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Abstract

An existing fan stage, redesigned to reduce stator lift fluctuations, was acoustically tested for reduced noise generation. The lift fluctuations on the stator were reduced by increasing the stator chord, adjusting incidence angles, and by adjusting the rotor velocity diagrams. The experiments showed significantly reduced broadband noise levels in the middle to high frequencies. Blade passage tone power was not reduced, but decreases in the harmonics were observed. Aerodynamic improvements in both performance and efficiency were obtained.

Introduction

One of the noise generation mechanisms in a fan stage for turbofan engines is the interaction of the rotor wakes with the downstream stator vanes. These incoming wakes create lift fluctuations on the stator vanes and in turn produce noise. A number of researchers have formulated analytical models to predict the noise from this mechanism by calculating the fluctuating lift on the stator vanes as they intercept the rotating pattern of rotor wakes. Elements of these analyses were used in a previous report[1] to obtain an expression for the magnitude of the stator lift fluctuation. Results presented in reference 1 indicated that the fluctuating lift and therefore the noise of a fan stage, could be reduced by increasing the chord of the stator vanes and by adjusting the fan aerodynamic design to provide a minimization of the total stator response.

An existing fan stage, referred to as QF-2,[2] was redesigned using the methods of reference 1 to reduce the fluctuating lift and therefore the noise of the fan while maintaining a relatively constant aerodynamic design performance. The redesigned fan, designated QF-11, was tested at the Lewis Full-Scale Fan Noise Facility (Fig. 1) in the same manner as the existing fan, QF-2, had been tested previously. Acoustic and aerodynamic data taken with the redesigned fan are compared with the data for the existing fan.

Theoretical Background

This section will discuss the theoretical background for the redesigned fan both for the tone noise and for the broadband noise. The fluctuating lift model that was used as the theoretical basis to reduce the rotor wake-stator interaction tone noise will be examined and the broadband noise effects will be discussed.

Rotor-Stator Interaction Tone Noise

One purpose of the redesigned fan was to reduce the tone noise generated by rotor wake-stator interaction. A number of researchers have formulated analytical models to predict the noise from this mechanism by calculating the fluctuating lift on the stator vanes as they are struck by the rotor wakes. Examples of these analyses are given by Kemp and Sears[3] and Horlock[4]. Portions of some of these analyses were used in a previous report[1]. The expression used in the redesign for the magnitude of the stator fluctuating lift was taken from reference 1 (page 14) and is repeated here.

$$|\Delta L_N| = \left| \pi\rho \frac{1.6\sqrt{C_D}}{\left(\dfrac{X}{C_R} + 0.025\right)^{1/2}} U_3 V_1 \sin(\bar{\beta}_3 + \beta_3) \times \left[|S(\omega)| - \alpha \cot(\bar{\beta}_3 + \beta_3)|T(\omega)|\right]\right| \tag{1}$$

where

ΔL_N	fluctuating lift, force/area
ρ	fluid density, mass/(length)^3
C_D	rotor profile drag coefficient
C_R	rotor chord
X	distance from trailing edge of rotor to leading edge of stator
U_3	absolute velocity at stator inlet, length/time
V_1	relative velocity at rotor inlet, length/time
β_3	relative flow angle from rotor after translation to stator inlet, deg
$\bar{\beta}_3$	absolute flow angle at stator inlet, deg
α	angle of attack of stators relative to flow, radians
$\|S(\omega)\|$	magnitude of transverse response function
$\|T(\omega)\|$	magnitude of longitudinal response function
ω	reduced frequency ($\pi C_S/\ell$).
C_S	stator chord
ℓ	disturbance wavelength

Two general techniques suggested by this expression to reduce the magnitude of the stator lift fluctuations were investigated. These methods were (1) to reduce the magnitude of the stator response function terms $|S(\omega)|$ and $|T(\omega)|$ and (2) to enhance the cancellation of the two response terms (minimize the enclosed expression in Eq. (1)) by adjusting $\alpha\cot(\bar{\beta}_3 + \beta_3)$.

The magnitude of both of the response functions $|S(\omega)|$ and $|T(\omega)|$ decrease with increasing values of the reduced frequency ω. Since $\omega = \pi C_S/\ell$ where C_S is the stator chord and ℓ is the disturbance wavelength, ω is increased by increasing the stator chord and by decreasing the disturbance wavelength. The original fan had 53 rotor blades and 112 stator vanes with a stator chord of 6.83 centimeters (2.69 in.). The redesigned fan had 58 rotor blades and 70 stator vanes with a stator chord of 12.45 centimeter (4.9 in.). The increased chord of the new stator vanes, along with the slightly increased number of rotor blades, resulted in an increased "reduced frequency parameter," ω, which reduced the response functions $S(\omega)$ and $T(\omega)$ and in turn should reduce the noise producing lift fluctuations.

A number of parameters were varied in the redesigned fan to increase the amount of cancellation of the lift response terms (minimize the enclosed expression in Eq. (1)). The base fan was found to have too much rotor turning (as large as 60^o) near the hub for good cancellation. The redesigned fan, in order to improve the rotor velocity diagrams, reduced this turning at the hub by reducing the desired pressure ratio from the rotor hub. The same total pressure ratio was maintained by increasing the tip pressure ratio. This change in hub turning improved the lift response cancellation through the adjustment of the angles $(\beta_3 + \bar{\beta}_3)$. A comparison of the two sets of design rotor pressure ratios is found in Table 1 for the eleven hub to tip design stations. The cancellation was also increased by adjusting the angle of stator incidence. The acoustic theory indicated more incidence should be applied at the stator tip. The reasonable limit chosen was an addition of 3^o to the incidence and a table of the two sets of incidence, base and redesigned, is shown in Table 2.

A more complete discussion of the fan redesign process and the application of this method is found in the example in reference 1. The base fan discussed in this paper (QF-2) is the same as the base fan in the example of reference 1. The "improved fan" in reference 1 is the fan that was to have been built as the redesigned fan for this testing. However, some aeroelastic considerations, namely flutter, forced the reduction of rotor blades from the 60 of reference 1 to the 58 tested. The fan casing configuration in the test facility necessitated that an even number of stator vanes be used. The number was reduced to 70 from the 71 specified in reference 1. The results of reference 1 were also modified by the inclusion of total stator area in the calculations and small differences in aerodynamics were also present. For the modified redesign there was a predicted reduction in the rotor-stator blade passage tone of approximately 5 decibels for the redesigned fan as opposed to the 6 decibels in the example of reference 1. The reduction in the second harmonic of the blade passage tone caused by rotor-stator interaction was predicted to be about 5 decibels. The reductions in tone levels were based on the fan design aerodynamics at the 100% design speed point of the fan.

Broadband Noise

The internally generated broadband noise comes from many sources. These include turbulence interacting with the fan blading, shed vorticity from a blade, and scrubbing of the flow over blade surfaces and the fan duct surfaces to name a few. Broadband noise generated by turbulence and rotor wake irregularities interacting with the stator blades is of specific interest in this study. Such broadband noise should be reduced by the increased stator "reduced frequency parameter," ω, of the redesigned fan. Lieppman[5] and Goldstein, et al[6] have shown that the broadband acoustic power generated in a turbulent flow is directly related to the Sear's function, $S(\omega)$, mentioned previously. Here again, the reduced frequency, $\omega = \pi C_S/\ell$ depends on the blade chord and the wavelength of the incoming gust or turbulence. The increase in the stator chord should result in a broadband noise reduction. The amount of the reduction achieved by the increased chord should depend on the properties of the incoming turbulence. Since no measurement of the turbulence striking the stator has been made, a prediction of the reduction is not undertaken. Observations of broadband noise reduction from increased stator chord have been made previously in reference 7. These previous reductions were significant, in some cases more than 5 dB, and occurred in the frequency range from 1000 to 20,000 Hz. It was therefore expected that a broadband noise reduction over this general frequency range would be observed as a result of the redesign of the fan stator vanes.

Apparatus and Procedure

Fan Stages

Both the original (QF-2) and redesigned fan (QF-11) stages compared in this study are full scale 1.83 meter (6 ft) diameter fans with a design pressure ratio of 1.5. The aerodynamic design features of the QF-2 and QF-11 stages are given in Tables 3 and 4, respectively. The redesigned fan featured an increase in rotor blade number from 53 to 58 and a decrease in stator vanes from 112 to 70. The stator vane decrease was to accommodate the increase in stator chord from 6.83 centimeters (2.69 in.) to 12.45 centimeters (4.9 in.). Redesign also included changes in stator incidence angles and changes in rotor pressure ratio distribution as given in Tables 1 and 2.

Aerodynamic and acoustic data for the original (QF-2) fan, to be used for comparative purposes, were obtained in a previous study and were reported in reference 2. The data were acquired at the same facility and reduced by the same procedures as those to be described for the redesigned fan.

Test Facility and Configurations

The experiments reported herein were conducted at the Full-Scale Fan Test Facility at the Lewis Research Center. Figure 1(a) shows the test site and Fig. 1(b) shows a plan view of the test facility. Acoustic data were obtained by 1.27 centimeter (0.5 in.) condenser microphones located at 10 degree increments from 10 to 160 degrees as shown in Fig. 1(b). The microphones were level with the fan centerline, 5.79 meters (19 ft) above the ground on a 30.48 meter (100 ft) radius. A complete description of the acoustic instrumentation and the data acquisition techniques are given in references 8 and 9.

Three samples of acoustic data were taken at each test condition and averaged to minimize the effect of short-term fluctuations in the generated noise. The data taken at 60, 70, 80, and 90 percent of design speed were recorded on magnetic tape and a 1/3-octave band analysis was performed.

Aerodynamic data were taken by instrumentation located in front of the fan to determine fan mass flows and downstream of the stator vanes to determine performance and efficiency. The upstream instrumentation consisted of six wall static pressure taps and the downstream instrumentation consisted of four aerodynamic rakes each of which contained nine thermocouples and ten total pressure tubes. Outside wall static pressure taps were located on either side of each of these aerodynamic rakes. The aerodynamic data from nine samples were averaged to yield the aerodynamic data presented for each test point. Aerodynamic and acoustic tests were run at 60, 70, 80, and 90 percent design speed with the design nozzle area.

Results and Discussion

Acoustic Data

As mentioned previously in the "Theoretical Background" section an existing fan stage was redesigned to reduce the rotor wake-stator interaction noise. The noise reductions achieved with this redesigned fan can be seen in the one-third octave sound power spectra of Fig. 2. Figures 2(a) through 2(d) are for 60 through 90% speed respectively.

Tone Noise. Although theory indicated that the blade passage tone generated by rotor wake-stator interaction should have been reduced with the redesign, no reduction in the sound power at the blade passage frequency is observed in the data of Fig. 2. (The differences at the 90% speed point are the result of the tone lying between two 1/3-octave bands and its energy being split between them.) A possible explanation for this lack of reduction is that an existing inlet flow distortion is interacting with the rotor blades and controlling the sound power level. This distortion control of blade passage frequency noise was observed in reference 2 in the same facility and its ability to control the tone was also shown in reference 7. Another possibility is that a counter-balancing effect is being observed. In the redesigned stage the propagation of the fundamental tone is not "cut-off" in the duct while in the original stage the proper number of rotor and stator blades existed to provide "cut-off". The reductions achieved by reducing the fluctuating stator forces may, in this stage, be counter-balanced by the removal of the cutoff criterion to provide the same blade passage tone sound power level.

Although no significant changes in the sound power level at the blade passage tone were observed, changes in the directivity of the tone were noted. Figure 3 is a plot of the sound pressure level in the 1/3-octave band containing the blade passage tone at the 60% design speed test point. The 60% speed case is chosen for illustration since both the original and the redesigned fan have their blade passage tone contained in the same 1/3-octave band (2000 Hz center frequency). As seen in Fig. 3 the blade passage tone has typically been reduced in the front of the fan (10 to 70°), with the redesigned fan but has been increased toward the rear (90° to 140°). The inlet reduction may be an indication that the two fan rotors have a different response to the same inlet flow distortion or possibly that the reduced stator lift fluctuations have reduced the inlet noise. The increase in the aft noise is possibly a result of the violation of "cutoff" in the redesigned fan. Heidmann[10] has indicated for this test facility that "cutoff" may be effective in reducing the fundamental tone propagation from the aft duct. The violation of "cutoff" for the redesigned fan may then be responsible for the increased aft noise and may provide some support for the counterbalancing effect mentioned previously.

Reductions in the harmonics of the blade passage tone, which are also indicated by the theory, are observed in the data of Fig. 2. As observed in previous papers[2,7] the second harmonic was not as greatly affected by the inflow distortion known to be present in this test facility. In addition the harmonics are not "cut-off" in either fan so no counterbalancing effect is expected. The noise reductions achieved at the harmonics are not as great as those predicted by the rotor-wake-stator interaction theory but are significant.

An interesting effect may be observed in the variation of the reduction of the second harmonic with percent of design speed as shown in Fig. 4. This figure indicates a growing reduction as the speed is increased with a leveling off toward 90% speed. As indicated in the "theoretical background" section the predicted reduction of around 5 decibels was based on the design point aerodynamic calculations (100% speed). The fall off of the reduction toward the low speed points may be a result of reduced cancellation between the response functions since the velocity diagrams at these low speed points are not the same as the design values used in the calculations. As the speed is increased the velocity diagrams should more closely match the design numbers used in the noise calculations and the data thus approach the expected reduction in noise.

In the redesigned fan stage, as well as the original stage, the perceived noise level was controlled by the blade passage tone and little reduction in PNL was observed due to the harmonic reductions. However, the reduction in such harmonics may be important for STOL-type fans with low numbers of rotor blades and low tip speeds where the harmonics are the largest contributors to the perceived noise level. In this case, the reductions in the harmonics could result directly in a perceived noise reduction.

Broadband. As can be observed in Fig. 2, significant broadband reductions were also achieved with the redesigned fan stage. The broadband noise generated by turbulence, rotor wake irregularities, etc. interacting with the stator vanes was expected to be reduced with the reduction of the stator lift response functions. This reduction was observed over a large frequency range, at all speed points, and amounted to up to 5 decibels at some frequencies. The reductions in the broadband noise appear to be relatively uniform with angle as can be seen in Fig. 5. Figure 5 is a plot of the broadband SPL for the 1/3-octave band centered at 5000 Hz for the 60% speed point versus angle. The redesigned fan exhibited a consistent reduction in broadband noise over a large frequency range at all polar angles. This result is particularly significant because the reduction was obtained without the weight or performance penalty usually encountered with acoustic suppression.

Aerodyanmic Results

As mentioned previously the redesigned fan stage was designed to have the same design pressure ratio and mass flow as the original fan stage at 100% design speed. Neither of the fans were run at 100% design speed so no comparisons are made at the design point. It was discovered during testing that the redesigned fan performed better than the original fan at the part speed points that were tested. This is seen in Fig. 6 which is a plot of pressure ratio versus mass flow for four speed points. The original fan stage (QF-2) is indicated with the open symbols where data from three nozzle sizes were available and the redesigned stage (QF-11) is indicated with the solid symbols for the design nozzle area. The design point pressure ratio and mass flow are also indicated on the figure. As can be observed the new fan exhibited larger pressure ratios and mass flows with the design nozzle than did the original fan at all part speed points tested. Aerodynamic efficiency data obtained on this outdoor noise test facility are somewhat compromised by not enough temperature sensors and comparisons with efficiencies taken on a specially instrumented quarter scale test facility have historically been only fair. The efficiency data taken on the outdoor noise facility, however, indicate that the redesigned fan did perform better than the original fan. This can be seen in Fig. 7 where the stage adiabatic efficiency is plotted versus percentage of fan design speed. The redesigned fan is more efficient than the original fan at all tested speed points with the design nozzle. However, in fairness, it should be noted that the trends of the data with corrected speed might indicate that the efficiency at design speed could be equal or lower than the base fan. Although the improvement in aerodynamic performance shown by this redesigned fan stage does not prove that designing for reduced lift fluctuation will give better aerodynamics, at least it indicates that reduced noise through lift fluctuation reduction and improved aerodynamics are compatible.

Summary of Results

An existing fan stage was redesigned to reduce the rotor wake-stator interaction noise. The redesigned fan had an increased stator chord, an increased number of rotor blades, altered rotor velocity diagrams, and altered stator incidence angles. The ultimate goal of these changes was to reduce the fluctuating lift on the stators and therefore reduce the generated noise. In comparing the redesigned fan with the original fan it was found that:

1. A reduction in the blade passage tone was observed in the front hemisphere, however, an increase in the tone was observed in the rear so that no net change in the blade passage tone power occurred. The lack of the predicted blade passage tone power reduction is possibly the result of an inlet flow distortion controlling the tone noise or a counterbalancing effect as the result of a violation of the "cutoff" ratio of rotor blades and stator vanes in the redesigned fan.

2. Reductions in the harmonics of the blade passage tone were observed. This result provides some evidence for tone reduction by reducing stator lift fluctuations even though the reductions observed were not as great as those predicted by theory. Such harmonic reduction could be effective for a low tip speed - low blade number fan stage where the harmonic reduction could result directly in perceived noise level reductions.

3. The broadband noise output of the fan was significantly reduced by the reduced lift fluctuation design. This reduction occurred over a large frequency range, at all speed points, and amounted to up to 5 decibels at some frequencies.

4. Aerodynamic data showed the redesigned fan had both better performance and slightly higher efficiency than the original fan.

References

1. Dittmar, J. H., "Methods for Reducing Blade Passing Frequency Noise Generated by Rotor-Wake-Stator Interaction," NASA TM X-2669, Nov. 1972.

2. Povinelli, F. P., Dittmar, J., and Woodward, R. P., "Effects of Installation Caused Flow Distortion on Noise from a Fan Designed for Turbofan Engines," NASA TN D-7076, Nov. 1972.

3. Kemp, N. H.; and Sears, W. R., "The Unsteady Forces Due to Viscous Wakes in Turbomachines," Journal of the Aeronautical Sciences., Vol. 22, July 1955, pp. 478-483.

4. Horlock, J. H., "Fluctuating Lift Forces on Aerofoils Moving Through Transverse and Chordwise Gusts," Journal of Basic Engineering, Vol. 90, Dec. 1968, pp. 494-500.

5. Lieppmann, H. W., "On the Application of Statistical Concepts to the Buffeting Problem," Journal of the Aeronautical Sciences, Vol. 19, Dec. 1952, pp. 793-800,822.

6. Goldstein, A. W., Glaser, F. W., and Coats, J. W., "Effect of Casing Boundary-Layer Removal on Noise of a Turbofan Rotor," NASA TN D-6763, Apr. 1972.

7. Dittmar, J. H., et al, "Effects of Long-Chord Acoustically Treated Stator Vanes on Fan Noise. I - Effect of Long-Chord (Taped Stator)," NASA TN D-8062, Oct. 1975.

8. Leonard, B. R., et al., "Acoustic and Aerodynamic Performance of a 6-Foot-Diameter Fan for Turbofan Engines. I - Design of Facility and QF-1 Fan," NASA TN D-5877, Jul. 1970.

9. Montegani, F. J., "Some Propulsion System Noise Data Handling Conventions and Computer Programs Used At the Lewis Research Center," NASA TM X-3013, Mar. 1974.

10. Heidmann, M. F., "An Observation on Tone Cut-Off in Static Test Data from Jet Engine Fans. NASA TM X-3296, Sept. 1975.

Table 1. - Pressure Ratio at Rotor Exit

Blade section	Pressure ratio at rotor exit	
	Base fan	Redesigned fan
1 (tip)	1.541	1.556
2	↓	↓
3	↓	↓
4	↓	↓
5	↓	↓
6	↓	↓
7	↓	↓
8	↓	↓
9	↓	1.556
10	↓	1.485
11 (hub)	↓	1.375

Table 2. - Angle of Incidence to Suction Surface of Stator Blade

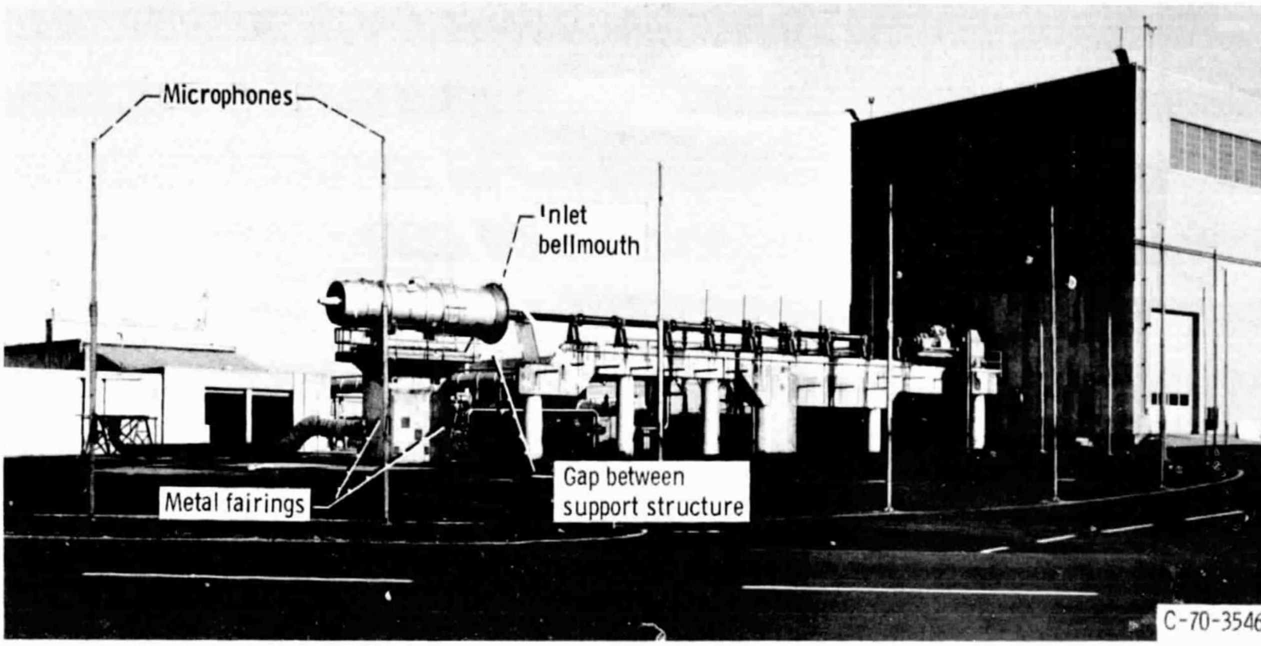
Blade section	Angle of incidence, degree	
	Base fan	Redesigned fan
1 (tip	0	3.0
2		
3		
4		
5		
6		3.0
7		2.9
8		2.6
9		2.1
10		1.2
11 (hub)	0	-0.7

Table 3. - Design Values for Base Fan

Rotor tip diameter, m(in.)	1.824(71.81)
Stator tip diameter, m(in.)	1.726(67.94)
Rotor tip speed (cruise design value, corrected), m/sec (ft/sec)	337.4(1107)
Design stagnation pressure ratio	1.5
Design weight flow (corrected), kg/sec (lbm/sec)	.396(873)
Rotor hub-tip radius ratio (inflow face)	0.50
Stator hub-tip radius ratio	0.59
Rotor-stator spacing (rotor trailing edge to stator leading edge at the hub), cm(in.)	50.8(20)
Number of rotor blades	53
Number of stator blades	112
Rotor chord length, cm(in.)	13.97(5.5)
Stator chord length, cm(in.)	6.83(2.69)

Table 4. - Design Values for Redesigned Fan

Rotor tip diameter, m(in.)	1.824(71.81)
Stator tip diameter, m(in.)	1.726(67.94)
Rotor tip speed (cruise desidn value, corredted), m/sec (ft/sec)	350.1(1149)
Design stagnation pressure ratio	1.5
Design weight flow (corrected), kg/sec (lbm/sec)	.396(873)
Rotor hub-tip radius ratio (inflow face)	0.50
Stator hub-tip radius ratio	0.59
Rotor-stator spacing (rotor trailing edge to stator leading edge at the hub), cm(in.)	50.8(20)
Number of rotor blades	58
Number of stator blades	70
Rotor chord length, cm(in.)	13.70(5.4)
Stator chord length, cm(in.)	12.45(4.9)



(a) Test site.

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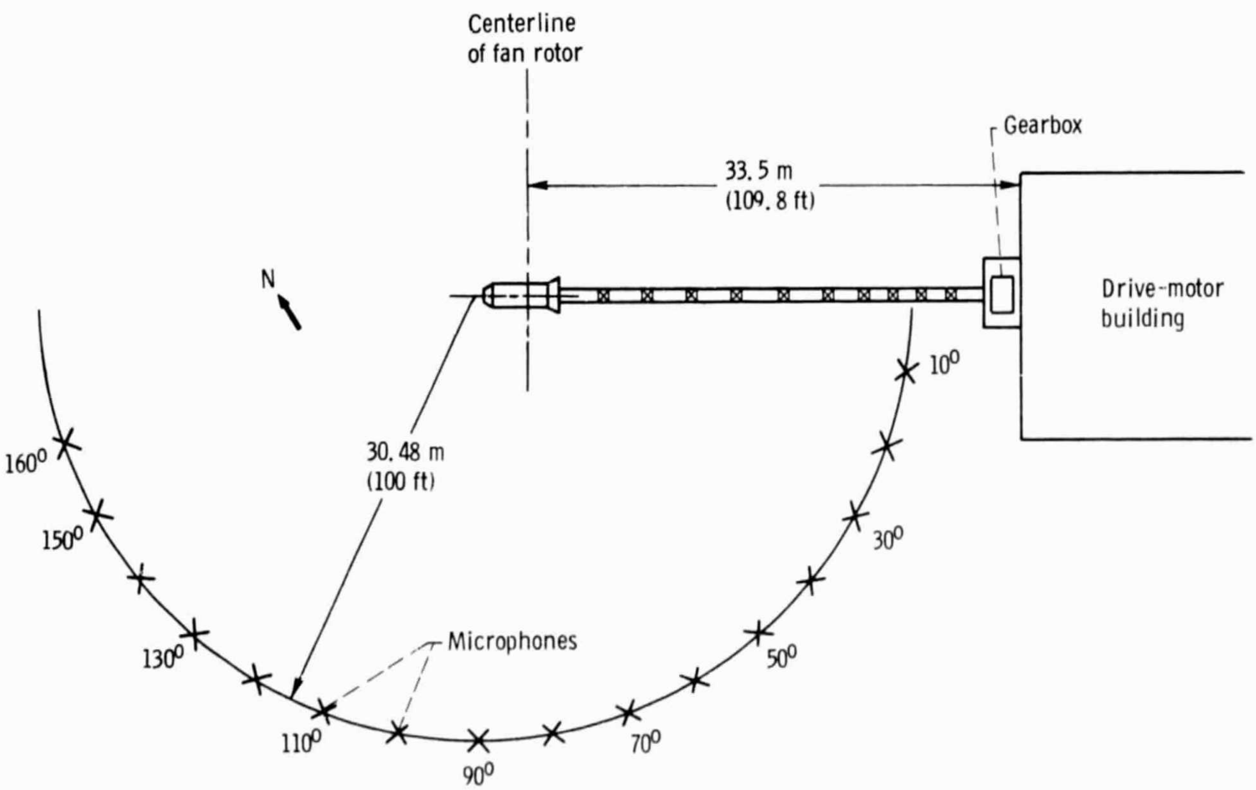

(b) Plan view of test site.

Figure 1. - Full-scale fan test facility.

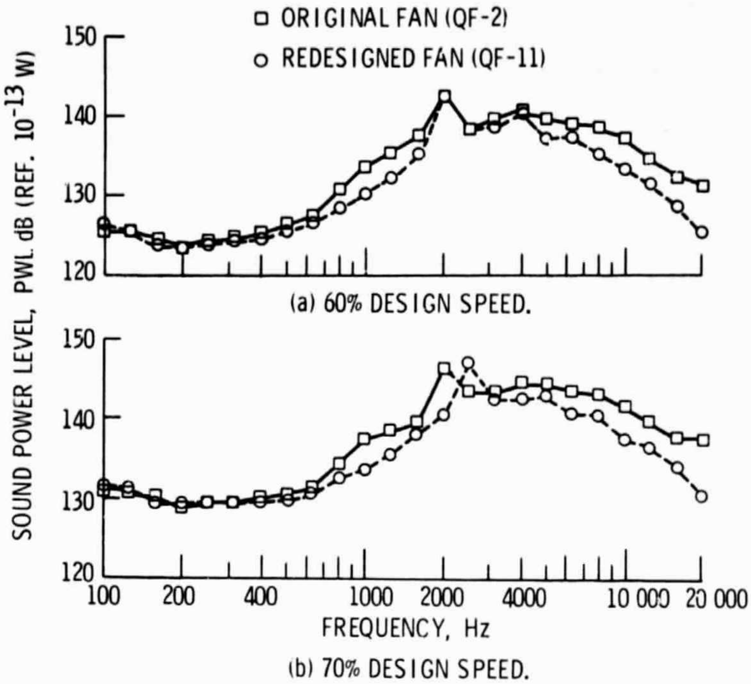

Figure 2. - One-third octave sound power spectra with design nozzle.

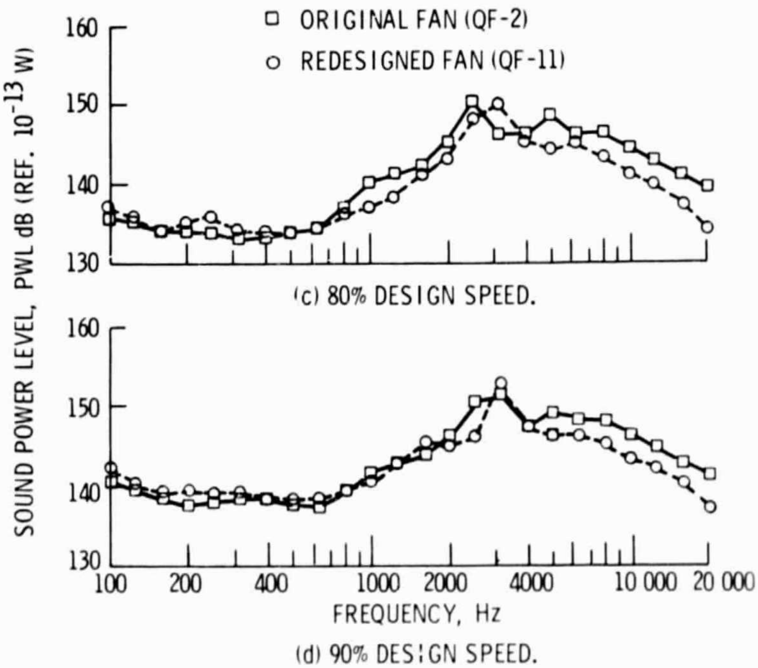

Figure 2. - Concluded.

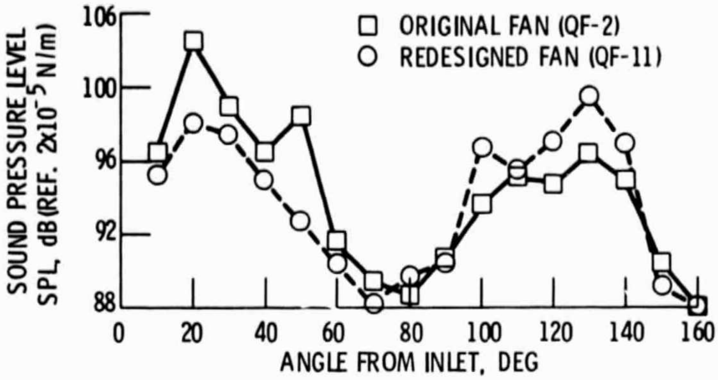

Figure 3. - Sound pressure level versus angle 2000 Hz center frequency one-third octave band containing blade passage tone at 60% design speed.

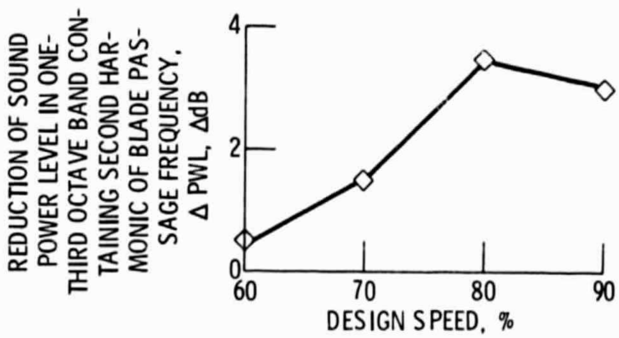

Figure 4. - Second harmonic reductions vs. percent speed.

Figure 5. - Sound pressure level versus angle for broadband noise in the 5000 Hz center frequency one-third octave band at 60% design speed.

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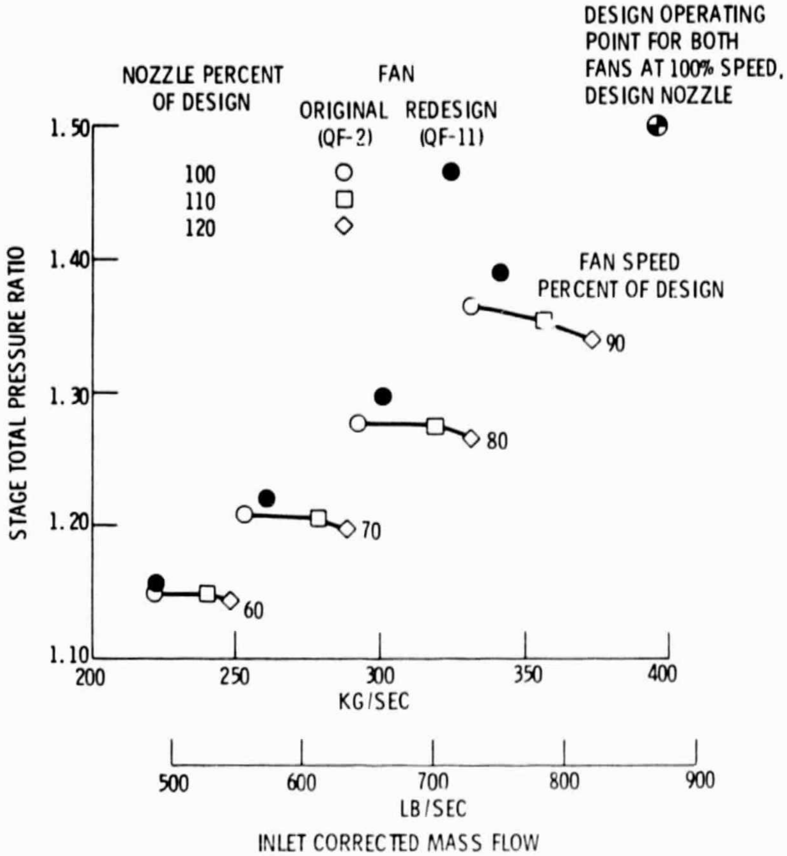

Figure 6. - Fan operating map.

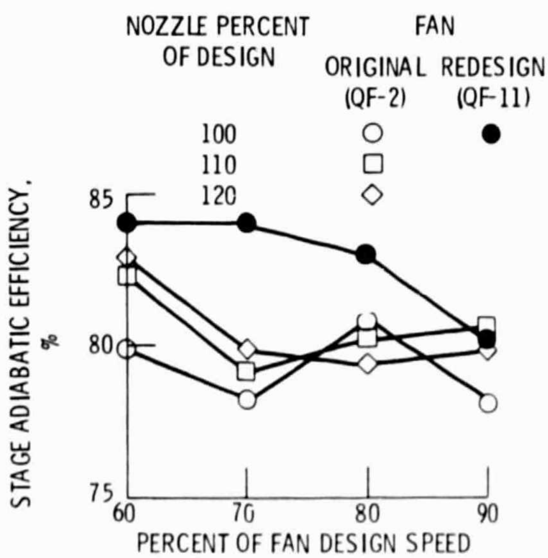

Figure 7. - Efficiency variation with percent speed.