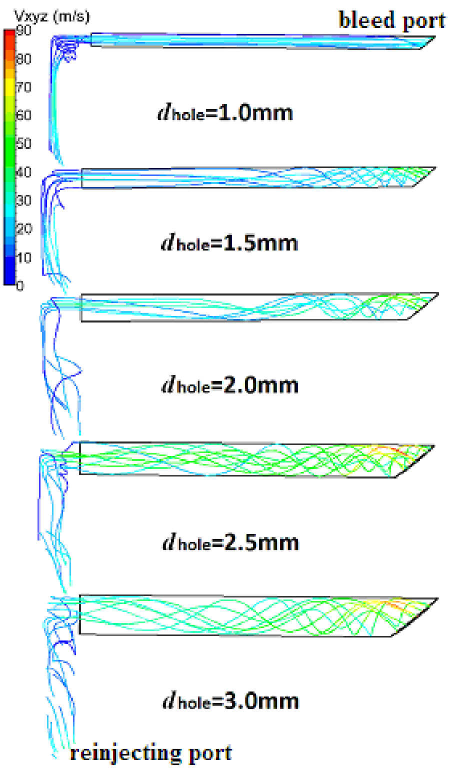
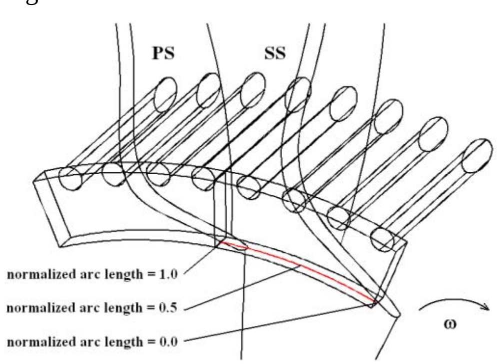
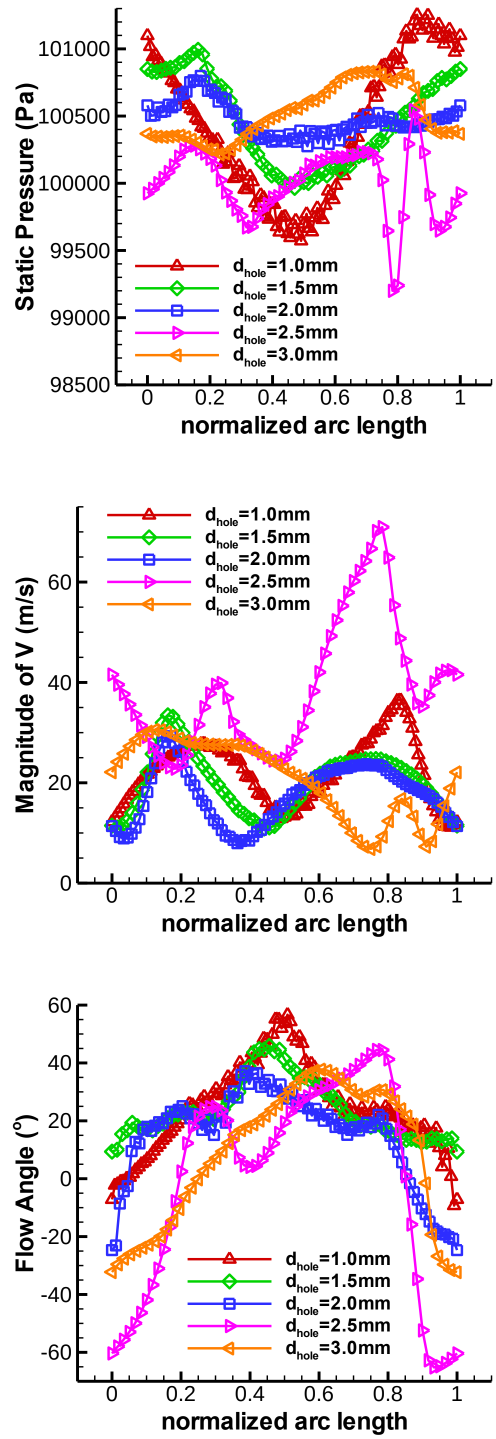


Fig. 10 Flow in holes of different cases

The periodic flow re-enters into the impeller passage inlet and forms a periodic excitation to the flow in the impeller passage. For 2.5mm hole, both the amplitude of the flow coming from the hole and the suppression of the small scale disturbance in the impeller passage are the maximum, as leads to a maximal SMI value. The rotation of the impeller forms an excitation to the gas column in the hole. The frequency is about 6,000Hz. The first order resonance frequency of the gas column is 3,400Hz to 4,100Hz and the second order resonance frequency of the gas column is 10,200Hz to 12,300Hz, so the resonance doesn't occur and there is no additional noise during the compressor operation. When the reinjected flow amount is determined, the influence of holed casing treatment on compressor at small flowrate conditions depends on the amplitude of the flow re-entered into the impeller passage which changes with the hole diameter.



(a) Position



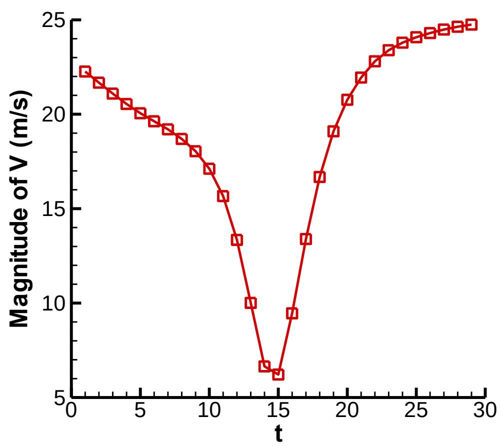
(d) Flow angle of absolute velocity

Fig. 11 Parameters on the arc of different cases

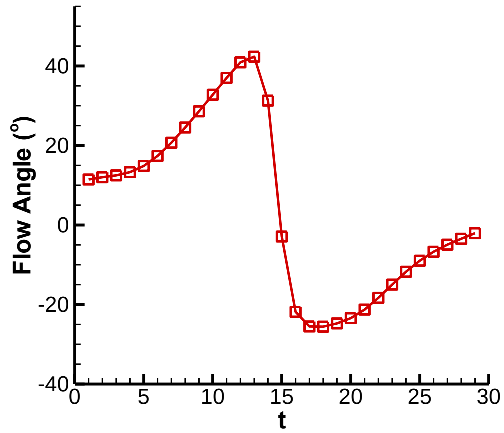
4.5 RESULTS OF UNSTEADY COMPUTATION

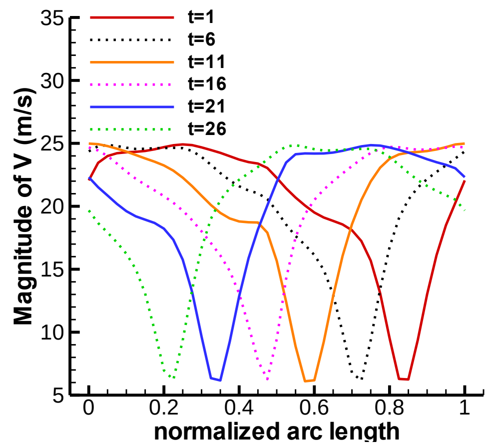
Besides the spiral flow pattern, the flow in the hole is periodic in nature because of the rotation. The frequency of the flow in the hole is $N/60 \times Z$. The results of unsteady analysis at a small flowrate in the case of d_{hole}=3.0mm are shown in Fig. 11 and the mesh for unsteady computation is the same with that for steady computation. The dual time stepping approach proposed by Jameson [24] is employed for unsteady computations in NUECA FINE/Turbo. This approach consists in adding to the time dependent Navier-Stokes equations pseudo-time derivative terms. At each physical time step, a steady state problem is solved in a pseudo time and all available acceleration techniques such as multigrid, local time stepping and implicit residual smoothing can be applied. The angular position per blade passage was chosen to be 40 and so the physical time step is $2\pi\times60/22790/16/40=2.8\times10^{-5}$s. This gave a reasonable and accurate result. 100 iterations for each time step were defined. The time accurate calculation was started using the steady solution as an initial solution.

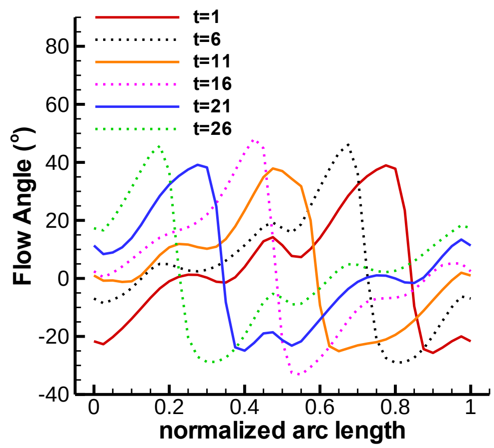
The flow parameters on positions along the red arc in Fig. 11(a) are discussed. First, the distributions of magnitude and flow angle of absolute velocity via time step at a point, whose normalized arc length is 0.5, are shown in Fig 12(a) and Fig. 12(b) respectively. It can be found that the flow is periodic obviously. Both the magnitude and the direction of the velocity change with time. The maximum value of velocity magnitude is nearly 25m/s and the minimum is only 6m/s. The difference between the maximum and minimum flow angle is nearly 70 degrees. The distributions of the magnitude and the flow angle of the absolute velocity along circumferential direction at different time steps are given in Fig. 12(c) and Fig. 12(d). It is indicated that the flow re-entered into the impeller passage is not only periodic with time but also periodic in circumferential coordinate. The influence of holes on flow angle is more distinct than that on velocity magnitude, e.g. the peak values of flow magnitude are almost the same at different time. However, the peak value of flow angle is periodic with time and this is because that the relative positions of the holes to the impellers always



(a) Magnitude of absolute velocity at the point of 0.5







(d) Flow angle of absolute velocity on the arc

Fig. 12 Unsteady results of 3.0mm hole's diameter

4.6 RESU

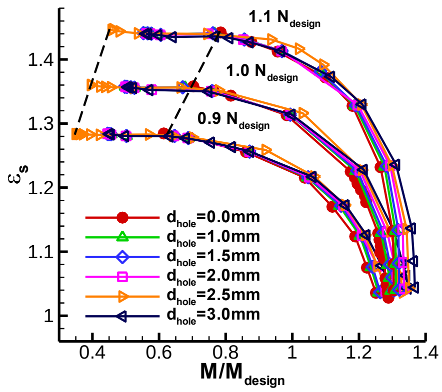


Fig. 13 Static pressure ratio curves at different rotation speeds

The static pressure ratio curves of holed casing treatments of different diameters at different rotational speeds are plotted in Fig. 13. It is shown distinctly that the stall boundary line of the compressor can be put left by holed casing treatment. The values of SMI and CMI at different rotational speeds are close to each other. The optimum diameter is 2.5mm. The maximum SMI of 30.67% can be gotten when hole diameter equals to 2.5mm while CMI is 4.33%. Besides, the efficiency at maximum flowrate is raised by 20%.

5 CONCLUSIONS

The holed casing treatment is a rather simple and effective method for enhancing stable running range of a centrifugal compressor with unshrouded impellers. In the method the most important parameters are the radial position and diameter of the holes. Their good matching will ensure that the compressor can get the best effect of enhancing stable running range and keeping higher efficiency. Both the stall margin and the choked margin of the compressor can be expanded. Five hole diameters, 1.0mm, 1.5mm, 2.0mm, 2.5mm and 3.0mm, have been investigated to study the effects of the holed casing treatment under a given radial position of the holes.

(1) There are two types of flow pattern in the bleeding recirculation holes. One is reinjected flow at the smaller flowrate and the other is bypass flow at the larger flowrate. At the design point, i.e. normal coming flowrate, the flow amount in the bleeding holes is close to 0. The phenomenon occurs in all 5 holes cases and can be used to explain the fairly good self-adaptive effect of the holed casing treatment. Therefore, a key principle of choosing the diameter and the radial position of the hole is to get the self-adaptive effect.

(2) As shown in Fig 13, the stall boundary line of the compressor can be put left obviously. The best effect occurs when the hole diameter is 2.5mm: SMI reaches 30.67% while CMI is 4.33%. Besides, the efficiency at maximum flowrate is raised by 20%.

(3) The flow in the holes is relative uniform when the hole diameter is quite small. It becomes spiral flow when the hole diameter increases. Then the flow re-entered into the impeller passage is periodic. Their amplitude grows with the increasing of hole diameter until it's up to 2.5mm. In the case of 3.0mm, because of strong interaction of the flows coming from adjacent holes, the re-entered flow is suppressed and the amplitude decreases. However, in the case of 2.5mm, the amplitude of the re-entered flow at the inlet of impeller is the maximum among all 5 cases. This is the main reason why the effect of holed casing treatment in the case of d_{hole}=2.5mm is optimal.

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