**NANYANG
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**Experimental and Numerical Evaluation of Aeroacoustic
Damping Performance of In-duct Double-layer Orifice
Plates**

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ABSTRACT

With more stringent noise regulation, more and more researches have been conducted to improve the design of acoustic dampers. As one of the typical acoustic dampers, perforated plates/liners have been widely applied in practice. In this work, aeroacoustic damping performances of in-duct double-layer perforated plates are experimentally studied. For this, a cold flow pipe system with double-layer perforated plates implemented is designed and tested in the Anechoic Chamber at Nanyang Technological University. A centrifugal pump is implemented to generate mean flow through the perforated plate. The mean flow Mach number M_a is varied by changing the flow rate of the pump. Two arrays of microphones are flush mounted to the pipe system in order to determine sound absorption coefficient α characterizing the aeroacoustic damping performance. The noise damping performance of these perforated plate is found to depend on 1) mean flow (also known as bias flow) Mach number M_a , 2) porosity σ (also known as open air ratio), 3) flow direction, 4) the distance L_c between two layers of the plates, 5) the distance d_v of the orifice relative to the plate's centre point. It can be seen that when mean flow Mach number M_a is low, the sound absorption coefficient α is observed to oscillate with increased frequency. However, when Mach number M_a is high, the sound absorption troughs are more separated and made shallower, at the meantime, the absorption peaks are widened and lowered. As the Mach number is further increased, general downward trend of sound absorption coefficient α came into view. When there is no mean bias flow, perforated plates with a smaller porosity σ is found to be associated with a large sound absorption coefficient α . However, when mean bias flow is present, perforated plates with a smaller porosity σ are associated with smaller sound absorption coefficient α . Furthermore, when the flow direction is changed, the damping performance of the

double-layer perforated plates is varied. Increasing L_c gives rise to increased sound absorption coefficient α as the duct flow is present. However, varying d_v is shown to have little effect on the damping performance. Finally, Double-layer perforated plates' damping performances are involved with large sound absorption coefficient α than that of single-layer perforated plates. However, when duct flow Mach number is low, the double-layer orifice plate has the potential to generate noise itself, which weakens the sound absorption ability. Because in practical application the orifices are not always widely separated, so holes interaction effect is also investigated in present work by varying the ratio ζ of orifice diameter D to the orifice centre distance b and orifices distribution pattern. The results show that with the increase of ζ the sound absorption capability is increased, however, an exactly converse trend is appeared as the excitation frequency is beyond 350 Hz and it is also shown that the presence of the bias flow can enhance this effect. When it comes to the orifices distribution pattern, the diamond distribution exhibits more promising results. To benchmark the experiment, a corresponding flow borne acoustic (acoustic field does not alter the mean flow field) simulation is conducted. In order to reduce the computational cost a hybrid approach in frequency domain is adopted. Firstly, the background flow field is calculated using k- ω turbulent model with finer mesh, and in acoustic field the Euler equation is linearized around the mean value obtained from the CFD with relatively coarser mesh. To stabilize the linearized Euler equation, the streamline upwind Petrov-Galerkin (SUPG) is considered. The problem is solved via a direct solver named MUMPS (Multifrontal Massively Parallel sparse) based on LU (lower and upper) decomposition. After comparison, it is found that the numerical results agree very well with the experimental results.

In summary, parametric measurements and numerical simulations are conducted to gain insight on the noise damping performance of double- and single-layer in-duct perforated plate in the presence of a mean flow. The critical roles of 1) M_a , 2) σ , 3) flow direction, 4) L_c and 5) d_v are examined one at a time. Also the effect of holes interaction is investigated. The experimental and numerical findings help to design effective acoustic liners.

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CHAPTER1 Introduction

1.1 Background

Sound, by definition is the phenomena responsible for the sensation of hearing in air and whatever else governed by analogous physical principle [1]. Generally, the same acoustic event and signal may be felt as a pleasant sound in specific circumstances; however, it is unpleasant under other situations. The noise is commonly defined as any kind of undesired sound. There are three basic noise sources [2]. The first one is impact noise; impact noise is generated when two solid objects collide suddenly. In this case, the sound source is a short impulse, which itself is not so strong. The high loudness of many impact noises is induced by ringing, which is the excited vibration of the colliding objectives. The second noise source is flow noise. Depending on the flow geometry and fluid viscosity once the flow velocity exceeds particular limit instability will rise and that is the way flow noise generated. For instance, the noise originated from jet aircraft is one kind of flow noise. The last one is shock wave; it is a plane where the property of a fluid, such as pressure, density, temperature, undergoes sudden change. Generally, the shock waves are generated by the exploitations or in the case where an object is moving with a velocity faster than the speed of sound. A famous example of shock wave is the ‘double bang’ produced by an aircraft flying with a supersonic speed.

The noise pollution brings too many troubles in our life. Long-time exposure to high decibel (loud) sound may lead to hearing impairment including such as loss of the sensitivity to a narrow band of frequencies or impaired cognitive perception of sound. The mechanism is that high

decibel sound damages vulnerable hearing cells, which leads to the permanent injury or death of the cells. Once lost, hearing cannot be restored [3]. Furthermore, if people are exposed to noise while sleeping, the heart rate and blood pressure will increase. Also, the after-effect may give you bad performance on the following day [4]. Another issue is that noise may cause cardiovascular disease. For instance, noise associated with aircraft around Schiphol Airport near Amsterdam is the conspirator behind more medical treatment of heart disease and more drug intake for cardiovascular [5]. Finally, noise pollution is associated with children cognitive development. If young children are always affected by the noise when they are listening to speech and reading, reading and speech difficulties can be developed. The reason is due to the compromised auditory functions. Investigation has shown that when children are learning in noisy classrooms, they have more difficulties to understand speech than those who are learning in quieter settings [6].

Due to the bad influence of noise, recent researches have been shifted on noise regulation. Generally, there are three ways to absorb noise in industrial field. The first is to use the Helmholtz resonator. A Helmholtz resonator is a container with an open neck. The air within the container can be analogous to springs so the air near the open neck can vibrate at a natural frequency, which is associated with the opening neck area, the volume of the cavity and the length of the neck. The highest amplitude sound is generated near and at the natural (resonant) frequency. In this way, the sound energy is trapped by the Helmholtz resonator and the sound level pressure (SPL) at the outlet will be much lower than that at the inlet. The drawback is that noise only near the resonant frequency can be attenuated, which indicates the Helmholtz resonator has a short effective range of sound absorption [7]. The second way is to use porous

and fibrous materials [8]. A porous absorbing material always contains countless cavities, channels or interstices in order to let sound go through them. The air molecules within the cavities are forced to vibrate as the incident sound wave hit the material. As a result, the kinetic energy of air molecules is transferred to heat through the viscous and thermal losses. The drawback is that porous and fibrous materials are vulnerable to extreme conditions so they may lose their function. As a result, they cannot be applied in mobile and jet engines. The third way is to use perforated structures; perforated plates/liners attached to bounding walls are widely applied in combustors as shown in Figure 1-1. They are usually metal sheets perforated with thousands of tiny orifices and worked as acoustic dampers to absorb acoustic disturbances [9-12]. To protect the perforated plates from being burned in such extremely high temperature combustor, a cooling air flow is needed to pass through the orifices. This air flow is also known as bias flow. The main physical mechanism involved in the sound absorption process by perforated plates with the presence of a bias flow is that when pressure waves propagated through the orifices acoustic energy is transferred to kinetic energy as vorticity is generated at edges or corners of the orifice, then the vorticity is convected away by mean flow into places where it cannot continue to interact with the acoustic field intensely, and finally its kinetic energy is dissipated as heat[13]. In this way the acoustic energy is dissipated by the orifice plate and cannot be reflected back to interact with the combustion.

The single-layer perforated plate is widely applied, however, this kind of perforated plate always has problem in the aspect of effective frequency range. So in present research, double-layer in-duct orifice plates are designed and tested aim to extend the effective frequency range.

Figure 1-1: Classical perforated plate and re-designed double-layer perforated plate.

1.2 Motivation and Objectives

Nowadays, with increased stringent noise emission regulations for aircraft engines and automobile exhaust systems [14], more effective noise damping devices are needed. Conventional single-layer orifice plate may hardly meet noise regulations due to its limited effective frequency range. To enhance the sound absorption capacity of perforated plates over a broad frequency range, double-layer in-duct perforated plates are designed and tested. To the best knowledge of the authors, detailed experimental study of this complex configuration of perforated plates has not been reported before. Theoretical models used to predict the sound absorption capacity are available only for the single-layer orifice plate and many details are neglected when predicting the noise damping performance. In addition, numerical models may provide more flow and acoustic information near the perforated plates. However, the numerical results are not reliable sometimes since there is no true value or accepted value to benchmark the simulation. Furthermore, numerical investigations by using Direct Numerical Simulation (DNS) and Large Eddy Simulation (LES) are so computational costly that they are not practical to be

used in practice. However, experimental measurements can reveal the roles of dominant parameters being examined and are of great reliability. Reasons mentioned above partially motivated the present work of conducting parametric measurements to study the noise damping performance of the double-layer-in-duct orifice plates. The critical roles of 1) M_a , 2) σ , 3) flow direction, 4) L_c and 5) d_v are examined one at a time. Comparison is then made between the noise damping performance of in-duct double- and single-layer perforated plates in the presence of a mean flow aim to find out whether the double-layer configuration are more effective and has a broader frequency range. In real application the orifices sometimes are closely separated. However, in most of the previous works, the hole interaction effect is neglected, so in present work, this effect is also investigated by considering the influence bring by factors such as ratio ξ of orifice diameter D to the orifice centre distance b and orifices distribution pattern. Then a series of numerical simulations using the hybrid approach, which has a relatively low computational cost and reasonably accuracy are conducted to benchmark the experimental measurements.

The main objectives consist of:

- (a) Experimental study of the effect of the mean flow (bias flow) on the aeroacoustic damping performance of the in-duct double-layer orifice plate. The Mach number is set to six different values. The damping performance characterized by sound absorption coefficient α is used to indicate the influence bring by variation of Mach number M_a .
- (b) Experimental study of the effect of the porosity σ of the orifice plate on the aeroacoustic damping performance of the in-duct double-layer orifice plate. The porosity is set to two

different values. The damping performance characterized by sound absorption coefficient α is used to indicate the influence bring by the variation of porosity σ .

- (c) Experimental study of the distance L_c between the two layers of the orifice plates on the aeroacoustic damping performance of the double-layer in-duct orifice plates. Four different L_c is investigated one at a time. The damping performance characterized by sound absorption coefficient α is used to indicate the influence bring by the variation of L_c .
- (d) Experimental study of the effect of the distance d_v of the orifice relative to the plate center point. Four different d_v are tested. The damping performance characterized by sound absorption coefficient α is used to indicate the influence bring by the variation of d_v .
- (e) Experimental study of the effect of number of orifice plate's layers N on the damping performance. The sound absorption performance of single-layer in-duct orifice plate is compared with the one of double-layer in-duct orifice plate under different Mach number M_a and porosity σ .
- (f) Experimental study of the effect of ratio ζ of orifice diameter D to the orifice centre distance b on the damping performance. Five different ζ are tested. The damping performance characterized by sound absorption coefficient α is used to indicate the influence bring by the variation of ζ .
- (g) Experimental study of the effect of orifices distribution pattern on the damping performance. Two different orifice distribution pattern is tested. The damping performance characterized by sound absorption coefficient α is used to indicate the influence bring by the distribution pattern.

(h) Numerical study of the noise damping performance of the double-layer orifice plates.

Factors investigated in the experiment are simulated. The damping performance characterized by sound absorption coefficient α is used to indicate the influence brought by these factors.

1.3 Structure and Organization

The report is divided into five chapters, which are briefly summarized as following:

- 1) Introduction. This chapter briefly introduces the background of aeroacoustics and noise dampers. The motivation and the main objectives of conducting this research are described.
- 2) Literature review. This chapter reviews and discusses the extensive research done on perforated liners via analytical, experimental and numerical approaches.
- 3) Experimental evaluation of aeroacoustic damping performance of double-layer in-duct orifice plates and holes interaction effect. In this chapter, the experiment setup and fabrication will be introduced together with the experimental procedure and signal sensor calibration. Then the main results from our experimental measurements are shown with insightful discussion.
- 4) Numerical evaluation of aeroacoustic damping performance of double-layer in-duct orifice plates. This chapter firstly introduced the numerical techniques used in the

simulation including the governing equation and finite element method adopted. Then the main results from our numerical simulation are shown with insightful discussion.

- 5) Conclusions and future work. In this chapter, the key findings obtained from the experiments are summarized and the future work is discussed.

CHAPTER2 Literature Review

Perforated plate has been practically applied in a huge variety of industrial fields including power generating plants, prime movers, construction plants and many kinds of fluid distribution systems. Above mentioned application area are listed in Davis's work [15]. Therefore it is of great importance to put effort on investigating orifice plates and many researchers have contributed a lot in this area by proposing analytical theory, designing experiments and using numerical method.

2.1 Theoretical Studies

The initial studies of acoustic liners are restricted to theoretical modelling [16-21]. In 1970s, Howe [22] and Bechert [21] gave the analytical explanation for the dramatic increased resistance when the bias (mean) flow is presence that acoustic energy is transferred to kinetic energy when vorticity is generated at an edge or corner, then the vorticity is convected away by mean flow into places where it cannot continue to interact with the acoustic field intensely, and finally its kinetic energy is dissipated as heat. Then based on this energy conversion, Howe [23] proposed an analytical model by employing the Rayleigh conductivity for an aperture with infinite thin plate implemented and the presence of a bias flow. Howe[24] in 1997 theoretically investigated the effect of the plate thickness on the fluid structure interaction. It was found that plate thickness has significant role on flow instability and the generation of the self-sustained fluid oscillation.

Wendoloski [25] made an analysis extension for Howe's model by using linearization to the absorption mechanism. In his case the radius of the duct becomes finite. The Green expansion was used in the formulation and a novel technic is formed to solve the convergence problem bring by the introduction of Green expansion. The model is checked by increasing the duct radius to infinite and the corresponding Rayleigh conductivity tends to agree the one calculated by Howe.

Eldredge and Dowling [26] developed a one-dimensional model based on the mechanism that the acoustic energy is transferred to flow energy through excitation of vorticity fluctuation at the edges of the orifices. This model utilized the homogenous liner compliance adapted from Rayleigh conductivities with the presence of a single aperture as well as mean flow. It has been shown that a large portion of acoustical energy can be absorbed by the system and how the liners arranged in the duct system can play an important role on sound absorption ability.

However, many flow details are typically unable to be captured. In addition, it is very challenging to theoretically model perforated liners in a complex configuration, such as a double-layer perforated plates being implemented in a duct with the existence of a mean flow. This indicates that an experiment needs to be conducted to predict the aeroacoustic damping performance of perforated plates in such complex configuration. This partially motivated the present work.

In most of the previous theoretical works mentioned above, the way of evaluating acoustic behaviour of the perforated plate is through isolation of a single hole from hundreds and

thousands of holes on the perforated plate under the assumption that the holes on the perforated plate are working separately. However, in real application the holes are sometimes closely separated, so it is very meaningful to pay attention on the holes interaction effect.

The research related holes interaction effect can be dated from Ingard [27]. In his work, a comprehensive study about orifice plate is conducted. Two eccentric holes with different distance between two holes' centre are used to investigate the holes interaction effect. He found that this effect is strongly related to the end correction. The widely accepted method used to deal with the holes interaction effect is proposed by Fok [28]. In his work, a function is derived by considering centre to-centre-distance between orifices and correction for the radiating impedance of orifice in linear regime (low sound level). To accurately account for the holes interaction and heterogeneity effect Tayong [29, 30] develop an expression including the geometry tortuosity, which is originally used for the granular materials. He showed that as the centre-to-centre distance is decreased, the peak of absorption curve is shafted to low frequency without decrease of peak value.

Above mentioned works related to holes interaction effect did not take the effect of bias flow into account. Considering the interaction between apertures, Lee, at al. [31] proposed a model to describe the acoustic impedance of perforated plate with the presence of bias flow. The model is fulfilled by solving incompressible linearized Euler equation. It is shown that the reactance is decreased as the increase of the porosity (open area ratio). The reason was explained as the decrease of the attached mass due to orifices interaction effect.

However, though Lee, et al. considered the effect of the bias flow, they did not investigate the effect of ratio ξ of orifice diameter D to the orifice centre distance b on the noise damping performance under various bias flow velocity. Tayong et al. studied the orifice distribution effect, however, no mean flow is present in his experimental setup. These limitations partially motivate authors to conduct present research.

2.2 Experimental Studies

As a way to reduce the aircraft noise radiated from air jet engine, acoustic liners which consists multiple millimetre-size small holes are widely applied to stabilize propulsion systems [32]. The main mechanism is that the acoustic energy is transferred kinetic energy of vortex. Two problems exist when implementing the acoustic liners, first it is of great importance of attenuating the noise in the wanted range the other is the cooling [33]. A lot of efforts have been put to experimentally investigate acoustic liners. [12, 19, 34-42]. Ingard and Labate [43] in 1950 firstly visually investigated the effect of incident sound amplitude, frequency, plate thickness on the induced motion of the fluid near orifice.

It is known that for perforated plate under low frequency excitation, the presence of bias flow can greatly increase the acoustic resistance. The physical process was experimentally investigated by Ingard and Ising [44] in 1967. Through simultaneously measuring the acoustic velocity at the aperture and pressure fluctuation, they found that the relation between pressure and velocity is linear in low pressure and in high pressure the relation approaches a square law. In addition to determine the acoustic impedance and its resistance part, the effect of

superimposed mean flow produced by suction through an outlet was investigated. It is shown that the orifice impedance increases proportionally with the increase of steady mean flow.

Hughes and Dowling [45] in 1990 found that if a solid plate was placed behind the screen the sound absorption ability of the absorber can increase dramatically. They also predict that if the geometry and bias flow are selected optimally, the noise can be completely absorbed and their experimental results benchmarked their prediction.

To understand the acoustic characteristics of a liner with the presence of bias flow, Jing and Sun [46, 47] in 1999 experimentally investigated the perforated liners' acoustic property with bias flow. It was found that firstly the presence of the bias flow can significantly increase the absorption ability in the aspect of higher sound absorption coefficient and wider effective band. Then the plate thickness plays a significant role to acoustic properties. Finally, the influence of different way to generation bias flow is found to be negligible.

Zhao, Morgans and Dowling designed a moveable pipe system to suppress the combustion instability. The experiment results shows that the damping ability depends on the length of the pipe and the bias flow velocity [11, 48].

To test the computational aeroacoustics code and better understand the physics of acoustic liners with grazing flow, Tam et al. conducted an experiment using the grazing flow impedance in 2014 [49]. Noise generated by acoustic liners is noted and the feedback acoustic resonance mechanism is the cause of the self-noise when grazing flow is presence.

The experimental measurements confirm the effect of dominant parameters being examined and are of great reliability. But it is very challenging (unpractical) to conduct parametric measurements, especially when there are too many parameters involved in practical design of perforated liners. Furthermore, some details of small scale cannot be observed during the conduction of experiments. This partially motivated the development of the numerical techniques.

2.3 Numerical Investigation

By 1995, computational fluid mechanics (CFD) has been widely applied in aerodynamics and aircraft design. Tam [50] suggested using CFD method to solve aeroacoustics problems. The question is that most aerodynamics problems are time independent but most aeroacoustics problems are time dependent. Further, the nature, characteristic and objectives of CFD problems are quite different from aeroacoustics problems. Finally, some computational issues are unique to aeroacoustics. Therefore independent thinking and development are required for computational aeroacoustics (CAA).

Due to the fact that the opening of the resonator liner is physically small hence flow and acoustics field cannot be observed by direct experiments. In 2000, Tam [51] and Kurbatskii did a direct numerical simulation (DNS) to simulate the flow around and inside a typical liner resonator under the excitation of an incident acoustic field. By applying carefully designed grid, small-scale features of the flow field can be captured. From the DNS result they found that when sound intensity is low, acoustic dissipation comes mainly from viscous losses and when sound

intensity is high, dissipation is due to the shedding of micro-vortices at the mouth of the resonator. A co-ordinate experiment was conducted to benchmark the simulation result and good agreement was found between measured absorption coefficients of physical experiment and direct numerical simulation over a broad range of frequencies and incident sound pressure level [52]. Through the validation, it is proved that there is sufficient confidence in the use of direct numerical simulation to explore the physics of acoustic liners.

Large eddy simulation (LES) is a mathematical model for turbulence flow simulation in fluid dynamics, and it is now widely applied in fields including acoustics, combustion and simulation of atmospheric boundary layer. Compared with DNS, which resolve for all eddies, LES only resolve large eddies. In this way the computational cost is reduced.

In 2009 [53], Mendez and Eldredge reported a large eddy simulation for the acoustic effect of perforated plates. Their study demonstrated that the compressible large eddy simulation can provide data on the flow around perforated plates and related acoustic damping can be captured. Then existing models including Howe's model (HM), Howe's model accounted effect of plate thickness (MHM), Jing and Sun numerical model (JSM) are assessed by the large eddy simulation results.

Although, scattering matrix obtained by DNS and LES can lead to high quantitative accuracy, the unavoidable drawback is the expensive computational effort. To reduce the high computational costs and complexity, a hybrid method is proposed [54-57]. The main idea is to compute the flow field independently from acoustic field and vice versa. In 2010, Kierkegaard,

Boij and Efraimsson applied this hybrid method to investigate sound propagation in a straight duct with an orifice plate and a mean flow present in frequency domain with linearized Navier-Stocks equation. Transmission and reflection results at the orifice plate obtained from the two-port scattering matrix are compared with experimental measurement with good agreement. Hence this method is found to be efficient for flow borne sound problems.

The Lattice-Boltzmann method based on kinetic gas theory, offers known advantage over other numerical methods based on Navier-Stocks equations for subsonic compressible flow. For example, corrugated shape-edge orifice in the computational domain will not considerably increase the computational cost. From first principles, The LBM is a computational fluid dynamics tool and has accurately modelled a lot of complex fluid phenomenon [58] such as flow over air foils [59] and flow inside rectangular cavities [60]. Also in the field of aeroacoustics, the LBM is often used in for low Mach number application. In 2010, Lew et al. [61] investigated the far-field noise generated by an unheated turbulent axisymmetric jet using the Lattice-Boltzmann method, Near-field flow quantities such as turbulence intensities and centreline velocity decay rate were in agreement with results from comparable LES results and experimental measurements. Therefore the LBM can be a viable approach, from their work; they found the main advantage of LBM is the ease of including nozzle in the computational domain and the ease of investigating nozzles with complex geometries.

By applying the two- and three-dimensional LBM [62-64], Chenzhen Ji and Dan Zhao evaluated the absorption mechanism and acoustic damping performance of perforated plate with different aperture edges in the presence of a bias flow. The damping performance of orifice plate with

different edges is examined in the time domain by forcing an oscillating flow with multiple tones. Their results agree well with theoretical and experiment results. It is found that the orifice with square edge has better absorptive ability than the circular one and the maximum sound absorption depends strongly on bias flow Mach number and plate thickness. The drawback is the compressibility criterion because of the commonly used Bhatnagar-Gross-Krook (BGK) assumption; this approximation reduces computational cost but at the mean time limits the Mach number to 0.5. However, the limitation can be solved by applying high order Lattice Boltzmann schemes [65].

In present work, to validate the experiment measurements and gain insight on the flow and acoustic field, a series of numerical simulations are conducted. Considering the computational cost and accuracy, the hybrid method proposed by Kierkegaard is adopted.

CHAPTER3 Experimental Evaluation of Noise Damping Performance of Double-layer In-duct Orifice and Holes Interaction Effect

3.1 Description of Experiments

To investigate the aeroacoustic damping performance and optimize the design of the in-duct double-layer perforated plate, a series of experiments are conducted on 27 different configurations of double-layer perforated plates in this chapter together with 7 orifice plates for the purpose of investigating the holes interaction effect. The fabrications of main components are introduced first. Both perforated plates and cylindrical pipes are made by applying 3D printing technology. Then the setup of a cold flow duct system is described. Two arrays of Brüel & Kjær 4957 microphones are implemented both in the upstream and downstream section of the duct. Before applying these microphones, the calibration is conducted as described in the following subsection. Finally, the damping noise performance characterized by sound absorption coefficient α is determined by applying the classical two-microphone method.

3.1.1 Experimental Equipments

All material used for the fabrication of experimental components are made of acrylic. The acrylic has the property of transparency, which allows researchers to apply colored gas setup. Present experimental setup has seven important components.

(a) Perforated Orifice Plates

(b) Upstream and Downstream Tubes with Microphones Implemented

(c) Connecting Pipes

(d) Digital Voltage Regulator and Grazing Flow Generator

(e) Digital Function Generator

(f) Data Acquisition (DAQ) Unit

a. Perforated orifice plates

Front orifice plates (OP) and back orifice plates are shown in the Fig. 3.1 and their respective parameters can be found in the Table. 1. Front orifice plates include OP1 to OP6, OP1, OP2 and OP3 have different diameter D of their respective aperture and are associated with investigation of the effect of porosity σ . OP4, OP5 and OP6 have different d_v and they are associated with the effect of the distance d_v of the orifice relative to the plate centre. Back orifice plates include OP7 to OP12, OP7, OP8 and OP9 have different diameter D of their respective aperture and are associated with investigation of the effect of porosity σ . OP10, OP11 and OP12 have different d_v and they are associated with the effect of the distance d_v of the orifice relative to the plate centre. OP13 to OP17 with varying ζ are used to study the holes interaction effect. OP18 and OP19 with different are used for the purpose of investigating the orifices distribution effect.

Orifice plates are manufactured using the well-known 3D printing technique. The first step is to design the orifice plate 3D model in CAD software. Then one can import the model to MarketBot Replicator 2 and adjust parameters, for example, resolution of printing, layer height, temperature, infill percentage and speed of the extruder. Finally above parameters are exported to printer and model can be printed.

Table 1: Lists of the orifice plates of double-layer orifice plates configuration.

Name	D (mm)	d_v (mm)	T (mm)	A_o (mm ²)	A_t (mm ²)	σ (%)
Front Orifice Plates						
OP1	4	0	2	50.24	5024	1.00
OP2	6	0		113.04		2.25
OP3	12	0		452.16		9.00
OP4	6	6		113.04		2.25
OP5	6	12		113.04		2.25
OP6	6	16		113.04		2.25
Back Orifice Plates						
OP7	4	0	2	50.24	5024	1.00
OP8	6	0		113.04		2.25
OP9	12	0		452.16		9.00
OP10	6	6		113.04		2.25
OP11	6	12		113.04		2.25
OP12	6	16		113.04		2.25

D is the diameter of aperture, d_v is the distance of the orifice relative to the plate's centre point, T is the plate thickness, A_o is the open area, A_t is the plate area and σ is the open area ratio (porosity).

Table 2: List of parameters of single-layer orifice plates used for the investigation of holes interaction effect

Name	D (mm)	b (mm)	T (mm)	A_o (mm ²)	A_t (mm ²)	P
OP13	4	0	2	100.48	5024	Concentric
OP14		5				Eccentric
OP15		8				
OP16		10				
OP17		20				
OP18		18		200.96		Diamond
OP19		18				Square

b is the centre-to-centre distance, P represents the pattern of orifices arranged on the plate.

Figure 3-1: CAD Drawings of front orifice plates and back orifice plates

Figure 3-2: CAD Drawings of orifice plates used for holes interaction effect.

b. Upstream and Downstream Tubes with Microphones Implemented

The upstream and downstream duct having the same length of 850 mm is made of acrylic materials. The inner and outer diameter is 40mm and 50 mm, respectively.

To measure the acoustic fluctuations, B&K microphones are flush mounted to the upstream and downstream tubes as shown in Fig. 3.2 as pressure sensors. The distance g_1 between the perforated plate and the axis of the first microphone is 50 mm. Distance g_2 and g_3 between the first microphone and second microphone as well as the second microphone and third microphone

is 100 mm. The microphones can be used in the frequency range from 50 to 10000 Hz and dynamic range from 32 to 134 decibel (dB).

Figure 3-3: CAD Drawing of a tube with microphones implemented.

c. Connecting Pipes

The connecting pipes are used to connect the front and back perforated plates. They are fabricated with the same inner and outer diameter as the upstream and downstream ducts. The length is varied from 30 mm to 100 mm to study its effect on noise damping performance.

Figure 3-4: CAD Drawings of connecting pipes with varying length from 30 mm to 100 mm.

d. Bias Flow Generating System

The bias (mean) flow generating system consists of a centrifugal pump and a diffusing box. The use of the diffusing box is to make the flow stable. The centrifugal pump is controlled by a voltage regulator. The mean flow velocity is measured using hot wire anemometer in front of the front perforated plate.

e. Digital Function Generator

Tektronix AFG320, a 16 MHz function generator has been applied in this experiment to drive the loudspeaker. This function generator are capable of producing waveforms of noise, DC, pulse, ramp, triangle, square, sweep, burst and sinusoid. One can adjust the frequency from 0.01 Hz to

16 MHz and amplitude from 50 mV_{p-p} to 10 V_{p-p}. In present measurements monochromatic sinusoidal waves are generated with frequency varied from 110 Hz to 700 Hz. Also, the input sound can be amplified by the Crown D-75, which is one of widely used amplifiers.

f. Data Acquisition (DAQ) Unit

The measured pressure data are logged through LabVIEW2010 via Data Acquisition (DAQ) unit consisting of NI PXIe-1062Q. The detailed description of the DAQ unit can be referred in the work of D. Zhao [12] .

3.1.2 Experimental Setup

Figure 3-5: Schematic of experimental setup for double-layer orifice plates.

To study the aeroacoustic damping performance of double-layer induct orifice plates, a cold-flow axial duct system containing a length of circular duct is experimentally set up in the anechoic chamber, the schematic drawing is shown above. A centrifugal pump is used to provide a mean

flow through the duct system and the mass flow rate is varied by a voltage controller. To make sure the flow is stable and uniform before entering the duct system, a rectangular diffusing box is connected to the inlet of the duct. Within the diffusing box, a loudspeaker is implemented to generate sound waves with sound pressure level (dB) in the range of 90 to 130 dB and frequency from 200 Hz to 600 Hz with a step of 10 Hz. The loudspeaker is driven by a digital function generator and an amplifier. The acoustic waves superimposed with a mean flow propagate from the inlet of the duct to the outlet opening. Double-layer induct orifice plates is applied to dissipate the acoustic noise. To monitor the pressure perturbation, an array of B&K 4957 microphones is inserted into the upstream- and downstream duct wall. Before the experiment is conducted microphones are appropriately calibrated by using a piston phone and a calibration box. The detailed information related to microphone calibration is described in following section.

Figure 3-6: Schematic of experimental setup for single-layer orifice plates used for holes interaction effect.

3.1.3 Pressure Sensors Calibration

Theoretically, all six microphones should measure the signal with the same amplitude and phase. However, due to the manufacture error, there may be some deference and the deference of amplitude and phase should be considered in the following date processing section. Therefore microphones must be calibrated before the experiment is conducted. The procedure is firstly inserting microphones on the top of the calibration box with a circular array as shown in figure below. The next step is switching on the function generator and amplifier so that the loudspeaker inside the calibration box can generate sinusoidal waves from 110 Hz to 700 Hz with a step of 10 Hz. Finally, data is collected by microphones.

Figure 3-7: CAD Drawing of a typical microphone calibration setup.

A widely applied method of microphone calibration is to use the transfer function $\hat{H}_{12}(f)$. All microphones are calibrated based on microphone 1 in present microphone calibration procedure. Then transfer function with respect to microphone 1 and microphone 2 is

$$\hat{H}_{12}(f) = \hat{S}_{12}(f)/\hat{S}_{11}(f) \quad (3.1)$$

where the f is the incident sound wave frequency. $\hat{S}_{11}$ is the auto-spectral density of the pressure measured at location x_1 , $\hat{S}_{12}$ is the cross spectral density between pressure acquired at point x_1 and x_2 .

$$\hat{S}_{11}(f) = \left(\frac{1}{T}\right) \{\hat{P}_1(f, t) \hat{P}_1^*(f, t)\} \quad (3.2)$$

$$\hat{S}_{12}(f) = \hat{C}_{12}(f) + i\hat{Q}_{12}(f) = \left(\frac{1}{T}\right) \{\hat{P}_1(f, t) \hat{P}_2^*(f, t)\}$$

where $\hat{P}_1(f, t)$, $\hat{P}_2(f, t)$ are the Fourier transformation of the time-domain pressure signal at measured point x_1 and x_2 , and $\hat{P}_1^*(f, t)$, $\hat{P}_2^*(f, t)$ are their corresponding conjugate transpose. T is the record length of time series, which is 1×10^{-4} s in present experiment.

The phase change is denoted as $\hat{\Phi}_{12}(f)$, it is can be calculated as

$$\hat{\Phi}_{12}(f) = \tan^{-1} \frac{\Im[\hat{H}_{12}(f)]}{\Re[\hat{H}_{12}(f)]} \quad (3.3)$$

where the $\Re$ and $\Im$ stands for the real and imaginary part of the transfer function $\hat{H}_{12}(f)$.

Therefore, with the microphone 1 is set as the reference, the spectra are corrected as:

$$\hat{S}_{11}(f) = [\hat{S}_{11}(f)]_u \quad (3.4)$$

$$\hat{S}_{22}(f) = [\hat{S}_{22}(f)]_u / [\hat{H}_{12}(f)]^2 \quad (3.5)$$

$$\hat{S}_{12}(f) = [\hat{S}_{12}(f)]_u / [\hat{H}_{12}(f)] \quad (3.6)$$

where the subscript u stands for uncorrected auto- and cross-spectral density.

Figure 3-8: Pressure amplitude ratio of microphone 1 to microphone 6 with respect to pressure measured by microphone 1.

Figure 3-9: Phase difference of microphone 1 to microphone 6 with respect to pressure phase measured by microphone 1.

From Figure 3-8 and Figure 3-9, the difference of amplitude measured by microphones 1 to 6 is quite obvious. Therefore if the measured difference among microphones is neglected, great error will be generated when predicting the aeroacoustic damping performance of double-layer induct orifice plates. However, phase difference among microphones 1 to 6 is negligible.

3.1.4 Characterizing Noise Damping Performance

Figure 3-10: Schematic drawing of double-microphone method setup with microphone 1 located at x_1 and microphone 2 located at x_2 .

To characterize the aeroacoustic damping performance of the double-layer in-duct orifice plates, two-microphone method is applied in present work. Figure 3-10 is a typical double-microphone method setup [66-68]. In the upstream section, the incident wave pressure is denoted as $\hat{p}_{ia}$ and reflected wave pressure is denoted as $\hat{p}_{ra}$. In the downstream section, the incident wave pressure is denoted as $\hat{p}_{ib}$ and reflected wave pressure is denoted as $\hat{p}_{rb}$.

$$\hat{p}_{ia}(x, t) = Ae^{i(\omega t - k_i x)}, \quad \hat{p}_{ra}(x, t) = Be^{i(\omega t + k_r x)} \quad (3.7)$$

where k_i and k_r denote the wave number of waves propagating in the flow direction and against the flow direction in the upstream section:

$$k_i = \frac{\omega}{c + v} = \frac{k}{1 + M_a}, k_r = \frac{\omega}{c - v} = \frac{k}{1 - M_a}, M_a = \frac{v}{c} \quad (3.8)$$

where A and B are pressure amplitude of incident and reflected waves, respectively. ω is the angular frequency. $c = 343$ m/s is the speed of sound. v is the mean flow velocity, M_a is the Mach number.

The acoustic pressure at location x_1 and x_2 are

$$\hat{p}(x_1, t) = \hat{p}_{ia}(x_1, t) + \hat{p}_{ra}(x_1, t) = [Ae^{-ik_ix_1} + Be^{ik_rx_1}]e^{i\omega t} \quad (3.9)$$

$$\hat{p}(x_2, t) = \hat{p}_{ia}(x_2, t) + \hat{p}_{ra}(x_2, t) = [Ae^{-ik_ix_2} + Be^{ik_rx_2}]e^{i\omega t} \quad (3.10)$$

In one-dimension wave field, the incident wave auto-spectral density can be described as $\hat{S}_{AA}$, reflected wave auto-spectral density can be described as $\hat{S}_{BB}$ and cross spectral density between incident and reflected wave is described as $\hat{S}_{AB}(f) = \hat{C}_{AB}(f) + i\hat{Q}_{AB}(f)$, $\hat{C}_{AB}$ is the real part and $\hat{Q}_{AB}$ is the imaginary part.

The auto- and cross-spectral density of incident and reflected wave can be related with the auto- and cross-spectral density of pressure at measured point x_1 and x_2 as:

$$[A][\hat{S}_{AA}, \hat{S}_{BB}, \hat{C}_{AB}, \hat{Q}_{AB}]^T = [\hat{S}_{11}, \hat{S}_{22}, \hat{C}_{12}, \hat{Q}_{12}]^T$$

where $[A]$ is a 4×4 matrix.

$$\begin{aligned} a_{14} &= 2\sin(k_i + k_r)x_1 & a_{13} &= 2\cos(k_i + k_r)x_1 & a_{24} &= 2\sin(k_i + k_r)x_2 \\ a_{23} &= 2\cos(k_i + k_r)x_2 & a_{32} &= \cos k_r(x_1 - x_2) & a_{31} &= \cos k_i(x_1 - x_2) \\ a_{41} &= \sin k_{ia}(x_1 - x_2) & a_{42} &= -\sin k_r(x_1 - x_2) & a_{11} &= a_{12} = a_{21} = a_{22} = 1 \\ a_{33} &= \cos(k_rx_1 + k_ix_2) + \cos(k_ix_1 + k_rx_2) & a_{34} &= \sin(k_rx_1 + k_ix_2) + \sin(k_ix_1 + k_rx_2) \\ a_{43} &= \sin(k_ix_1 + k_rx_2) - \sin(k_rx_1 + k_ix_2) & a_{44} &= \cos(k_rx_1 + k_ix_2) - \cos(k_ix_1 + k_rx_2) \end{aligned}$$

The same procedure can be applied to calculate the incident wave auto-spectral density $\hat{S}_{CC}$, and reflected wave auto-spectral density $\hat{S}_{DD}$, also cross-spectral density between incident and reflected wave $\hat{S}_{CD}$ in the downstream. Finally, one can characterize the damping performance with sound absorption coefficient denoted as α representing the fraction of energy dissipated by the sound absorption devices.

$$\alpha = 1 - \frac{|\hat{S}_{CC}(f)| + |\hat{S}_{BB}(f)|}{|\hat{S}_{AA}(f)| + |\hat{S}_{DD}(f)|} \quad (3.11)$$

where $\hat{S}_{AA}(f)$, $\hat{S}_{BB}(f)$ stands for the auto-spectral density of incident and reflected wave of the upstream and $\hat{S}_{CC}(f)$, $\hat{S}_{DD}(f)$ stands for the auto-spectral density of incident and reflected wave of the downstream. The reflection coefficient describing how much of a pressure wave is reflected by an impedance discontinuity in the transmission medium is defined as:

$$R = \left| \frac{\hat{S}_{BB}(f)}{\hat{S}_{AA}(f)} \right| \quad (3.12)$$

The acoustic impedance $z(\omega)$, which measures of the opposition that a system presents to the acoustic flow resulting of an acoustic pressure applied to the system can be calculated as:

$$z(\omega) = \frac{\exp(2ikx) + R(\omega)}{\exp(2ikx) - R(\omega)} \quad (3.13)$$

where x is the location where the front orifice plate is located, ω is the excitation angular frequency. Also here define the real part of acoustic impedance is resistance $\Re(z(\omega))$ and imaginary part of acoustic impedance is reactance $\Im(z(\omega))$.

3.2 Results and Discussion

The experiment is conducted and following main parameters are investigated one at a time through the switch of the in-duct orifice plates, connecting pipes as well as the mean duct flow Mach number.

- 1) Mean flow (bias flow) Mach number M_a
- 2) Porosity σ_1 and σ_2
- 3) Distance L_c between two layers of orifice plates
- 4) Distance d_v of the orifice relative to the plate centre point
- 5) Number of orifice plate's layers N
- 6) Ratio ζ of diameter D to orifice distance b
- 7) Orifices distribution pattern

3.2.1 Experimental Measurement Errors and Uncertainty Analysis

Physical quantity cannot be measured with full certainty; the appearance of errors cannot be avoided in various experiments. Experimental error is defined as the difference between a measured value and the accepted value or the difference between two measured values. accuracy and precision are always applied to evaluate experimental error [69, 70].

Accuracy represents the difference between the measured value and the true value or value being accepted. Since a true or accepted value for a physical quantity is always unavailable, it is sometimes hard to calculate the accuracy of a measurement. Instead, Precision represents how closely two or more measurements agree with each other. Precision is also known as 'repeatability' or 'reproducibility'. In other words, a measurement having the property of

repeatability and reproducibility can generate multiple data sets with negligible difference each time.

Because in this case the real values are not available, so the experimental error only can only be described by precision. The repeatability tests are conducted and the results are shown below.

Figure 3-11: Repeatability tests for the front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm and $M_a = 0.037$.

Then one can use Percent Difference (PD) to describe Experimental Precision. Percent Difference calculates precision of two measurements expressed as the ratio of the absolute value of difference between the measurement value E_1 and E_2 to the average of the two values. The Percent Different (PD) is represented as:

$$PD = \frac{|E_1 - E_2|}{\frac{E_1 + E_2}{2}} \quad (3.14)$$

where E_1 , E_2 are measured experimental values.

Form figures below one can find out that this experiment possesses the repeatability or reproducibility.

Figure 3-12: Percent difference between test 1 and test 2; test 2 and test 3; test 1 and test 3 for the front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm and $M_a = 0.037$.

3.2.2 Effect of Mean Flow Mach Number M_a

The effect of duct flow Mach number M_a on the aeroacoustic damping performance of the double-layer in-duct orifice plate is investigated in this section. It has been observed that as the orifice plate with $D = 12$ mm is implemented in the configuration of the double-layer in-duct orifice plate whether as the front or the back orifice plate, the sound absorption coefficient α is periodically changed over the measured frequency range. In addition, in figures describing the variation of sound absorption coefficient α with duct flow Mach number M_a , at the first stage, the sound absorption coefficient α is increased with the increased duct flow Mach number M_a . However, in the second stage, the sound absorption coefficient α is decreased with the increased duct flow Mach number M_a where the frequencies selected are corresponding to the local maximum sound absorption coefficient. These findings are shown in the Figure 3-13 and Figure 3-14 with the orifice plate having $D = 12$ mm implemented as the front orifice plate. Also in Figure 3-15 and Figure 3-16 with the orifice plate having $D = 12$ mm implemented as the back orifice plate. The same observations are found when investigating the aeroacoustic damping performance of single-layer in-duct orifice plate in present experiment and in available literatures such as work done by Eldredge [26] and Zhao [71]. To make it more clear, a dash line describing the aeroacoustic damping performance of single-layer orifice plate is added in Figure 3-13.

Furthermore, compared with the cases when the orifice plate with $D = 12$ mm is implemented in the double-layer in-duct orifice configuration, when the orifice plate with $D = 12$ mm is replaced by orifice plates with smaller aperture diameter, how sound absorption coefficient α varied with the frequency and Mach number M_a is significantly different. It is most likely duo to the

diameter of $D = 12$ mm is too large, so configurations including the orifice plate with $D = 12$ mm is not considered as double-layer orifice plates in following discussions.

It can be seen from the Figure 3-17 that when the Mach number M_a is low, there are sound absorption peaks and troughs over the low frequency range. However, when the Mach number M_a is increased to 0.023, the absorption troughs are more separated and made shallower. In addition, the absorption peaks are widened as well. In other words, the sound absorption coefficient α seems does not depend on the frequency so much. Furthermore, when the Mach number M_a is further increased, the general downward trend of sound absorption coefficient α is observed. Finally, it is also has been observed that the sound absorption coefficient α becomes negative in the frequency range of 540 Hz to 590 Hz with Mach number $M_a = 0.016$. This indicates the double-layer in-duct orifice plate is not damping on the noise but producing sound. Since the front and back orifice plate can compose a cavity the negative sound absorption coefficients can be explained by the resonance whistling effect [72]. Detailed investigation will be conducted in the future work.

In Figure 3-18, it can be seen that the sound absorption coefficient α is periodically changed over the whole frequency domain when Mach number M_a is low. However, when Mach number M_a is increased, the absorption troughs are more separated and made shallower. In addition, the absorption peaks are widened as well. When the Mach number M_a is further increased, general downward trend of sound absorption coefficient α is also observed.

Figure 3-13: Variation of measured sound absorption α with (a) frequency; (b) Mach number M_a for front orifice plate $D = 12$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm. For the single-layer orifice plate, $D = 12$ mm, $T = 2$ mm and $d_v = 0$ mm.

Figure 3-14: Variation of measured sound absorption α with (a) frequency; (b) Mach number M_a for front orifice plate $D = 12$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm.

Figure 3-15: Variation of measured sound absorption α with (a) frequency; (b) Mach number M_a for front orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 12$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm.

Figure 3-16: Variation of measured sound absorption α with (a) frequency; (b) Mach number M_a for front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 12$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm.

Figure 3-17: Variation of sound absorption α with frequency for front orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm.

Figure 3-18: Variation of sound absorption α with frequency for front orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm.

Figure 3-19: Variation of sound absorption α with frequency for front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm.

Figure 3-20: Variation of sound absorption α with frequency for front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm.

From Figure 3-19, it can be seen that at low Mach number M_a the sound absorption coefficient α is periodically varied with the frequency over the range from 110 Hz to 500 Hz. However, the sound absorption coefficient α is dramatically decreased at 500 Hz and remained at a very low value of around 0.2 until 700 Hz. It is may be explained that the double-layer in-duct orifice plate may generate noise in the range of 500 Hz to 700 Hz. When at high Mach number M_a , it can be seen that the sound absorption peaks are separated and decreased while troughs are made more widened and shallower. Also sound absorption ability does not so depend on the mean duct flow Mach number M_a , in other words, when the Mach number M_a is further increased, the change of sound absorption coefficient α is negligible. At around 700 Hz, the sound absorption coefficient α is suddenly going down, this may indicate noise is generated at this frequency.

From Figure 3-20, it can be seen that when Mach number M_a is under 0.037. The sound absorption coefficient α is change with frequency in sinusoidal form. Also the sound absorption is dramatically decreased at a specific frequency. For example, when Mach number is 0.016, the frequency is 500 Hz and when Mach number is 0.023, the frequency is 600 Hz. This indicates that the double-layer in-duct orifice plate generates noise at these mentioned frequencies. As the Machu number is increased to 0.037, sound absorption peaks are separated and decreased while troughs are made more widened and maintained at 0.65.

3.2.3 Effect of Porosity σ

The effect of porosity σ on the damping performance of the in-duct double-layer orifice plate is investigated by changing the aperture's diameter D . Two configurations are being compared. It can be seen from the Figure 3-21 that, when there is no mean duct flow, configuration with smaller porosity has large sound absorption coefficient α . However, with the mean duct flow Mach number gradually increased from 0.016 to 0.041, the sound absorption ability of the configuration with larger porosity σ gradually outfit the one of the configuration with smaller porosity σ .

Figure 3-21: Variation of measured sound absorption coefficient α with frequency for $\sigma = 0.01$, front orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm. And for $\sigma = 0.0225$, front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm. Mach number is set to (a) $M_a = 0$; (b) $M_a = 0.016$; (c) $M_a = 0.023$; (d) $M_a = 0.041$;

Generally, a single-layer in-duct orifice plate with a large diameter aperture and a thinner plate tends to have a smaller absorption coefficient and effective bandwidth [46]. Therefore, it has been found that the sound absorption property of double-layer in-duct orifice plate is quite different with the single-layer one.

When it comes to the double-layer in-duct orifice plate configurations having orifice plates with different porosity σ implemented as the front and back layers, it is meaningful to find out that whether the axial location of these perforated plates (or flow direction) will have influence on the

sound absorption ability. In Figure 3-22, it can be observed that the axial location of the orifice plates (flow direction) does not affect the sound absorption ability of the double-layer in-duct orifice plate significantly when duct flow is absent. Furthermore, when duct flow Mach number is low (in the range from 0.016 to 0.023), the one with orifice plate possessing larger porosity implemented as the front layer shows better sound absorption ability in the frequency range from 110 Hz to 450 Hz. However, configuration with orifice plate possessing smaller porosity implemented as the front layer shows better sound absorption ability over the frequency range from 450 Hz to 700 Hz. In addition, when the duct flow Mach number is high, it can be seen that orifice plate with larger porosity implemented as the front layer has overwhelming superiority. Finally, when the duct flow Mach number is set to 0.032, it can be observed from Figure 3-22(d) that the sound absorption coefficient α approaches to 1.0 as the frequency is set around 450 Hz. This indicates that the incident sound is absolutely absorbed by the double-layer in-duct orifice plate. This finding also reconfirms the conclusion made by Hughes and Dowling [45] that all the incident sound can be absorbed for certain combinations of the geometrical parameters and bias flow speed.

Figure 3-22: Variation of sound absorption coefficient α with frequency as the Mach number is set to (a) $M_a = 0$; (b) $M_a = 0.016$; (c) $M_a = 0.023$; (d) $M_a = 0.032$; (e) $M_a = 0.037$; (f) $M_a = 0.041$. For $\sigma_1 = 0.01$, $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm. For $\sigma_2 = 0.025$, $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm.

3.2.4 Effect of the Distance L_c between the Two Layers of the Plates

The effect of distance L_c between two layers of the orifice plates on the damping performance of the in-duct double-layer orifice plate is investigated in this section as shown in Figure 3-23. Four lengths are tested, 30 mm, 50 mm, 80 mm and 100 mm. It can be found out that, when the duct flow is absent, increasing the L_c has negligible effect on the sound absorption ability. However, when the duct flow is present, increasing the L_c slightly increases the sound absorption ability. This is most likely due to the viscous effect of the duct wall between two perforated plates. As the wave propagate in the duct, a small amount of energy will be dissipated by the duct wall, so in this way, the longer the duct wall the more acoustic energy will be dissipated, which leads to the slightly increase of the sound absorption coefficient α .

3.2.5 Effect of the Distance d_v of the Orifice Relative to the Plate Centre

In this section the effect of distance d_v of the orifice relative to the plate centre point is investigated. First of all, configurations, of which front orifice plate and back orifice plate have the same d_v are evaluated. The d_v is set to 6 mm, 12 mm and 16 mm one at a time. The reference is the one with $d_v = 0$ mm. It should be noted that parameters of orifice plates used in this section is the same except the d_v . Other parameters are aperture diameter $D = 6$ mm and plate thickness $T = 2$ mm. It can be seen that the effect of the d_v is negligible on the aeroacoustic damping performance of the double-layer in-duct orifice plate. In previous works, the main parameters considered are only the aperture diameter, plate thickness, geometry of the aperture, geometry of the aperture edge and grazing or bias flow Mach number while d_v is assumed to play no role on aeroacoustic damping performance. Therefore present experiment confirms that the

location of the aperture is a negligible factor. However, in the Figure 4-14 (f), the sound absorption coefficient α of the tested case does not agree so well with the reference data. It is can be explained that the d_v is so large ($d_v = 16$ mm) that the jet layer generated near

Figure 3-23: Variation of sound absorption coefficient α with frequency as Mach number is set to (a) $M_a = 0$; (b) $M_a = 0.037$; (c) $M_a = 0.041$ with front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm.

Figure 3-24: Variation of measured sound absorption coefficient α with frequency for (a) $d_{v1} = 6$ mm, $d_{v2} = 6$ mm, $M_a = 0$; (b) $d_{v1} = 6$ mm, $d_{v2} = 6$ mm, $M_a = 0.041$; (c) $d_{v1} = 12$ mm, $d_{v2} = 12$ mm, $M_a = 0$; (d) $d_{v1} = 12$ mm, $d_{v2} = 12$ mm, $M_a = 0.041$; (e) $d_{v1} = 16$ mm, $d_{v2} = 16$ mm, $M_a = 0$; (f) $d_{v1} = 16$ mm, $d_{v2} = 16$ mm, $M_a = 0.041$. For the reference setup $d_{v1} = 0$ mm, $d_{v2} = 0$ mm as $L_c = 100$ mm.

Figure 3-25: Variation of sound absorption coefficient α with frequency for (a) $d_{v1} = 6$ mm, $d_{v2} = 0$ mm, $M_a = 0$; (b) $d_{v1} = 6$ mm, $d_{v2} = 0$ mm, $M_a = 0.041$; (c) $d_{v1} = 12$ mm, $d_{v2} = 0$ mm, $M_a = 0$; (d) $d_{v1} = 12$ mm, $d_{v2} = 0$ mm, $M_a = 0.041$; (e) $d_{v1} = 16$ mm, $d_{v2} = 0$ mm, $M_a = 0$; (f) $d_{v1} = 16$ mm, $d_{v2} = 0$ mm, $M_a = 0.041$; For the reference setup $d_{v1} = 0$ mm, $d_{v2} = 0$ mm as $L_c = 100$ mm.

Figure 3-26: Variation of sound absorption coefficient α with frequency for (a) $d_{v1} = 6$ mm, $d_{v2} = 12$ mm, $M_a = 0$; (b) $d_{v1} = 6$ mm, $d_{v2} = 12$ mm, $M_a = 0.041$; (c) $d_{v1} = 12$ mm, $d_{v2} = 6$ mm, $M_a = 0$; (d) $d_{v1} = 12$ mm, $d_{v2} = 6$ mm, $M_a = 0.041$; (e) $d_{v1} = 6$ mm, $d_{v2} = 16$ mm, $M_a = 0$; (f) $d_{v1} = 6$ mm, $d_{v2} = 16$ mm, $M_a = 0.041$; For the reference setup $d_{v1} = 0$ mm, $d_{v2} = 0$ mm as $L_c = 100$ mm.

Figure 3-27: Variation of sound absorption coefficient α with frequency for (a) $d_{v1} = 16$ mm, $d_{v2} = 6$ mm, $M_a = 0$; (b) $d_{v1} = 16$ mm, $d_{v2} = 6$ mm, $M_a = 0.041$; (c) $d_{v1} = 12$ mm, $d_{v2} = 16$ mm, $M_a = 0$; (d) $d_{v1} = 12$ mm, $d_{v2} = 16$ mm, $M_a = 0.041$; (e) $d_{v1} = 16$ mm, $d_{v2} = 12$ mm, $M_a = 0$; (f) $d_{v1} = 16$ mm, $d_{v2} = 12$ mm, $M_a = 0.041$; For the reference setup $d_{v1} = 0$ mm, $d_{v2} = 0$ mm as $L_c = 100$ mm.

the orifice may interact with the duct. This cannot happen when the d_v is smaller. So when d_v is smaller, the sound absorption coefficient α of tested cases have nearly no difference with the reference one.

To make sure the d_v is a negligible factor, configurations, of which the front and back orifice plate have different d_v are also tested in this section. It can be seen that though the front and back orifice plate have different d_v , the effect of d_v is still negligible. This is most likely due to the fact that only plane waves at low frequency are considered in this work. However, in the case of single-layer orifice plate, the sound absorption capability is strongly depend on the location of the aperture due to the end correction. In present work, it is shown that the end correction is the same whether the aperture is located concentric or eccentric, this difference compared with the single layer one is mostly likely due to the special configuration of the double-layer orifice plates.

3.2.6 Comparison of Sound Absorption Performance between Single- and Double-layer In-duct Orifice Plates

In this section the sound absorption ability of single-layer and double-layer in-duct orifice plate is compared. The aim is to find out whether an addition of a layer can enhance the sound absorption ability. From Figure 3-28(a) and Figure 3-29(a), it can be observed that when mean (bias) flow is absent, three peaks are observed from the curve corresponding to the double-layer orifice plate and two peaks are observed from the curve corresponding to the single-layer orifice plate. Also it can be seen that the local maximum sound coefficients related to the double-layer orifice plate are larger than that of the single-layer orifice plate. Furthermore, sound absorption coefficients of the double-layer orifice plate and single-layer orifice plate are maintained almost at a fixed value over the frequency range from 500 Hz to 700 Hz. Generally, when there is no bias flow involved, double-layer orifice plate shows better sound absorption ability.

Figure 3-28: Variation of sound absorption coefficient α with frequency as the Mach number is set to (a) $M_a = 0$; (b) $M_a = 0.016$; (c) $M_a = 0.032$; (d) $M_a = 0.037$ for single-layer orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm and for double-layer orifice plate with front orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm.

Figure 3-29: Variation of sound absorption coefficient α with frequency as the Mach number is set to (a) $M_a = 0$; (b) $M_a = 0.016$; (c) $M_a = 0.032$; (d) $M_a = 0.037$ for single-layer orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and for double-layer orifice plate with front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm.

In Figure 3-28(b) and Figure 3-29(b), it can be observed that compared with the sound absorption coefficient of single-layer orifice plate, which is periodically changed with frequency, the sound absorption peaks of double-layer orifice plates have higher values over the frequency range before 500 Hz. However, when the frequency exceeds 500 Hz, the sound absorption coefficient of single-layer orifice plate is still periodically changed with frequency while the one of double-layer orifice plate is maintained at a low value. This may indicate that the configuration of double-layer in-duct orifice plate may have the potential to generate noise in this frequency range. In Figure 3-28(c), Figure 3-28(d) and Figure 3-29(c), Figure 3-29(d), it is can be observe that the double-layer orifice plate shows absolute advantage in the aspect of sound absorption when duct flow Mach number is high. The sound absorption coefficient of the double-layer orifice plate is almost linearly increased or decreased with the frequency as Mach number is set to $M_a = 0.037$ while the one of the single-layer orifice is still keep the sinusoidal pattern. Finally, it can be concluded that the sound absorption ability of double-layer in-duct orifice plate is better than that of the single-layer in-duct orifice plate. This is most likely due to the special configuration of the double-layer in-duct orifice plates. As shown below, when the incident wave going into the front orifice plate, the wave will be trapped inside the cavity and it will bounce with the front and back orifice plate many times until it can be transmitted into downstream, in this way more acoustic energy will be dissipated. Therefore the double-layer orifice plates are more effective than the corresponding single-layer one.

Figure 3-30: Schematic view of the energy dissipation mechanism of double-layer orifice plates

3.2.7 Effect of Ratio ζ of diameter D to orifice distance b on Single-layer Orifice Plates

The effect of Ratio ζ of diameter D to orifice distance b on the damping performance of the single-layer orifice plate is investigated in this section as shown in Figure 3-6. Four ratio ζ are tested they are 0.2, 0.4, 0.5 and 0.8. The schematic view of tested orifice plates can be found in Figure 3-2 and corresponding parameters are shown in Table 2. Larger ratio ζ means the distance between the two apertures is closer. The reference orifice plate with a single hole has the same porosity and thickness with other orifice plates. The results show that with the increase of ζ the sound absorption capability is increased, however, an exactly converse trend is appeared as the excitation frequency is beyond 350 Hz and it is also shown that the presence of the bias flow can enhance this effect. The figures also show that the resistance is the main cause for the variation of the sound absorption coefficient and the variation of the ratio ζ does not play a role on the reactance.

Figure 3-31: Variation of the (a) sound absorption coefficient α ; (b) acoustic resistance; (c) acoustic reactance with the frequency. $D = 4$ mm, $T = 2$ mm, $\sigma = 0.02$ and $M_a = 0$.

Figure 3-32: Variation of the (a) sound absorption coefficient α ; (b) acoustic resistance; (c) acoustic reactance with the frequency $D = 4$ mm, $T = 2$ mm, $\sigma = 0.02$ and $M_a = 0.015$.

3.2.8 Effect of the Orifices Distribution on Single-layer Orifice Plates

The effect of orifices distribution on the damping performance of the single-layer orifice plate is investigated in this section. Two different single-layer orifice plate distribution patterns are examined as shown in Figure 3-2 and Table 2. One is square-arrayed and the other is diamond-distributed. Each plate includes four circular orifices with a diameter of 4 mm, the side length of the square and diamond is the same. The only difference between these two plates is the distribution pattern. The results show that the one with diamond-arrayed orifices possesses better sound absorption capacity than the one with the square arrayed orifices in the whole interested frequency range and under all tested bias flow velocity conditions. The figures also show that the resistance is the main cause for the variation of the sound absorption coefficient and the variation of the ratio ζ does not play a role on the reactance.

Figure 3-33: Variation of the (a) sound absorption coefficient α ; (b) acoustic resistance; (c) acoustic reactance with the frequency. $D = 4$ mm, $T = 2$ mm, $\sigma = 0.04$ and $M_a = 0$.

Figure 3-34: Variation of the (a) sound absorption coefficient α ; (b) acoustic resistance; (c) acoustic reactance with the frequency. $D = 4$ mm, $T = 2$ mm, $\sigma = 0.04$ and $M_a = 0.015$.

Figure 3-35: Variation of the (a) sound absorption coefficient α ; (b) acoustic resistance; (c) acoustic reactance with the frequency. $D = 4$ mm, $T = 2$ mm, $\sigma = 0.04$ and $M_a = 0.029$.

Figure 3-36: Variation of the (a) sound absorption coefficient α ; (b) acoustic resistance; (c) acoustic reactance with the frequency. $D = 4$ mm, $T = 2$ mm, $\sigma = 0.04$ and $M_a = 0.044$.

CHAPTER4 Numerical Evaluation of Noise Damping Performance of Double-layer In-duct Orifice

4.1 Numerical Techniques

The governing equation is the Linearized Euler Equation (LEE), which is widely used to study acoustic noise generation and propagation [50, 73, 74]. To linearize the Euler equation, instantaneous variables including density ρ , velocity $\mathbf{u}$ and pressure p are decomposed into a mean part and fluctuating part.

$$\begin{aligned}\rho(\mathbf{x}, t) &= \bar{\rho}(\mathbf{x}) + \rho'(\mathbf{x}, t) \\ \mathbf{u}(\mathbf{x}, t) &= \bar{\mathbf{u}}(\mathbf{x}) + \mathbf{u}'(\mathbf{x}, t) \\ p(\mathbf{x}, t) &= \bar{p}(\mathbf{x}) + p'(\mathbf{x}, t)\end{aligned}\tag{4.1}$$

where $\bar{\rho}$, $\bar{\mathbf{u}}$, $\bar{p}$ are the mean part, ρ' , $\mathbf{u}'$, p' are the fluctuating part of density, velocity and pressure, respectively. Here assuming all fluctuating quantities are small compared to mean quantities as $p'/\bar{p} \sim O(1)$, $\mathbf{u}'/\bar{\mathbf{u}} \sim O(1)$ and $\rho'/\bar{\rho} \sim O(1)$. Therefore higher order fluctuating terms can be neglected. Because there is no loss mechanism within the LEE (no viscous loss and heat conduction), the instabilities can grow exponentially if the LEE is solved in time domain or in frequency domain using iterative solver. Therefore the equations are reformulated in frequency domain by assuming fluctuating variables as harmonic quantities and the direct solver is adopted.

$$\begin{aligned}\rho'(\mathbf{x}, t) &= \hat{\rho}(\mathbf{x})e^{-i\omega t} \\ \mathbf{u}'(\mathbf{x}, t) &= \hat{\mathbf{u}}(\mathbf{x})e^{-i\omega t} \\ p'(\mathbf{x}, t) &= \hat{p}(\mathbf{x})e^{-i\omega t}\end{aligned}\tag{4.2}$$

where ω is the angular frequency. Introducing these ansatz to the linearized Euler equation in two dimensions leads to

$$\begin{aligned}
-i\omega\hat{p} + \nabla \cdot (\hat{p}\bar{\mathbf{u}} + \bar{p}\hat{\mathbf{u}}) &= 0 \\
-i\omega\hat{\mathbf{u}} + (\bar{\mathbf{u}} \cdot \nabla)\hat{\mathbf{u}} + (\hat{\mathbf{u}} \cdot \nabla)\bar{\mathbf{u}} - \frac{\hat{p}}{\bar{p}^2}\nabla\bar{p} + \frac{1}{\bar{\rho}}\nabla\hat{p} &= 0
\end{aligned} \tag{4.3}$$

$$-i\omega\hat{p} + \nabla \cdot (\gamma\bar{p}\hat{\mathbf{u}} + \hat{p}\bar{\mathbf{u}}) + (1 - \gamma)[(\hat{\mathbf{u}} \cdot \nabla)\bar{p} - (\nabla \cdot \bar{\mathbf{u}})\hat{p}] = 0$$

where γ is the specific heats ratio. To omit the energy equation thereby reducing the implementation and computational cost, the relation between the pressure and density is assumed to be linear. This condition is not always true, for example, when the explosion exists in the considered problem. However, Early works [56] proved that the assumption is appropriate for the present simulation.

$$\hat{p} = \bar{c}^2\hat{\rho}, \quad \bar{c} = \sqrt{\gamma\bar{p}/\bar{\rho}} \tag{4.4}$$

where $\bar{c}$ is the speed of sound and γ is the specific heat ratio.

It is well known that the traditional Galerkin methods are unstable for convection-dominated problems [75-77], so the streamline upwind Petrov-Galerkin (SUPG) method is considered here. By introducing an artificial dissipation function to the trial space function of the Galerkin formulation, the traditional Galerkin methods are stabilized. The artificial dissipation is controlled by the stabilization parameter τ , and it is determined by the element size and convective velocity. Many researchers have put effort on determining the stabilization parameter here $\tau = 5 \times 10^{-3}$.

For simplicity the LEE can be reformulated in the following form in cylindrical coordinate:

$$-i\omega\hat{\mathbf{q}} + \frac{\partial \mathbf{A}_r \hat{\mathbf{q}}}{\partial x_r} + \mathbf{C}\hat{\mathbf{q}} = \hat{\mathbf{s}} \quad r = 1, 2 \tag{4.5}$$

where $r = 1$ is the radial direction and $r = 2$ is the vertical direction. $\hat{\mathbf{q}}$ is the time harmonic variable matrix $[\hat{p}, \hat{u}, \hat{w}, \hat{p}]^T$ and the $\hat{u}, \hat{w}$ are the velocity perturbations in radial and vertical

direction within the cylindrical coordinate. $\mathbf{A}_r$ represents the coefficient matrix and matrix $\mathbf{C}$ consists of derivatives of mean flow quantities. $\hat{\mathbf{s}}$ is a vector of time-harmonic acoustic source terms.

The computational domain Ω is divided into N non-overlapping elements Ω^e . The variable $\hat{\mathbf{q}}$ can be approximated by $\hat{\mathbf{q}}^e$ using a basis function set ψ_j :

$$\hat{\mathbf{q}} \approx \sum_{j=1}^N \hat{\mathbf{q}}^e \psi_j \quad (4.6)$$

The size of the basis set N is a function of the degree of the basis polynomials p_d and the number of spatial dimensions. The weak form of SUPG over an element for the Eq. (4.3) for can be formulated as:

$$\begin{aligned} \int_{\Omega^e} \psi \left(-i\omega \hat{\mathbf{q}} + \frac{\partial \mathbf{A}_r \hat{\mathbf{q}}}{\partial x_r} + \mathbf{C} \hat{\mathbf{q}} \right) d\Omega^e + \int_{\Omega^e} \boldsymbol{\tau}^e \mathbf{A}_j^T \frac{\partial \psi}{\partial x_j} \left(-i\omega \hat{\mathbf{q}} + \frac{\partial \mathbf{A}_r \hat{\mathbf{q}}}{\partial x_r} + \mathbf{C} \hat{\mathbf{q}} \right) d\Omega^e \\ = \int_{\Omega^e} \psi \hat{\mathbf{s}} d\Omega^e + \int_{\Omega^e} \boldsymbol{\tau}^e \mathbf{A}_j^T \frac{\partial \psi}{\partial x_j} \hat{\mathbf{s}} d\Omega^e \end{aligned} \quad (4.7)$$

Where $\boldsymbol{\tau}^e$ is the stabilization parameter matrix from [75] and it can be expressed as:

$$\boldsymbol{\tau}^e = \max \left(\frac{\alpha h_r^e}{\lambda_r} \right) \mathbf{I} \quad (4.8)$$

Here four harmonic variables are considered, so the $\mathbf{I}$ is a 4×4 identity matrix. The element size h_r^e is the width of the element in the r -th direction. λ_r is the spectral radius of the coefficient matrix $\mathbf{A}_r$. α is a constant. The problem considered is linear and only contains four variables, so the direct solver MUMPS is adapted, which is based on LU decomposition and implemented in Comsol MULTIPHYSICS v5.2.

Due to the fact that the model is built in cylindrical coordinate, no frequency scaling and geometry modification are needed. An overview of boundary conditions of flow and acoustic

field are shown in Figure 4-1. Because there is no acoustic boundary layer need to be considered in LEE, so on the surface of the orifice plate and the duct wall the slip boundary condition is applied:

$$\hat{\mathbf{u}} \cdot \hat{\mathbf{n}} = 0 \quad (4.9)$$

$\hat{\mathbf{n}}$ is the unit vector normal to the wall. For the downstream outlet the isentropic pressure boundary condition is applied:

$$p_t = 0 \quad (4.10)$$

p_t is the instantaneous pressure. At the upstream inlet boundary an incident pressure is prescribed.

The orifice plates, the upstream, downstream duct sections are meshed with free triangular as shown in Figure 4-2. Due to the property of the LEE, no acoustic boundary layers needs to be resolved, this reduces the computational cost significantly. The mesh generates 462 triangular elements, 25 quadrilateral elements, 207 edge elements and 16 vortex elements. The minimum element is 36mm, so in the worst case the wavelength and element ratio is approximately 10, which indicates for the shortest wave length (corresponding to the frequency of 1000 Hz), there is near 10 elements per wavelength.

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Figure 4-1: An overview of boundary conditions of (a) Flow Field and (b) Acoustic Field.

Figure 4-2: Overview of the mesh near the orifice plates with front orifice plate $D = 6$ mm, $T = 2$ mm and back orifice plate $D = 12$ mm, $T = 2$ mm, distance between two orifice plates $L_c = 100$ mm.

4.2 Numerical Model Validation

To evaluate and validate the present numerical model, the sound absorption coefficient α of a single-layer in-duct orifice plate is calculated using present numerical technic and compared with experimental results. Simulation was conducted in the frequency range from 110 Hz to 700 Hz. According to the experimental measurements, the frequency increment is set to 10 Hz.

As shown in Figure 4-3, the numerical results and experimental results are in excellent agreement except the discrepancies appeared at very low frequency. It is most likely due to the truncated upstream duct in present simulation. Above comparison indicates that the present numerical model can be applied to simulate the noise damping performance of in-duct orifice plates.

Figure 4-3: Variation of sound absorption coefficient α with frequency for (a) $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and $M_a = 0.041$; (b) $D = 12$ mm, $T = 2$ mm, $d_v = 0$ mm and $M_a = 0.041$, comparison between measurements and numerical results.

4.3 Results and Discussion

In order to make the calculations converges easily and considering computational cost, based on RANS (Reynolds-averaged Navier-Stocks equations) k- ϵ model is selected to calculate the bias flow. The bias flow field is assumed to be stationary (time-independence). The mean flow Mach number is the ratio of inlet flow velocity and sound speed. The background flow field in the region $-0.05 \text{ m} \leq x \leq 0.125 \text{ m}$ is shown in Figure 4-4. The Reynolds number based on the pipe diameter is 1.2×10^4 . It can be seen that there is an obvious shear layer near the orifice plates.

Figure 4-4: Mean flow field in the vicinity of the double-layer in-duct orifice plate for the front orifice plate $D = 6 \text{ mm}$, $T = 2 \text{ mm}$, $d_v = 0 \text{ mm}$ and back orifice plate $D = 12 \text{ mm}$, $T = 2 \text{ mm}$, $d_v = 0 \text{ mm}$ as $L_c = 100 \text{ mm}$. The Mach number is set to $M_a = 0.016$.

The acoustic pressure distribution in the region $-0.6 \text{ m} \leq x \leq 0.6 \text{ m}$ is shown in Figure 4-5. An obvious plane wave pattern can be identified. Also it also can be observed that the amplitude of plane wave propagating in the downstream pipe is smaller than the one of the upstream pipe, the reason is that the acoustic energy is transferred to the kinetic energy of the vorticity generated at the edge of the orifice. A close-up view of acoustic velocity in the radical and vertical direction as the solution of governing equation is displayed in Figure 4-6. Furthermore, the vorticities generated in the vicinity of the orifice is shown in Figure 4-7. It is seen that the vorticity can be convected away to the downstream as the bias flow is present, which makes increased amount of sound energy transferred to kinetic energy.

Figure 4-5: Acoustic pressure field for the front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 12$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm. The Mach number is set to $M_a = 0.016$ and $f = 700$ Hz.

Figure 4-6: Acoustic (a) radial (b) vertical velocity for the front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 12$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm. The Mach number is set to $M_a = 0.016$ and $f = 600$ Hz.

Figure 4-7: Vorticity near the front orifice for the front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 12$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm. The Mach number is set to (a) $M_a = 0.016$; (b) $M_a = 0$ as $f = 600$ Hz.

Figure 4-8: Variation of sound absorption coefficient α with frequency for the front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 12$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm. The Mach number is set to (a) $M_a = 0.016$; (b) $M_a = 0.032$; (c) $M_a = 0.041$ and (d) Variation of sound absorption coefficient α with M_a as the frequency is set to 300 Hz.

The numerical results are compared with the experimental measurements in this section. Figure 4-8 shows how absorption coefficient α varied with frequency as Mach number M_a is set to different values. It can be seen that in Figure 4-8(a), the numerical results are in excellent agreement with the measurements. However, in the Figure 4-8(b) and Figure 4-8(c) the sound absorption coefficient α is underestimated in the frequency range from 400 to 600 Hz. This is most likely due to the flow field simulation assumption that the flow field is assumed to be stationary. Also in physical aspect, it means that the mean flow contributes more kinetic energy for the generation of the vorticity. Therefore, less acoustic energy is required. In the Figure 4-8(d), the trend that the sound absorption coefficient is first increase and then decrease with the increased frequency is well reproduced by the simulation.

In Figure 4-9, it can be observed that the same observation in experimental measurements is well captured by the present simulation. In Figure 4-9(a), the predicted sound absorption coefficient α is smaller than the measured data. This may be explained that in the experiment the flow velocity is measured at the centre point in the middle section of upstream duct, then this flow velocity value is set as the inlet velocity in the simulation by assuming there is negligible flow losses in the upstream duct, as a results, the simulated velocity in front of the front orifice may be overestimated, which leads to the underestimate of the sound absorption coefficient α . In Figure 4-9(b), it can be observed that the simulation results and experimental data are in great agreement in most frequency. However, it should be noted that the partial discrepancy may be generated by the measurement errors.

Figure 4-9: Variation of sound absorption coefficient α with frequency for the front orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm. The Mach number is set to (a) $M_a = 0.016$; (b) $M_a = 0.041$.

The simulation results related to the effect of distance between two layers L_c on the sound absorption ability of double-layer in-duct orifice plate is shown in Figure 4-10 together with experimental data. It is observed that discrepancy has appeared in the low frequency range. This can be explained that the flow field is not appropriately simulated. In the present RANS, $k-\epsilon$ model, the pipe wall is set to slip condition in order to reduce the computational cost, as a result, the boundary layers are not accurately resolved. The simulation results also depicts that as the L_c is increased, the sound absorption ability is increased when the mean flow is present. Another issue can cause the discrepancy is that present code can only deal with flow borne problem, in other words, flow induced noise, which may not be avoided in experimental measurements is not considered in present simulation.

Figure 4-10: Variation of sound absorption coefficient α with frequency for the front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as (a) $L_c = 100$ mm; (b) $L_c = 30$ mm. The Mach number is set to $M_a = 0.041$.

Figure 4-11: Variation of sound absorption coefficient α with frequency (a) for the front orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm (b) for the front orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as Mach number is set to 0.016.

The simulation results of the effect of flow direction on the sound absorption ability of double-layer in-duct orifice plate are shown in Figure 4-11 together with experimental data. The trend observed in the experimental measurements that flow direction or the axial position of the double

layer orifice plate can also affect the sound absorption capacity is reasonable captured by the numerical simulation.

Figure 4-12: Variation of measured and calculated sound absorption coefficient α with frequency (a) for $\sigma = 0.01$, front orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 4$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm; (b) for $\sigma = 0.0225$, front orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm and back orifice plate $D = 6$ mm, $T = 2$ mm, $d_v = 0$ mm as $L_c = 100$ mm as Mach number $M_a = 0.041$.

The simulation results related to the effect of porosity σ on the sound absorption ability of double-layer in-duct orifice plate is shown in Figure 4-12 together with experimental data. The numerical results show a good agreement in higher frequency and a discrepancy in the low frequency. The observation in the experimental measurements is well captured. Referring the comparison of experimental measurements and numerical results before, the discrepancy in the very low frequency seems like a common problem. This is most likely due to the instability of the present scheme in the low frequency range. Another thing is that the agreement between numerical results and experimental measurements of the one of higher porosity is better than the one with lower porosity. This can be explained by the mesh property. It is well know that smaller

structure like the low porosity case need finer mesh to precisely resolve the flow field. However, in present study the mesh scheme does not change so much as the simulated perforated plate is varied.

CHAPTER5 Conclusions and Future work

5.1 Conclusions

Parametric measurements are conducted to gain insights on the noise damping performance of in-duct double- and single-layer perforated plates in the presence of a mean flow. The critical roles of 1) M_a , 2) σ , 3) flow direction, 4) L_c and 5) d_v are examined one at a time. Furthermore, the effect of ratio ζ of orifice diameter D to the orifice centre distance b and orifices distribution pattern on the damping performance are also investigated. A series of numerical simulations have been conducted to validate the experimental measurements. Based on our experimental measurements and numerical studies, the concluding remarks are summarized as:

- 1) The sound absorption coefficient α is periodically changed over the whole frequency domain when Mach number M_a is low. However, when Mach number M_a is increased, the absorption troughs are more separated and made shallower. In addition, the absorption peaks are widened as well. When the Mach number M_a is further increased, general downward trend of sound absorption coefficient α can be observed.
- 2) When there is no mean flow, i.e. $M_a = 0$, in-duct double-layer perforated plates with smaller porosity σ is associated with a larger sound absorption coefficient α . But when $M_a > 0$, these perforated plates with a smaller porosity are shown to be involved with smaller sound absorption coefficient α .
- 3) When the mean flow direction is changed, (or swapping the axial locations of these perforated plates), the noise damping performance of the double-layer plates are found to be varied.

- 4) As the connecting pipe length L_c is increased, the sound absorption coefficient is increased as mean flow is present. This is most likely due to the ‘resonance’ effect of the connecting cavity built between the neighboring perforated plates.
- 5) As the relative position of the aperture is changed, i.e. varying d_v . The perforated plate’s noise damping performance is little changed. This is most likely due to the fact that only plane waves at low frequency are considered in this work.
- 6) Double-layer perforated plates are found to be associated with a larger sound absorption coefficient than that of a single-layer perforated plate. However, the double-layer perforated plates have the potential to generate noise itself, which will weaken the noise damping ability.
- 7) Increasing ξ leads to an increase of sound absorption capacity, however, an exactly converse trend is appeared as the excitation frequency is beyond 350 Hz and it is also shown that the presence of the bias flow can enhance this effect.
- 8) The diamond orifices distribution exhibits more promising results than the one with square-arrayed orifices.
- 9) Numerical results show a reasonable agreement with the experimental measurements.

5.2 Future Work

In present work, when investigating the noise damping performance of the double layer in-duct orifice plates, orifice is assumed to work separately and it is circularly isolated from the whole perforated plate. However, many other geometry can be used to isolate the orifice, such as square, diamond and triangular. The author considers that whether the isolation geometry can be influential. This factor may be investigated in the future and not included in the Meng study.

Perforated liners are generally applied in real gas turbine and aeroengines. In those systems, the temperature is very high. To protect perforated liners, a cooling flow is applied, which is also known as grazing flow. The effect of joint-grazing and bias flow needs to be studied in the future. In addition, the operating conditions of the engine systems are varied from time to time. This indicates that the noise generated will have different intensity and frequency spectrum. Thus it is critical to study the effect of the noise amplitude and frequency on the damping performance of the perforated liners. However, due to the tight schedule and limitation of experimental facilities, these works are not included in the Meng study.

Finally, in present work, only bias flow is considered. However, in the place where the orifice plates are implemented the temperature is very high, so the effect bring by the hot flow may become important. Therefore this factor may be investigated in the future.

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