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P.11

NASA Technical Memorandum 105736

A Concept for a Counterrotating Fan With Reduced Tone Noise

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August 1992

(NASA-TM-105736) A CONCEPT FOR A COUNTERROTATING FAN WITH REDUCED TONE NOISE (NASA) 11 p

N93-11370

Unclas

G3/71 0123608

NASA

A CONCEPT FOR A COUNTERROTATING FAN WITH REDUCED TONE NOISE

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E-7138

SUMMARY

As subsonic jet engine designs incorporate higher bypass ratios to reduce jet noise and increase engine cycle efficiency, the fan noise becomes a significant part of the perceived total noise. The conventional method of reducing fan tone noise is to design a low tip-speed device. An alternative approach of using a counterrotating fan with a high number of rotor blades is investigated in this report. The source of noise at the blade passing frequency of this device is the rotor-only mechanism, which is cut off for a subsonic tip speed rotor. The first interaction tone occurs at twice the blade passing frequency, which, for this fan, was shifted high enough in frequency to be above the perceived noise rating range. The result was a counterrotating fan which had more potential for tone noise reduction than does the conventional fan. A potential broadband noise reduction was also indicated.

INTRODUCTION

To reduce jet noise and increase cycle efficiency, subsonic jet engines are being designed with higher bypass ratios. As a result, the fan noise is becoming the most important part of the total engine perceived noise level. The common method for reducing fan tone noise is to design a fan with a low tip speed. Such a fan would have a low number of rotor blades and a high number of stator vanes to achieve cutoff conditions for the blade passing tone noise from the rotor-stator interaction. The cutoff condition is one where the sound pressure level of a tone decays rapidly as it passes down the duct (ref. 1).

Some alternative tone-reduction ideas were investigated in a previous work (ref. 2). The use of a fan with a high number of rotor blades to reduce the perceived noise level was investigated. The perceived noise level is a subjective scale developed to measure the annoyance of a sound (ref. 3). It represents not only the intensity of the sound, but also its frequency content. The high number of rotor blades generates higher frequency tones that are not weighted as strongly by the perceived noise scale, and this results in a noise reduction.

In reference 2, the noise reduction potentials of the conventional low-speed design and a high-blade-number design were evaluated by starting with a baseline fan. The baseline fan had 53 rotor blades and 90 stator vanes, which resulted in a blade passing tone at 3000 Hz, and harmonic tones at 6000 and 9000 Hz, all of which are included in the perceived noise rating. In reference 2, the high-blade-number design was shown to have as much noise reduction potential as the conventional low-speed design. For practical limitations, the highest blade number considered in reference 2 was a fan with 106 rotor blades. This gave a blade passing tone at 6000 Hz. The perceived noise rating scale does not rate noise at frequencies higher than the 10 000-Hz, one-third octave band. This is partly because of the limits of human hearing and partly because of the large atmospheric attenuations at higher frequencies. So, for this 106-blade fan, only the blade passing tone at 6000 Hz contributed to the rating.

If the blade passing tone of the fan could be moved to a frequency higher than the 10 000-Hz, one-third octave band (i.e., higher than 11 225 Hz), then none of the blade passing and harmonic tone noise would be rated. One method of doing this might be to increase the blade number to shift the frequency. This would require a fan with over 200 blades, which would be very difficult, if not impossible, to build.

The other approach might be to increase the tip speed. A tip speed high enough to shift the frequency would result in unacceptable blade stresses and would bring in many multiple pure tones at frequencies below the blade passing frequency. These multiple pure tones would be included in the perceived noise rating, thus defeating the purpose of the concept of shifting the frequency. The high-tip-speed device would then not be a way to shift the tones beyond the rating range.

The additional alternative fan noise reduction concept investigated in this report is the use of a counterrotating fan design to effectively accomplish the frequency shift. The tone noise generated by a fan in clean inflow can be broken into two parts: the rotor-only noise and the noise from the interaction of the two blade rows. The rotor-only noise is generated at the blade passing frequency of each rotor. The first interaction tone is generated at a frequency which is the sum of the two rotor blade passing frequencies. If both rotors had the same blade passing frequency, "X," then the interaction tone would occur at a frequency of "2X." Using the proper number of blades in each rotor, the interaction tone can then be shifted above the perceived noise rating range.

The rotor-only noise is generated by the rotating pressure pattern of the rotor, and it rotates at the rotor rotational speed. The interaction noise is generated by the interaction of the wakes and vortices of the upstream row striking the downstream row and by the potential flow interactions between the two rows. These patterns rotate at various multiples of the fan rotational speed. The patterns which rotate at speeds greater than the speed of sound propagate through the duct without decay and are called cuton tones. (see refs. 1 and 4). Those which rotate slower than the speed of sound are cut off and decay rapidly. A rotor only tone, which rotates at the rotor rotational speed, is therefore cut off for a subsonic tip speed fan.

The concept here is to have both rotors of a counterrotating fan turn at a subsonic tip speed, so that the blade passing tones, which are in the perceived noise rating range, would be cut off. Then, with a high enough number of blades, the interaction tone at twice the blade passing frequency would be shifted beyond the perceived noise rating range. Such a counterrotating fan design would have a significantly lower rated tone noise. As indicated previously in reference 2, building a fan with a high number of rotor blades can result in many design complexities. These problems are recognized, but the acoustic benefits of such a design may be sufficient to justify these complications. This report evaluates the potential noise reduction of such a counterrotating design with a high number of blades by comparing it with the baseline fan described in reference 2.

METHOD OF EVALUATION

The perceived noise levels for the fan tones are calculated using the table of noys from reference 4. The perceived noise level (PNL) is defined as

$$\mathrm{PNL} = 40 + 33.22\log_{10}\underline{N}$$

where

$$\underline{N} = n_{max} + 0.15\left(\sum n - n_{max}\right)$$

and where n_{max} is the maximum noy value, and $\sum n$ is the sum of the noy values in all of the bands. The noy values are obtained for each one-third octave band that contains a tone. When tones occur at frequencies between two one-third octave bands, the tonal energy is split equally between them. Tones occurring above the 10 000-Hz band are not considered in perceived noise level calculations. No specific tone corrections are made here for the additional perceived noisiness of tones as opposed to broadband noise.

RESULTS AND DISCUSSION

Baseline Fan

The baseline fan used for these comparisons is the same as the one used in reference 2 and is loosely based on the QF-1 fan previously tested at the NASA Lewis Research Center (ref. 5). This fan had a diameter of 1.83 m (6 ft), with 53 rotor blades and 90 stator vanes, so it was cut on. It had a pressure ratio of 1.5, with a rotor tip speed of 335 m/sec (1100 ft/sec). The stator vane chord was 6.78 cm (2.67 in.), and the inlet Mach number was assumed to be 0.3. The blade passing frequency of this fan at design speed was approximately 3000 Hz. A sketch of this fan configuration is shown in figure 1(a).

For purposes of obtaining relative perceived noise numbers, it is assumed that, at the maximum noise location, the blade passing tone sound pressure level is 100 dB. The tone levels of the baseline fan are assumed to decrease by 3 dB per harmonic, as indicated in figure 8 of reference 6. The baseline fan then has a 100-dB tone at 3000 Hz, a 97-dB tone at 6000 Hz, and a 94-dB tone at 9000 Hz. Tones at and above 11 225 Hz are not rated in the perceived noise calculation. The tone at 9000 Hz is assumed to split equally between the 8000-Hz and 10 000-Hz one-third octave bands. The resulting perceived noise level is 113.4 PNdB. The predicted reductions for the counterrotating fan are then in reference to this baseline fan. A summary of these noise levels is given in table I.

Counterrotating Fan

The counterrotating fan (fig. 1(b)) is assumed to have the same 1.83-m (6-ft) diameter and have the same 1.5 overall pressure ratio as the baseline fan. The comparison is then based on equal thrust for the two devices. To ensure that the rotor-only tone is cut off, the rotor tip speed for both rotors is reduced slightly to 320 m/sec (1050 ft/sec) but no noise reduction is taken for this speed reduction. The counterrotating fan is assumed to have 106 blades in each rotor, so the blade passing frequency is 5727 Hz, and twice the blade passing frequency is 11 454 Hz. (The possible problems of designing a rotor with 106 blades are discussed in ref. 2.) The tone at 5727 Hz is assumed to be totally in the 6300-Hz band, and the tone at 11 454 Hz is above the 10 000-Hz band, so it is not rated.

In order to rate the counterrotating fan, some assumptions have to be made concerning the relative source strengths. For the baseline fan, the level at the fundamental frequency is the sum of the rotor-only noise and the rotor-stator interaction noise. Since the rotor-only noise is cut off, the total noise in the far field is dominated by the rotor-stator interaction noise. For the counterrotating fan, the level at the fundamental frequency is due solely to rotor-only noise. For purposes of comparison it will be assumed that the level at the blade passing frequency, if it were not cut off, would be equal to the total level of the baseline fan (i.e., 100 dB). This is equivalent to assuming that the generated rotor-only noise of the counterrotating fan equals the rotor-stator interaction noise of the baseline fan. This is a very conservative assumption, because the generated rotor-stator interaction noise is expected to be larger than the rotor-only noise, and the rotor-only noise of a 106-blade fan would be less than that for a 53-blade fan. (The two 106-blade rotors of the counterrotating fan are assumed to produce the same noise as a single 106-blade fan would produce, because each rotor is doing only one-half of the work.) A short discussion of the relative strengths of rotor-only noise and rotor-stator interaction noise is included in the appendix.

The interaction tones can come from the interaction of the two rotors or the interaction with the support strut. For this evaluation, it is assumed that the support strut is located far enough downstream so that its interaction noise level can be neglected with respect to the rotor-only noise generated by the two rotors (i.e., it is approximately 20 dB lower than the generated rotor-only noise).

For the counterrotating fan, the interaction tone frequency is shifted to twice the blade passing frequency, which leaves the rotor-only noise as the only source remaining at blade passing frequency. The generated rotor-only noise, which is assumed to be equal to the level of the blade passing frequency tone of the baseline fan, is cut off in the duct and reduced in passing to the far field. If the practical effect of cutoff is assumed to be 8 dB, then the counterrotating fan would have a blade passing tone of 92 dB at 5727 Hz. (This is the 100 db of the baseline fan tone which was cut on reduced by 8 dB, the expected effect of cutoff.) The interaction tone of the counterrotating fan could be larger than that for the baseline fan as a result of the larger relative velocities. On the other hand, since the upstream rotor is only doing half the work, the wakes and vortices would be smaller, and the lift flucuations on the downstream rotor would be less. Regardless of a larger or smaller interaction tone, the frequency is above the rated range, so no additional perceived noise is observed.

A calculation for the 92-dB tone at 5727 Hz yields a noy value from reference 4 of 67.2 and results in a perceived noise level of 100.7 dB. This is the value shown in table I for the counterrotating fan with the tone reduced 8 dB by cutoff. Even with the conservatively high tone level, this is a reduction of 12.7 dB from that of the baseline fan. This is a larger reduction than that of any of the conventional fan approaches evaluated in reference 2. The reduction is even larger than the reductions for those configurations where cutoff was assumed to completely remove the blade passing tone.

If the theoretical removal of the rotor-only noise as a result of cutoff were to occur, no blade passing tone would be observed (see table I). Since twice the blade passing frequency has been shifted beyond the rating range, this would leave no tone noise to be evaluated. The counterrotating fan would then theoretically have no tone noise rated on the perceived noise level scale. In actuality, the noise generated by the interaction with the support strut would provide a noise floor. Another possible noise floor might be provided by the interaction of the rotor with incoming flow distortions. These tone noises would be generated at the blade passing frequency and would contribute to the perceived fan noise. If the support strut were far enough downstream and the inlet flow were clean enough, it would be possible to reduce these sources below the broadband noise, and no tone noise would contribute to the rating.

Broadband Noise

If the complete removal of the blade passing tone were accomplished by cutoff, then, with no tones in the rated region, only the broadband noise would contribute to the perceived noise. Even in other cases, where the blade passing tone is present, the broadband noise may be a significant contributor to the total perceived noise. Therefore, even though this report deals specifically with tone noise reduction, some discussion of the broadband noise is included.

Reference 7 describes the typical broadband fan noise as consisting of two lobes as shown in figure 2. The first lobe is centered at about 2-1/2 times the blade passing frequency, and the second lobe is somewhere between 8 and 16 times the blade passing frequency. The broadband center frequency of the primary lobe shifts with the blade passing frequency, and it is implied that the generation mechanism may be tied to the interaction of the two blade rows. The importance of this is that a shift in blade passing frequency will shift the broadband noise also. This would move the broadband noise into regions where it is not rated as highly. Other sources of broadband noise, such as scrubbing noise on the walls of the duct, might not shift with the blade passing frequency. However, the results of reference 7 indicate that the sources which shift with frequency dominate the broadband noise spectra.

Figure 3 illustrates a shift of the broadband noise primary lobe. The lobe is shown first as it appears for the 3000-Hz blade passing tone of the baseline fan. The lobe is centered at 2-1/2 times the blade passing frequency, or 7500 Hz. Then, the lobe is shown as if it had been shifted directly with the shift in the blade passing frequency to the counterrotating value of 5727 Hz. This then shifts the center frequency to 14 317 Hz. This moves the peak noise level and a significant portion of the area under the curve beyond the perceived noise level rating range. The final curve shows the shift that would occur if the broadband noise were a result of blade row interaction. Here, the interaction noise has shifted to 11 454 Hz, and the broadband peak is shifted to 28 635 Hz. This shifts even more of the broadband noise beyond the rating range. The broadband noise which remains in the range has been shifted to higher frequencies which have a lower rating on the perceived noise level scale. The counterrotating fan may then have some potential for reducing broadband noise.

As indicated in the interaction noise discussion, the strength of the interaction might be increased because of the higher interaction velocities, or it could be reduced because of the smaller wakes and vortices. If the broadband peak were higher, it could negate some portion of the benefit associated with the frequency shift. The counterrotating fan would appear, nonetheless, to have a reasonable potential for reducing the broadband perceived noise as well as the tone perceived noise.

CONCLUDING REMARKS

The conventional approach to reducing perceived fan tone noise is to lower the tip speed of the device. An alternative approach of increasing the blade number of the fan to shift the tones to a higher frequency where they are not weighted as strongly in the perceived noise level calculation was investigated in reference 2. The additional concept in this report is to use a counterrotating fan to shift the cuton tone at twice the blade passing frequency to a frequency outside the rated range while leaving the cutoff rotor-only tone at blade passing frequency as the only tone in the rating range.

A counterrotating fan with 106 blades in each rotor was investigated in this report. The generated noise results from two sources; the rotor-only pressure field and the interaction of the two blade rows. The rotor-only tone for each rotor is generated at the blade passing frequency of each rotor, and if the rotor tip speed is subsonic, the noise is cut off. The first interaction tone for this fan occurs at twice the blade passing frequency. The blade passing frequency of the rotors is 5727 Hz and twice the blade passing frequency is 11 454 Hz. The interaction tone at twice the blade passing frequency is then at a frequency which is higher than that included in the perceived noise rating and does not contribute to the perceived noise rating of this fan.

The only tone that is rated is the cutoff blade passing tone. If cutoff is assumed to give an 8-dB reduction in the rotor-only tone at blade passing frequency, the counterrotating fan is calculated as having a 12.7-dB noise reduction relative to the baseline fan. This is more reduction potential than any of the conventional approaches investigated in reference 2. If cutoff provides the complete theoretical removal of the rotor-only tones, then the counterrotating fan could have no rated tone noise on the perceived noise level scale.

In addition to the shift of the tone noise to higher frequencies, the broadband noise is also shifted. This moves some of the broadband noise outside of the rating range and shifts other portions to regions which are not as highly rated. This indicates that the counterrotating fan has a potential for reducing broadband noise as well as reducing tone noise.

APPENDIX—RELATIVE ROTOR-ONLY AND ROTOR-STATOR INTERACTION TONE LEVELS

An exact comparison of the generated rotor-only tone and the rotor-stator interaction tone is not easily determined. Various references (e.g., ref. 8) have shown that the noise of a fan significantly increases when stators are added behind a rotor. Figure 10 of reference 8 indicates that this difference is 8 to 10 dB at the higher tip speeds tested. These experiments were performed in a duct, so they indicate the relative level of the cutoff rotor-only noise with respect to the cuton interaction noise after the generated noise has propagated down the duct. If, as indicated by reference 7, the practical effect of cutoff is to reduce the tone by 8 dB, the generated noise level of the rotor-only source is, at most, approximately equal to the generated interaction source.

An indication of the relative levels may also be inferred from counterrotating propeller data. Figure 16 of reference 9 shows the relative levels of the blade passing tones and the interaction tone as a function of rotor-to-rotor spacing. The blade passing tone level of the 11-blade front propeller (rotor only) is almost 10 dB lower than the first interaction tone at the 1.6 chord spacing, and it is only the same level after the spacing is increased to 3.5 chords. A trend with rotor blade number is also seen in this figure, where a 9-blade rear rotor is some 6 dB noisier than the 11-blade rotor. The baseline fan has a spacing of about 3.5, which would indicate that the noise of one 11-blade rotor is about the same as that of the interaction noise. The rotor-only noise of two 11-blade rotors would then be 3 dB noisier than the interaction noise.

As the blade number is increased, the rotor-only noise is decreased. The following expression from reference 10 roughly predicts the trend with blade number in reference 9.

$$mBJ_{mB}(0.8M_t mB \sin \theta)$$

where m is the order of the harmonic, B is the number of blades, $J_n(x)$ is a Bessel function of the first kind of order n and argument x, M_t is the blade tip rotational Mach number, and θ is the angle from the inlet. Calculated by this expression, the level of the rotor-only tone would be reduced 3 dB if the blade number were increased to 14. This would be the condition where the sum of the rotor-only tones from the two rotors would be equal to the interaction noise. At any higher blade number, the rotor-only noise from the two rotors would be less than the interaction noise. Since the counterrotating fan has considerably more than 14 blades (actually 106), the generated rotor-only noise is significantly less than the interaction noise.

Apparently, the generated rotor-only noise of a fan is, at most, the same and is probably considerably lower than the interaction noise. Since a larger noise level is predicted for the counterrotating fan when the rotor-only noise is assumed equal to the interaction noise, this is the conservative assumption used in the calculations of perceived noise levels.

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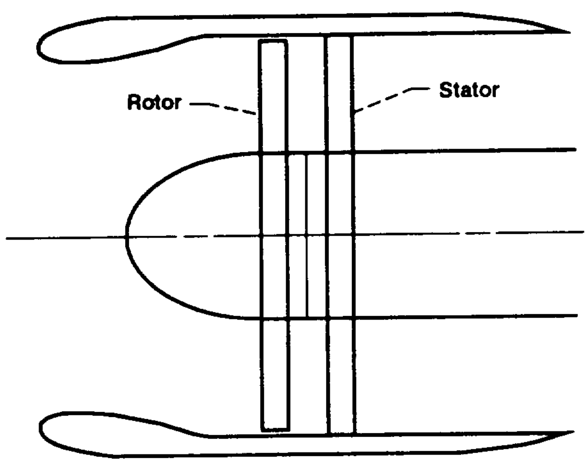
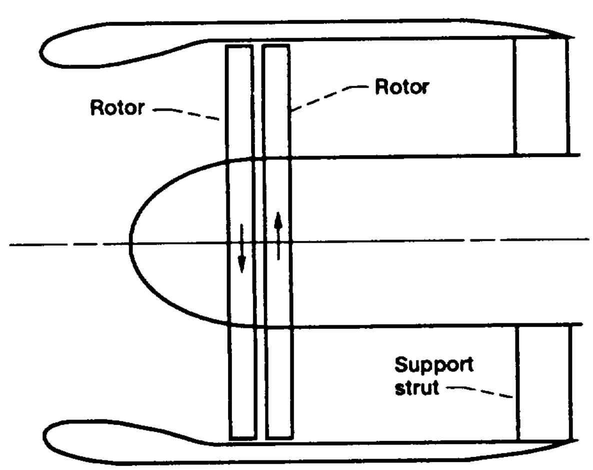
TABLE I.—PERCEIVED NOISE LEVELS OF BASELINE FAN AND COUNTERROTATING FAN

Harmonic of blade passing frequency	Baseline fan[a]		Counterrotating fan[b]			
			Blade passing tone reduced 8 dB by cutoff		Blade passing tone completely removed by cutoff	
	Sound pressure level, dB	Noy	Sound pressure level, dB	Noy	Sound pressure level, dB	Noy
1	100	134.0	92	67.2	0	0
2	97	94.9	(c)	-----	(c)	---
3	94	[d]51.0 [d]41.5	(c)	-----	(c)	---
4	(c)	------	(c)	-----	(c)	---
Perceived noise level, dB	113.4		100.7		(e)	

[a]Number of rotor blades, 53; number of stator vanes, 90; blade passing frequency, 3000 Hz.
[b]Number of rotor blades in each rotor, 106; blade passing frequency, 5727 Hz.
[c]Frequency above rated range.
[d]Tone split between two one-third-octave bands.
[e]No rated tone noise.



(a) Baseline fan, with 53 rotor blades and 90 stator vanes.



(b) Counterrotating fan, with 106 blades in each rotor.

Figure 1.—Fans used for comparison of noise reduction concepts.

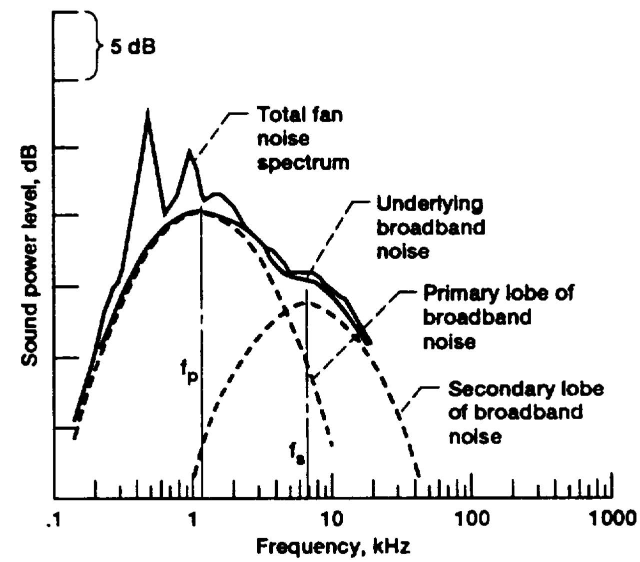


Figure 2.—Underlying broadband noise (from ref. 7).

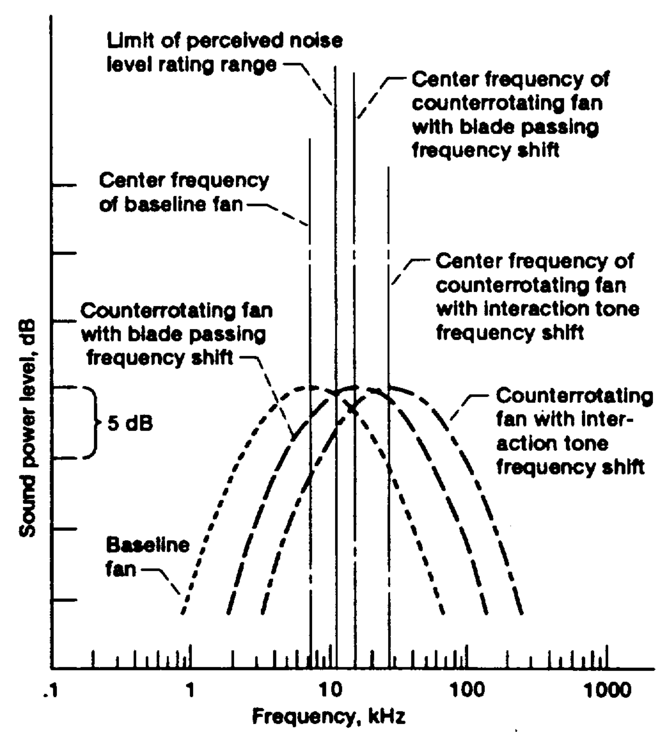


Figure 3.—Frequency shift of broadband noise.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (*Leave blank*)

2. REPORT DATE
August 1992

3. REPORT TYPE AND DATES COVERED
Technical Memorandum

4. TITLE AND SUBTITLE
A Concept for a Counterrotating Fan With Reduced Tone Noise

5. FUNDING NUMBERS
WU-535-03-10

6. AUTHOR(S)
James H. Dittmar

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
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Cleveland, Ohio 44135-3191

8. PERFORMING ORGANIZATION REPORT NUMBER
E-7138

9. SPONSORING/MONITORING AGENCY NAMES(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
Washington, D.C. 20546-0001

10. SPONSORING/MONITORING AGENCY REPORT NUMBER
NASA TM-105736

11. SUPPLEMENTARY NOTES
Responsible person, James H. Dittmar, (216) 433-3921.

12a. DISTRIBUTION/AVAILABILITY STATEMENT
Unclassified - Unlimited
Subject Category 71

12b. DISTRIBUTION CODE

13. ABSTRACT ***(Maximum 200 words)***
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14. SUBJECT TERMS
Noise; Fan; Turbofan

15. NUMBER OF PAGES
12

16. PRICE CODE
A03

17. SECURITY CLASSIFICATION OF REPORT
Unclassified

18. SECURITY CLASSIFICATION OF THIS PAGE
Unclassified

19. SECURITY CLASSIFICATION OF ABSTRACT
Unclassified

20. LIMITATION OF ABSTRACT

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. Z39-18
298-102

National Aeronautics and
Space Administration

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Cleveland, Ohio 44135

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