Noise Reduction of a Turbofan Bleed Valve

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A combined experimental and computational research effort investigated the noise sources of a pneumatic bleed valve used in turbofan engines and developed engineering solutions for attenuating those sources. The experimental effort employed 1/4-scale, rapid-prototyped valve designs which enabled the exploration of a large parameter space. Microphone array systems surveyed the sound and its sources, and Pitot surveys measured the mean velocity downstream of the valve. The numerical code solved the Revnolds-Averaged Navier Stokes (RANS) in both steady and unsteady modes. The research addressed the flow and acoustics of the valve without and with a muffler. The noise of the isolated valve consists of conventional jet noise with strong excess sound originating from the valve exit. The excess sound was traced to vortex shedding by support struts. Streamlining of the struts, aided by computational parametric studies, eliminated this noise component. The noise of the complete valve, with muffler attached, was attenuated by two principal means: using the streamlined struts and adding a honeycomb flow straightener to the muffler cavity. Proper installation of the honeycomb is critical for maximizing noise reduction. The resulting best design achieved a reduction of approximately 8 decibels in perceived noise level.

Nomenclature

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Symbols	
A	= cross-sectional area
D	= effective valve diameter
f	= full-scale frequency
M	= Mach number
р	= pressure
Sr	= Strouhal number
t	= strut thickness
U	= velocity
β	= porosity
θ	= polar angle
ϕ	= azimuth angle
Acronyms	
IDDV	internet dista museren hlas

IPBV	= intermediate pressure bleed valve
OASPL	= overall sound pressure level
NPR	= nozzle pressure ratio
PNL	= perceived noise level
SPL	= sound pressure level
TUT	

⁼ turbulent kinetic energy тке

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I. Introduction

A significant but little-reported noise from turbofan engines is produced by bleed valves that discharge compressor air into the bypass duct. The bleed valves operate to prevent compressor surges and typically activate during transients or throttle back. Figure 1 illustrates the installation of the valves on a typical high-bypass engine. Noise from the bleed valves is particularly troublesome at approach conditions when the engine is near idle. This paper presents a joint experimental-computational effort to understand, model, and reduce noise from pneumatic bleed valves used in aircraft engines. The baseline valve is based on the design by Parker Hannifin Corporation of an Intermediate Pressure Bleed Valve (IPBV) used on an existing commercial aircraft engine.



Fig. 1 Example installation of bleed valves on a turbofan engine (courtesy: www.sciencemuseum.org.uk).

The basic elements of the bleed valve are shown in Fig.2a. The valve body includes a centerbody, a plunger that controls the opening/closing of the valve, and three radial struts that support the centerbody. One of the struts encloses a pneumatic line for operating the plunger and is consequently thicker than the other two. The baseline muffler consists of two perforated plates, the baffle plate at the valve body exit and the dome plate a short distance downstream. The dome plate is curved, presenting a convex surface to the bypass flow, although this feature was not reproduced in the present experiments. Figure 2b shows a 1/4-scale stereolithography model of the baseline valve body used in aeroacoustic tests at U.C. Irvine (UCI). Figure 3 explains the angle nomenclature for presenting the acoustic results. The polar angle is defined with respect to the downstream axis and the azimuthal angle is defined from the direction opposite of the thick strut.



Fig. 2 Intermediate pressure bleed valve (IPBV) for gas turbine application. a) Cross section showing the main elements; b) view of exhaust side of scaled prototype.



Fig. 3 Definitions of polar (θ) and azimuthal (ϕ) angles.

In contrast with traditional jet noise suppression efforts, the constraint in this project is not thrust loss. Instead, the challenge is to achieve significant sound attenuation within the very short distance between valve inlet and exhaust into the bypass duct while preserving mass flow rate and minimizing complexity and weight.

The paper is organized as follows. Sections II and III discuss the experimental and numerical methods, respectively. The results are broken down into two major parts: treatment of the valve body alone (without muffler) in Section IV; and treatment of the complete valve (valve body plus muffler) in Section V. The paper concludes with Section VI.

II. Experimental Methods

A. Valve Design

The intermediate pressure bleed valve geometry was provided by Parker Hannifin Corporation to UCI, which adapted the design for 1/4-scale models. An adaptor was fabricated to allow connection of the valve to the UCI facility pipe fitting. The adaptor, valve body, and muffler chamber were constructed with Accura 50 Gray Polymer using high definition stereolithography (Solidconcepts, Inc.). Several variations of the valve body were tested; here we report only on the variations of the strut cross sections. Figure 4 shows two views of the baseline subscale IPBV.

The 1/4-scale valve has an effective diameter D=0.90 in, based on pressure-flow relations for the full-scale valve measured at Parker facilities. In addition to the IPBV models, we tested a round convergent nozzle with exit diameter matching the effective diameter of the IPBV (Fig.4).



Fig. 4 Photos of the ¼-scale IPBV baseline model (left and center) and of the equivalent round nozzle (right).

B. Flow Facility and Diagnostics

Experiments were conducted in the UCI Jet Aeroacoustics Facility, shown in Fig.5. Noise surveys and source location studies utilized a microphone array. The array consisted of eight 3.2-mm condenser microphones (Bruel & Kjaer, Model 4138) arranged on a circular arc centered at the vicinity of the nozzle exit. The polar aperture of the array was 30 deg and the array radius was 1 m. The angular spacing of the microphones was logarithmic. The entire array structure was rotated around its center to place the array at the desired polar angle. Positioning of the array was done remotely using a stepper motor. An electronic inclinometer displayed the position of the first microphone. The arrangement of the microphones inside the anechoic chamber, and the principal electronic components, are shown in Fig.5a. The microphones were connected, in groups of four, to two amplifier/signal conditioners with low-pass filter

set at 300 Hz and high-pass filter set at 120 kHz. The four-channel output of each amplifier was sampled at 250 kHz per channel by a multi-function data acquisition board (National Instruments PCI-6070E). Two such boards, one for each amplifier, were installed in a Pentium 4 personal computer. National Instruments LabView software was used to acquire the signals.

Sound spectra were computed with a Fast Fourier Transform (FFT) size of 4096. The sound pressure level spectra were corrected for actuator response, free-field correction, and atmospheric absorption. The overall sound pressure level (OASPL) was obtained by integrating the corrected spectrum. The perceived noise level (PNL) was calculated using industry standards. All the acoustic results presented here are scaled-up to full size and are referenced to a radius r/D=100 from the valve exit. We consider the PNL as the most relevant metric in assessing the noise of the valve, although we note that the changes in OASPL are typically very close to the changes in PNL.

Knowledge of the noise source distribution is important for understanding and modeling the sound generation process in the bleed valve. Jet noise sources vary with polar emission angle and, in the case of non-axisymmetric jets, with azimuth angle. The narrow-aperture phased array of Fig. 5 allowed differentiation among the various types of jet noise sources that have different directivity patterns (for example, noise from large-scale turbulence that radiates in the aft direction and noise from fine-scale turbulence that radiates in the broadside direction). Beamforming techniques provide the axial and frequency distribution of the noise sources for the valve. The deconvolution method of Papamoschou [1] was used for the noise source imaging.

In addition to acoustic testing, we conducted mean-velocity surveys using a 3D Pitot rake depicted in Fig.2b. The rake system consisted of five Pitot probes with 0.5-mm internal diameter tips attached to a three dimensional traverse system. The very small diameter of the probe tips allowed a fine resolution of the jet plume. The Pitot pressure was converted to Mach number under the assumption of constant static pressure (equal to the ambient value). The resulting Mach number (or velocity) field is valuable for our overall understanding of the behavior of the jet plume and for validating the computational predictions.

Tests were conducted at nozzle pressure ratios NPR=1.5, 2.0, and 2.5. In addition to the bleed valve models, a convergent round nozzle having the same flow area as the bleed valve was tested.



Fig. 5 Experimental setup: a) aeroacoustic facility; b) mean velocity measurement.

III. Numerical Methods

A. Governing Equations

The numerical code solved the Unsteady Reynolds-Averaged Navier-Stokes (URANS) equations. The continuity, momentum, and energy equations are presented in the set below.

$$\frac{\partial \overline{\rho}}{\partial t} + \frac{\partial}{\partial x_{i}} (\overline{\rho} \widetilde{u}_{i}) = 0$$

$$\frac{\partial}{\partial t} (\overline{\rho} \widetilde{u}_{i}) + \frac{\partial}{\partial x_{j}} (\overline{\rho} \widetilde{u}_{i} \widetilde{u}_{j}) = -\frac{\partial \overline{p}}{\partial x_{i}} + \frac{\partial}{\partial x_{i}} (\overline{\tau}_{ij} - \overline{\rho u_{i}^{"} u_{j}^{"}})$$

$$\frac{\partial}{\partial t} (\overline{\rho} \widetilde{E}) + \frac{\partial}{\partial x_{j}} (\overline{\rho} \widetilde{u}_{j} \widetilde{H}) = \frac{\partial}{\partial x_{j}} \left[\widetilde{u}_{i} \left(\overline{\tau}_{ij} - \overline{\rho u_{i}^{"} u_{j}^{"}} \right) \right] - \frac{\partial}{\partial x_{j}} \left(\overline{q}_{j} + \overline{\rho u_{i}^{"} h^{"}} - \overline{\tau}_{ij} u_{i}^{"} + \overline{\rho u_{j}^{"} u_{i}^{"} u_{j}^{"}} \right)$$
(1)

The overbar denotes time averaging and the tilde denotes the Favre averaging. Primes and double primes are fluctuations with respect to those averages. The commercial software FLUENT v6.3 [2] was used to solve the above equations using the flux-difference splitting method of Roe [3] and the shear-stress transport (SST) turbulence model [4].

The above equations are cast into the unsteady form, which can be used to drive the computations to a steady state by using time marching. In such a case, the averages in the equations should be regarded as long-time averages that eliminate any large scale unsteady motion of the flow field. However, if we accept that the averages may be performed over a time period that is large enough to include all the turbulence effects but smaller than the large scale unsteady motions of the "mean" flow, we may then use it to perform true unsteady computations (URANS). In such cases we interpret the time average as

$$\widetilde{u}_{i}(x,t) = \frac{1}{T} \int_{t}^{t+T} u_{i}(x,t) dt, \quad T_{1} << T << T_{2}$$
⁽²⁾

where T_1 is the turbulence fluctuation time and T_2 is the slow time that we do not want to associate with turbulence.

B. Computational Grid

Unstructured grids were generated using the GAMBIT preprocessor of FLUENT. Figure 6 shows a representative grid (coarsened for clarity) used for computing the flow near the valve body. The figure includes a solution for the flow at NPR=2.0, which illustrates the capability of the code to produce details that are extremely hard to measure experimentally. For the 3D configuration exemplified by Fig.6, typically 1.7 million grid points were used. Axisymmetric configurations associated with the muffler (to be presented in Section V) utilized approximately 0.5 million grid points.



Fig. 6 Example of computational grid and solution for NPR=2.0.

IV. Investigation of Valve Body

In this first part of the investigation we focused on the internal and external flow fields of the valve body alone. The goal was to understand and quantify the salient acoustic and flow phenomena, identify sources of sound particular to the valve, and come up with solutions for suppression of those sources. In addition, the investigation served as a validation of the computational code against experimental mean flow data.

A. Acoustic Surveys of Baseline Valve

Acoustic measurements were collected at polar angles $\theta = 30$, 45, 60, and 90 deg. Tests were run on the baseline IPBV and on a round nozzle with the same effective diameter. The IPBV tests compare very well with acoustic data from large-scale tests conducted at Parker facilities.

The comparison with the round nozzle helps us understand how much of the noise is "regular" turbulent mixing noise and how much is due to the valve features. Figures 7 and 8 plot narrowband sound pressure level (SPL) spectra for the round jet and the IPBV for all the polar angles and NPR=1.5 and 2.0, respectively. The spectra are plotted versus full-scale frequency. We observe spectral peaks at frequencies f=4.0 kHz and f=8.0 kHz for NPR of 1.5 and f=5.5 kHz and f=11 kHz for NPR=2.0. The peaks for NPR=2.0 are not as pronounced as for NPR=1.5. At NPR=2.5 the spectral peaks are diminished further.

B. Noise Source Location

The ability to image the noise sources provides an additional tool for understanding the nature of sound generation in the IPBV as compared to that of the simple round jet. In Figure 9 we plot contour maps of the intensity of the noise source as a function of axial distance, normalized by the equivalent jet diameter, and full-scale frequency. The sources correspond to a "view angle" θ =30 deg; similar results were obtained at larger view angles.

For NPR=1.5 the noise source for the round jet peaks at about x/D=5, a typical value that coincides with the end of the jet's potential core. At low frequency, the noise source of the bleed valve is similar to that of the round jet, indicating dominance by turbulent mixing noise. At high frequency, two very strong localized sources, one at f=4 kHz and the other at f=8 kHz, occur at the nozzle exit x/D=0. It is these sources that cause the spectral peaks seen in Fig.7. For NPR=2.0 the noise source of the bleed valve is very similar to that of the round jet, except for moderate localized sources at the nozzle exit. The frequencies of those sources are 5.5 kHz and 11 kHz. They correspond to the spectral peaks seen in Fig. 8.

In an effort to eliminate possible mechanisms for the localized noise sources, a version of the valve was built with very thin struts. The spectra did not exhibit any resonant tones, indicating that the struts were responsible for the excess noise seen in Figs. 7 and 8. However, the struts are structural and cannot be eliminated from the valve. There was need, therefore, to redesign the struts in such a way that the tones were suppressed while maintaining their structural integrity and functionality of the struts.



Fig. 7 SPL spectra of bleed valve (red) and equivalent round jet (green) at NPR=1.5.



Fig. 8 SPL spectra of bleed valve (red) and equivalent round jet (green) at NPR=2.0.



Fig. 9 Noise source location at view angle θ =30 deg.

C. Mean Velocity Surveys

In addition to acoustic testing, we conducted mean-velocity surveys using the 3D Pitot rake depicted in Fig.5. The Pitot pressure was converted to Mach number under the assumption of constant static pressure (equal to the ambient value). The resulting Mach number (or velocity) field is useful for our overall understanding of the behavior of the jet plume and for validating computational predictions. Tests on the baseline IPBV subscale model were conducted at NPR=1.5 and 2.0.

Figure 10 shows Mach number distributions for NPR=2.0 on several transverse planes from the valve exit. Near the valve exit, the geometry of the struts is captured in fine detail. Further downstream those details are smeared out but the flow retains a triangle-like cross section. This asymmetry persists all the way to the end of the measurement region (6.7 equivalent diameters from the jet exit). Figure 11 presents Mach number distributions on the *x*-y plane (i.e., a vertical plane through the plots of Fig. 10). The asymmetry in the near field is due to the presence of the struts. Further downstream, the flow approaches the state of a "classic" round jet. We conclude that the valve exit flow has a significant degree of asymmetry, which causes an appreciable azimuthal directivity of noise emission.



Fig. 10 Isocontours of Mach number for NPR=2.0 on several transverse planes from the valve exit.



Fig. 11 Isocontours of Mach number for NPR=2. 0 on the *x*-*y* plane of Fig. 10.

D. Validation of the Computational Code

Validation of the numerical scheme was based on the mean flow field of the plume exiting the valve. Computational predictions of the Mach number field using RANS were compared with experimental values extracted from the Pitot surveys. We examined Mach number profiles at different axial locations, as well as Mach number contours on several transverse planes from the nozzle exits. Figures 12 and 13 depict such comparisons at NPR=1.5. The computational results are in general agreement with experimental data. The effects of the support struts can be seen near the valve exit in Fig. 13. Further downstream, the effects of the struts are smeared out but the flow retains a triangle-like cross section. The same degree of agreement is observed for NPR=2.0.



Fig. 12 Comparison between experimental and computational Mach number profiles of jet plume exiting the valve. NPR=1.5.



Fig. 13 Comparison between experimental and computational Mach number contours on several transverse planes from the valve exit. NPR=1.5.

E. Flow Field Around Support Struts

As discussed in Section IV.B, the excess noise of the valve was connected to the support struts. To gain further insight into the noise sources related to the fluid dynamics of the struts, we conducted a time-resolved numerical investigation of this problem using the URANS code. To constrain the computational power needed, the struts were modeled as two-dimensional cylinders in a "freestream" that matched the local inflow conditions in the actual valve. The upper and lower boundary conditions were specified as periodic. An example simulation for the thick strut is presented in Fig. 14 for NPR=1.5. In this case, the "freestream" Mach number is 0.7. There is clear evidence of a shock wave formed near the leading edge, and a vortex-shedding pattern in the wake of the strut. The thin strut also experiences similar flow instability.

We surmise that the resonant behavior of the flow is associated with a transonic feedback process between the wake instability and the shock phenomena over the strut. We expect this resonance to abate as the NPR (Mach number) increases and upstream communication is cut off. Indeed, this is the trend in the acoustic spectra presented in Section IV.A.



Fig. 14 Mach number contours of flow over the thick strut at NPR=1.5.

Figure 15 plots the power spectra of the fluctuation of lift for the thick and thin struts at NPR=1.5. The tonal content of the vortex shedding is evident. For the thick strut, vortex shedding occurs at 4 kHz. The shedding frequency for the thin strut is roughly double at 7.5 kHz. These results match very well the acoustic tones of Fig. 7. Similar agreement occurs for NPR=2.0, where the shedding frequencies are moderately higher. We noted that by increasing the pressure ratio the resonant frequency increases but the magnitude of oscillations decreases, which is again in agreement with the experiments.

It is instructive to cast these frequencies in terms of Strouhal number based on the inlet velocity and the strut thickness, Sr=ft/U. Table 1 shows the frequencies and Strouhal numbers of the strut instabilities. The Reynolds based on strut thickness is 60,000 for NPR=1.5 and 90,000 for NPR=2.0. The instability of both the thick and thin struts occurs at a Strouhal number *Sr*=0.275 for NPR=1.5 and *Sr*=0.33 for NPR=2.0, values consistent with the vortex shedding of a circular cylinder at the respective Reynolds numbers [5].

To summarize, the URANS computation provides strong evidence that the spectral tones come from vortex shedding amplified by transonic resonance - the feedback between wake and shock phenomena over the strut. Therefore we expect noise from the blunt-shaped struts to be most pronounced in a transonic environment, with the inflow Mach number in the neighborhood of 0.5-0.8. This is relevant to the operation of the valve with muffler attached, to be presented in Section V.



Fig. 15 Spectra of lift coefficient fluctuations at NPR=1.5. a) Thick strut; b) thin strut.

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	Inflow velocity, m/s		Shedding Frequency, Hz		Strouhal Number				
	NPR=1.5	NPR=2.0	NPR=1.5	NPR=2.0	NPR=1.5	NPR=2.0			
Thin strut	170	200	7300	10100	0.275	0.323			
Thick strut	170	190	3700	4900	0.277	0.329			

 Table 1
 Shedding characteristics for the thin and thick struts

F. Reshaping of the Struts

After observing that vortex shedding was the likely cause of excess noise generation, we proceeded with redesigning the struts to make the cross-section more aerodynamic while meeting structural and utility constraints. Because the problem is unsteady, using any regular optimization method like adjoint method or genetic algorithm is very computationally intensive. This prompted us to a trial and error approach which can significantly reduce the amount of computations. About twenty different profiles were tested numerically. Figure 16 shows a subset of those profiles, including the more promising ones. Figure 17 illustrates the numerical suppression of vortex shedding by streamlining the strut cross-section.

The more promising struts were tested experimentally. Figure 18 shows the effectiveness of the new design in removing the excess noise. The tones are completely eliminated, as is the excess broadband noise around the tones. The resulting spectra are very close to the spectra of a round turbulent jet at the same nozzle pressure ratio.



Fig. 16 Different designs for the strut profile.



Fig. 17 Suppression of vortex shedding by reshaping the strut. Left: original cross section; right: streamlined cross section.



Fig. 18 SPL spectra of the valve body with original (red) and streamlined (blue) struts.

V. Investigation of Complete Valve

Having understood and suppressed the major excess noise sources of the IPBV valve body, we proceeded with investigation of the acoustics and flow field of the complete valve, encompassing the valve body and the muffler. In subscale (UCI) and full-scale (Parker) tests, the muffler has been shown to reduce PNL levels around 10-15 dB. Reducing jet noise by such amount using alternative techniques is impossible, therefore the muffler is an essential element of the valve design. The muffler also causes a mass flow rate reduction of around 20%.

The installation of the muffler (Fig. 2) further complicates the flow field and expands tremendously the variables of the problem. The experimental investigation covered a very large parameter space; here we report only on the key parameters that led to the quiet design of this investigation. This does not necessarily mean that other parameters may not play a role in alternative quiet designs.

A. One-Dimensional Model of Muffler

Before proceeding with detailed experimental and computational investigations, it is important to have a basic understanding of the fluid dynamics with the muffler installed, and particularly how it changes the internal environment of the valve. For this purpose, a one-dimensional analytical model of the muffler was developed. The model and its approximations are illustrated in Fig. 19.



Fig. 19 One-dimensional model of muffler.

Referring to the station numbering of Fig. 19, the modeling of the flow was as follows:

- The processes from 0 to 1, 2 to 3, and 4 to 5 were treated as isentropic. The open areas A_3 and A_5 were multiplied by a discharge coefficient of 0.85, based on experiments on perforated plates [6].
- The processes from 1 to 2 and 3 to 4 were treated with 1D, frictionless and adiabatic conservation equations.
- The exit process 5 to 6 was treated with the assumptions $p_5=p_a$, $A_6=A_4$, and preservation of axial momentum.

The resulting system of equations was solved using the Mach number M_5 as an independent variable. M_5 was increased from a low value until the nozzle pressure ratio NPR= p_0/p_a matched the experimental value. Results for the total-pressure, static-pressure, and Mach-number distributions are shown in Fig. 20 for plate porosities close to the baseline design and for NPR=2.0. Station "1n" refers to the nominal conditions for the isolated value body.



As far as the internal fluid dynamics are concerned, the flow resistance created by the muffler increases the internal pressure, as is evident in Fig. 20. Therefore, the internal nozzle pressure ratio is less than when the valve body exhausts directly into the atmosphere. For plate porosities near those of the baseline design, the internal NPR ranges from 1.2 to 1.3 and is fairly insensitive to the overall nozzle pressure ratio. The drop in NPR is evident in the reduction of the Mach number in Station 1 (exit of valve body) compared to its unrestricted value. The upshot is

that we expect internal Mach numbers near the struts to be around 0.6. This is still high enough to cause transonic flow over a blunt-shaped strut and the resulting resonance covered in Section IV.

Another useful result is that the fully-mixed exit Mach number, M_6 , is very low compared to the exit Mach number of the valve without muffler (0.38 versus 1.05). Assuming the dimensional scaling of sound intensity as M^8 proposed by Lighthill [7] the jet noise level associated with M_6 is insignificant, some 30 dB below the jet noise level of the isolated valve. Of course, in reality we need to consider the noise from the individual small jets coming from the perforations of the dome plate, whose Mach number is not small ($M_5 \sim 0.8$). However, the frequency content of those jets is very high and the sound would be rapidly attenuated by atmospheric absorption. We should also keep in mind that the model is purely 1D, whereas the exhaust flow could be non-uniform. Nevertheless, there is a strong suggestion that we need to pay a lot of attention to noise sources *internal* to the muffler.

B. Experimental and Computational Approaches

Figure 21 shows the experimental simulation of the complete valve. The muffler had an internal diameter of 2.0 in and its length was set originally at 0.51 in. Later designs evaluated a longer 1.0-in muffler. The perforated plate components of the muffler assembly were simulated using commercially available perforated sheet metal. Pressure ports were drilled at two positions: one immediately upstream of the baffle plate and another immediately upstream of the dome plate (Figure 21) to measure the static pressure at these locations. As with the valve body, the complete valve was tested at NPR=1.5, 2.0, and 2.5. The diagnostics were the same as described in Section II.

Because of the great complexity of the muffler, the computations were purely axisymmetric, treating each perforation as an annulus. In addition, all the computations reported here are steady RANS. Despite these limitations, the computations provide valuable information and insight as to the fluid phenomena inside and outside the muffler, a feat that is virtually impossible using experimental methods. The capability of the computations is illustrated by the Mach number and turbulent kinetic energy (TKE) results of Fig. 22. The TKE field is important in this investigation because it provides a measure of the noise production in the internal and external flows, as evidenced by acoustic analogy models (e.g., Ref. 8).





Fig. 21 Experimental simulation of muffler.



Fig. 22 Examples of computational results with muffler installed. a) Mach number field; b) TKE field.

C. Effects of Porosity and Strut Streamlining

The experiments covered a very large parameter space including hole size, plate thickness, plate porosity (including non-uniform distributions of porosity), and characteristics of the additional devices used to quiet the valve. With regard to the perforation parameters, the experiments indicated that porosity was the most important variable. Hole size, hole pattern, and plate thickness, all within reasonable ranges, produced moderate effects on the noise emission of the valve.

Even though the dominance of the porosity on acoustics might seem obvious as it controls the open area, the relationship is actually more complex. In the baseline UCI muffler design, the porosities of the baffle and dome plates were 0.37 and 0.22, respectively (this design approximates the porosities of the actual muffler, which are slightly different). Given that the full cross-sectional area of the muffler (A_2 or A_4 in Fig. 19) is about five times larger than the flow area of the valve, this means that the dome acts as the main flow restrictor. However, moderate noise reduction was achieved by reducing the porosity of the baffle plate. Experiments discussed in Section V.G indicate that the baneficial effect of reducing the baffle plate porosity is related to a more uniform flow distribution, and thus smaller gradients, between the baffle and dome plates. A baffle plate with little restriction will allow the jet flow to penetrate into the muffler section without much spreading. Increasing the pressure drop across the plate by decreasing its porosity forces the jet flow to spread out as it hits the first plate, and thus flow more uniformly inside the muffler section. The spreading of the jet flow is evident in the Mach number contours of Fig. 22.

An additional, very important factor in the noise of the muffler is the design of the support struts of the valve body. Recall that streamlining of the strut cross section eliminated resonant tones induced by vortex shedding. Even though the Mach number environment of the struts decreases with installation of the muffler, the Mach number is still large enough (around 0.6) to support transonic resonance if the struts are blunt-shaped. We expect internal noise sources (turbulence, vortex shedding) to dominate the muffler noise, so the strut redesign may still be relevant.

The effects of the strut redesign and the reduced baffle-plate porosity on perceived noise level are illustrated in the acoustic results of Fig. 23. Streamlining of the struts has a major impact on the noise, with reduction up to 5 dB in the azimuthal direction $\phi=0^{\circ}$. Reduction in the direction of the thin strut, $\phi=60^{\circ}$, is modest at 1-2 dB. Decreasing the baffle-plate porosity from 0.37 to 0.22 adds 1-2 dB in noise reduction in the direction of the thin strut, with minimal changes in the noise reduction at $\phi=0^{\circ}$. Note that the final design has a baffle-plate porosity of 0.30, slightly smaller than the original value of 0.37.



Fig. 23 Effects of streamlined struts and reduced baffle-plate porosity on perceived noise level. NPR=2.0.

D. Honeycomb Flow Straightener

The noise reduction results of Fig.23 represented close to the best acoustic performance we could achieve after an exhaustive experimentation with variable porosities, hole sizes, and thicknesses for the baffle and dome plates. Driven by the consideration to reduce internal turbulent and acoustic fluctuations as much as possible, it was decided to evaluate a honeycomb flow straightener as an element of the design. The honeycomb was made of aluminum and its cell size ranged from 1/16 in. to 1/4 in. (corresponding to 1/4 in. to 1.0 in. full-scale). In the initial trials, the honeycomb was sandwiched between the dome and baffle plates as illustrated by the insert of Fig. 24.

The results of the initial trials were promising, as demonstrated by the SPL and PNL plots of Fig. 24. Particularly encouraging was the fact that the honeycomb reduced noise in the direction $\phi=60^{\circ}$, which was not attenuated substantially before. Given this promise, the computations also incorporated a "honeycomb" in the form of annular chambers, with very fine thickness, between the baffle and dome plates. Initial numerical results, shown in Fig. 25, showed that the honeycomb induces a more orderly flow structure inside the muffler with a general decline of internal TKE levels.



Fig. 24 Effect of honeycomb flow straightener on acoustics of complete valve, with baseline muffler and streamlined struts. NPR=2.0.



Fig. 25 Numerical results on the effect of honeycomb on the streamline patterns and TKE field at NPR=2.0. a) Without honeycomb; b) with honeycomb.

E. Refinement of Honeycomb Installation

The encouraging initial trials with honeycomb flow straighteners triggered another extensive investigation, this time in terms of the honeycomb parameters and installation. The first parameter studied in the design of Fig. 24 was the honeycomb cell size. There was a distinct benefit in reducing the cell size from 1/4 in. to 1/8 in., and minimal changes by reducing it further to 1/16 in.

The main design variations in the installation of the honeycomb are illustrated in Fig. 26. It was quickly realized that the configuration in the initial trials, with the honeycomb length occupying the entire muffler section, was not ideal. For reasons that will be explained shortly, it is important to leave a space between the upstream end of the honeycomb and the baffle plate. Less important, but still valuable, is a space downstream of the honeycomb. The desirability of these spacing led to a doubling of the muffler length from 0.5 in to 1.0 in. This length is still deemed practically feasible and it would be roughly equal to the maximum length of the actual IPBV muffler with the curved dome plate (Fig.2).

In Fig. 26a the honeycomb spans the entire space between the baffle and dome plates. While this arrangement provides the maximum length of honeycomb, it leads to strong non-uniformity of the flow field because the flow is ducted into the honeycomb immediately past the baffle plate. Doubling the honeycomb length (by doubling the length of the muffler), Fig. 26b, did not provide significant benefit. Likewise, back-to-back assemblies of honeycombs with different cell sizes, Fig. 26c, proved ineffective. In Fig. 26d, a space is provided between the baffle plate and the honeycomb. Numerical predictions and experimental mean velocity profiles, to be covered in Section V.G, indicate that this space allows the flow to spread between the baffle plate and the honeycomb, thus improving the uniformity of the exit flow. Figure 26e depicts a variation of this arrangement with an additional perforated plate, referred to as the "middle plate", on the upstream face of the honeycomb. The middle plate further enhances the spreading of the flow before it enters the honeycomb and provides structural support for the honeycomb. In Figure 26f, the honeycomb is installed such as to allow space between the baffle plate and the dome plate. Acoustic experiments indicated that this type of arrangement provides superior noise reduction, the aft space enabling whatever fluctuations emerge from the honeycomb to die down before the dome plate.



Fig. 26 Some of the honeycomb design variations tested. a) full length in short muffler; b) full length in long muffler; c) full length with dissimilar cell sizes; d) partial length with gap upstream; e) partial length with gap upstream and middle plate; f) partial length with gaps upstream and downstream, and middle plate.

The RANS computations provided strong guidance in understanding the benefits and drawbacks of the various honeycomb installations. Figure 27 compares TKE fields of four different configurations. In the first configuration, Fig. 27a, the muffler is devoid of honeycomb. We notice the chaotic motions inside the muffler with strong levels of TKE. Installation of a full-length honeycomb, Fig. 27b, regularizes the flow and confines the peak TKE levels within the honeycomb cells. Adding a gap upstream of the honeycomb, Fig. 27c, allows the flow to spread before becoming straightened by the honeycomb. Finally, allowing a gap downstream of the honeycomb, Fig. 27d, allows a rapid decay of the TKE before the dome plate. This last configuration has the least level of integrated TKE inside and outside the muffler.



Fig. 27 Numerical results on TKE field inside and outside muffler. a) Muffler without honeycomb; b) short muffler with full-length honeycomb; c) long muffler with half-length honeycomb against dome plate; d) long muffler with half-length honeycomb installed in the middle.

F. Best Design and its Acoustic Performance

The recommendation for the muffler design comes after extensive parametric studies involving perforated plates of various porosities, hole diameters, and thicknesses; honeycomb flow straighteners of varying cell size and installation; and different lengths of the muffler chamber. In brief, the best valve design of this investigation is represented by Fig. 28 and has the following properties:

- Streamlined struts.
- Porosities in the neighborhood of 0.30 and 0.22 for the baffle and dome plates, respectively.
- Honeycomb of length approximately half of the muffler chamber length, allowing gaps between the baffle and dome plates.
- A middle plate with porosity of around 0.22 on the upstream face of the honeycomb.
- Honeycomb cell size preferably 1/4 in. or less.

The acoustic performance of the best design is described in Figs. 29-31. Figures 29 and 30 plot SPL spectra, OASPL, and PNL for ϕ =180° and for NPR=1.5 and 2.5, respectively. We note the very substantial reductions, on the order of 10 dB, in all the noise metrics. Some "hay-stacking" of the spectra occurs at large polar angles, particularly at low NPR, but this did not impact the overall acoustic benefit. A comprehensive assessment of the acoustic performance is possible by plotting the peak level of PNL versus azimuthal angle in Fig. 31. As seen in the plots emission in the general direction of the thick strut (ϕ =180°) is quietest overall. Emission at ϕ =0° (direction opposite the thick strut) is suppressed the most, however there is a moderate peak at ϕ =30°. Optimal orientation inside the fan duct may exploit this directivity of the noise of the redesigned valve.

Azimuthally-averaged assessment of the reduction in peak PNL, for each NPR, is obtained by averaging the "intensities" (10^{0.1 PNL}) corresponding to the decibel levels plotted in Fig. 32 converting to decibels, and subtracting the baseline from the best design. Accordingly, the average reductions are as follows: 8.4 dB for NPR=1.5; 8.3 dB for NPR=2.0; and 7.9 dB for NPR=2.5. In summary, the design modifications resulted in a valve that is about 8 dB quieter than the baseline.



Fig. 28 Key elements of the best design arising from the parametric study.



Fig. 29 Comparison of acoustics of best design versus baseline at NPR=1.5.



Fig. 30 Comparison of acoustics of best design versus baseline at NPR=2.5.



Fig. 31 Azimuthal directivity of peak PNL for baseline valve (red) and best design (blue).

G. Flow Field and Mass Flow Rate

In addition to the acoustic tests, we conducted Pitot rake surveys of the pressure field downstream of the muffler, from which velocity profiles were inferred. The purpose of the mean-flow surveys was to enhance our understanding of the noise suppression of the various muffler designs and to quantify the corresponding mass flow rate. The survey and processing methods were the same as those used for the valve body, Section IV.

Mean velocity data were measured for a large variety of mufflers. Here we examine the baseline muffler (Fig. 32) and two characteristic design variations: one with the honeycomb occupying the entire muffler chamber (Fig. 33) and one with the honeycomb occupying only the aft end of the muffler (Fig. 34). Contour plots of the velocity are presented in several transverse planes downstream of the dome plate. At the first axial station, the Pitot probe tip was within 0.01 in of the dome plate. The survey was so fine that details of the perforations are evident in all the figures at x=0.00 in. Concerning the baseline muffler, Fig. 32, the flow is fairly uniform except near the middle where a small velocity "spike" occurs. Installation of a full-length honeycomb causes strong non-uniformity in the flow at all axial stations. As noted earlier, the full-length honeycomb "captures" the non-uniform jet flow from the valve body and does not allow it to mix. By allowing a gap between the baffle plate and the honeycomb, flow spreads before entering the honeycomb and the uniformity of the flow is restored and is in fact slightly better than that of the baseline muffler.

Cross-sectional integration of the results exemplified by Figs. 32-34 yielded an assessment of the mass flow rate versus axial distance from the dome plate. The mass flow rate is computed under the assumption that the static pressure in the plume exiting the dome plate is uniform and equal to the ambient pressure. This assumption is dubious right at the dome plate, but is expected to hold well a short distance downstream. Further, measurements right at the dome plate are expected to be affected by the finite spatial resolution of the Pitot probes. Figure 35 shows the mass flow rate versus axial distance for the baseline configuration and the best design. In both instances, the mass flow rate starts from an unrealistically high value (due to the inaccuracies mentioned above), settles to a minimum, then gradually rises. The rise has to do with entrainment of ambient air into the jet. We consider the minimum value as a reasonable assessment of the actual mass flow rate through the valve system. Comparing the best and baseline configurations we note a modest decrease in the mass flow rate, around 4%, for the best case.



Fig. 32 Cross-sectional contours of mean velocity, normalized by local maximum value, for valve and muffler with honeycomb occupying the entire chamber.



Fig. 33 Cross-sectional contours of mean velocity, normalized by local maximum value, for valve and muffler with honeycomb occupying the entire chamber.



Fig. 34 Cross-sectional contours of mean velocity, normalized by local maximum value, for valve and muffler with honeycomb occupying the aft half of the chamber.



Fig. 35 Mass flow rate, inferred from Pitot surveys, versus axial distance for baseline and best configurations.

VI. Conclusions

A combined experimental and computational research effort addressed the noise sources of pneumatic bleed valves in turbofan engines and developed engineering solutions for reducing their sound. The experimental effort employed 1/4-scale, rapid-prototyped valve designs which enabled the exploration of a large parameter space. Microphone array systems surveyed the sound and generated noise source maps. The numerical code solved the Reynolds-Averaged Navier Stokes (RANS) in both steady and unsteady modes; it produced results that shed insight into the physics of the flow field and helped in arriving at quieter designs.

The research addressed the flow and acoustics of the valve without and with a muffler. The noise of the isolated valve consists of conventional jet noise and strong excess noise originating from the valve exit. The excess noise was traced to vortex shedding by support struts. Streamlining of the struts, aided by computational parametric studies, eliminated this noise component.

The noise of the complete valve, with muffler attached, was attenuated by two principal means: using the streamlined struts and adding a honeycomb flow straightener to the muffler cavity. Proper installation of the honeycomb is critical for achieving the maximum noise reduction. Additional, moderate benefits were achieved by reducing the porosity of the upstream perforation of the muffler. Computational results indicate that the optimal design has the least production of turbulent kinetic energy. The resulting best design achieved a reduction of approximately 8 decibels in perceived noise level compared to the baseline valve/muffler configuration.

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