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AEROACOUSTIC CHARACTERISTICS OF A LARGE, VARIABLE-PITCH, VARIABLE-SPEED FAN SYSTEM

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INTRODUCTION

This study was undertaken to evaluate the acoustic and aerodynamic performance of the new drive system for the NASA Ames 40- by 80-/80- by 120-Foot Wind Tunnel [1]. The drive system is unusual in size and operation because the 12.2-m diam fans can be operated at variable speed or variable blade pitch, or both. Thus, it was possible to document fan noise versus fan-operating condition and mass flow to minimize the noise at all wind-tunnel airspeeds. This paper shows the important parametric tradeoffs as well as an improved empirical noise prediction scheme. Although variable-pitch and variable-speed fans have been studied extensively in the aerospace industry as propulsion devices, they are rarely used for air-moving ducts, probably because of their complexity and cost compared with those of small, fixed-pitch fans. However, for large fans which must generate a range of mass flows, variable-pitch fans can be cheaper than variable-speed fans that need a variable-frequency power source.

FAN SYSTEM AND AERODYNAMIC PERFORMANCE

Fans

The six 12.2-m diam fans are placed in the wind-tunnel drive section in two rows of three fans each (fig. 1). Table 1 lists the geometric and performance characteristics of each fan. The fans can be operated from 0 to 180 rpm and with blade-pitch angles from -18° to 52° relative to the fan disc. (Blade pitch is measured at the 3/4 radius station.) The maximum mass-flow rate generated is 48,323 kg/s that results in a maximum airspeed in the 40- by 80-Foot Wind Tunnel of 150 m/s and a maximum (average) airspeed through the fan section of 69 m/s. A given mass-flow rate can be achieved by using one of many speed/pitch combinations. Further characteristics such as blade camber, efficiency, etc., are described in [1].

Wind tunnel

Figure 2 shows the wind-tunnel circuit and microphone measurement station downstream of the fan section. This paper deals only with the 40 by 80 closed-loop circuit with the 80 by 120 leg closed off and with *all* six fans operating. The single microphone station can be used to determine fan sound power because the sound field is fairly diffuse in that area; previous studies that used calibrated noise sources established the relationship between fan-sound pressure levels in that area and fan-sound power levels.

ACOUSTIC RESULTS AND PREDICTIONS

Experimental results

With fixed-blade pitch, the fan-sound power variation with tip speed followed typical fan laws. As shown in fig. 3, the sound (W) varied with $V^{5.5}$. With fixed tip speed, the fan-sound power variation with blade-pitch angle is shown in fig. 4. On a linear scale, the sound power (dB) varied as 0.3 $\Delta\beta$, where $\Delta\beta$ is the change in blade pitch in degrees. This relationship held over a wide range of blade angles, but does not hold at very low or very high pitch angles. At very low pitch angles the fan-induced flow is weak and the highly twisted blades can stall at the root and tip and cause a noise increase. At high pitch angles, well above those shown in fig. 4, the blade tips will also stall and cause a noise increase.

We now have the information necessary to explain the acoustic-performance map of the fan-drive system operating at variable speed or variable pitch as shown in fig. 5. (Data were obtained only from part of the fan-operating range.) Over most of the mass-flow range, the sound power follows the curves of figs. 3 and 4. A significant result shown on fig. 5 is the strong effect on sound due to operating condition at a given mass flow. Obviously, the fan is quietest at low tip speed and high blade pitch, providing the fan does not stall. At the same configuration, the electrical-power consumption was less as it followed the same trend. This logical correlation of noise, power consumption, and speed is supported by the fact that sound from aerodynamic bodies is proportional to the sixth power of speed (ideal dipole), and aerodynamic power required is proportional to speed cubed.

An important parameter that is not shown on fig. 5 is the fan efficiency. The fan has a fairly flat efficiency versus mass-flow curve [1] that peaks at a high pitch angle of 50°. Flat efficiency was achieved by designing the rotor blades with high twist and camber so that the spanwise blade loading would be flat at the design point ($\beta = 50°$). Therefore, the noise variation with pitch-angle increase (fixed tip speed) is relatively shallow on fig. 5 because the blade tip is lightly loaded and is slowly approaching design loading. Beyond the design point ($\beta > 50°$) the tip will stall, efficiency will drop, and noise will rise. If the fan were designed with different twist and camber, the fan efficiency might drop and the noise might go up faster, relative to the mass-flow rate, than shown on fig. 5. Thus, the slope of the constant speed curves on fig. 5 may be configuration-dependent; different fans could have curves with different slopes. On the other hand, the fixed-pitch curves on fig. 5 have slopes that will be similar for any reasonably well-designed fan. This is because a fan operating along a fixed-pitch curve operates with constant efficiency since the advance ratio (inflow velocity/tip speed) is constant along that curve.

Prediction

The most accurate way to predict the noise of a specific fan is to use an analytical method that incorporates the important features of the flow field, blade loading, source radiation, duct effects, etc. A simpler way is to use an empirical equation that predicts the noise of an "average" fan operating under "average" conditions. Such an equation that gives the fan sound power in a third-octave band is given by [2]

$$Lw(f) = -48.2 - 10 \log [1 + (4.4X)^2] + 10 \log f + 40 \log N + 70 \log D_t + 10 \log Q + 10 \log F_n. \quad (\text{dB re } 10^{-12}\text{ W}) \tag{1}$$

where $X = Qf/N$; $Q = 1 - (D_H/D_t)^3$; D_H = hub diam, m; D_t = tip diam, m; f = center frequency of band, Hz; N = rotational speed, rpm; and F_n = number of fans.

The overall sound power level can be obtained by summing the power (W) in the third-octave bands. Equation (1) does not account for blade-pitch variation. Therefore, it was modified according to the results of fig. 4 to give

$$Lw(f) = -58.2 - 10\log[1 + (4.4X)^2] + 10\log f + 40\log N + 70\log D_t + 10\log Q$$
$$+ 10\log F_n + 0.3\beta, \qquad (\text{dB re } 10^{-12}\text{ W}) \qquad (2)$$

where β is the blade-pitch angle at the 3/4 radius station. Figure 6 illustrates a comparison of the predicted overall sound power levels with the data of fig. 5. Figure 7 shows that Eq. (2) also predicted a third-octave power spectrum reasonably close to that measured. However, the corrections to Eq. (1) may depend on the specific fan-blade design as explained in the previous section.

CONCLUSIONS

This study has shown that a fan system with variable-speed and variable-pitch rotor blades allows the operator to control noise and energy consumption, at a given mass flow rate, through the choice of blade speed and pitch. A low speed and high blade pitch will generally create the least noise at the least energy cost, whereas high speed and low blade pitch will have the opposite effect. An empirical method was described which includes the effect of pitch angle on noise and predicts the sound power of this fan system reasonably well.

REFERENCES

[1] K.W. Mort, D.F. Engelbert, and J.C. Dusterberry, "Status and capabilities of the national full scale facility," AIAA Paper 82-0607, March 1982.

[2] P.T. Soderman and V.R. Page, "Acoustic performance of two 1.83-meter-diameter fans designed for a wind-tunnel drive system," NASA TP 1008, Aug. 1977.

TABLE 1.– FAN GEOMETRY AND PERFORMANCE CHARACTERISTICS

	ROTOR	STATOR
NUMBER OF BLADES	16	23
DIAMETER	12.2 m	12.2 m
HUB-TIP RATIO	0.438	0.438
ROOT CHORD	1.22 m	0.74 m
TIP CHORD	0.90 m	0.74 m
TWIST, ROOT TO TIP	41.4°	0°
MAXIMUM rpm	180	0

ROTOR/STATOR SPACING = 2.4 m

WIND TUNNEL CIRCUIT LOSSES, $\Delta p/q$ = 0.176

FAN TOTAL PRESSURE RISE, $\Delta p = 1.05\,(10^{-6})\,\dot{m}^2$, N/m²
WHERE q = TEST SECTION DYNAMIC PRESSURE, N/m²
$\dot{m}$ = MASS FLOW RATE IN kg/sec

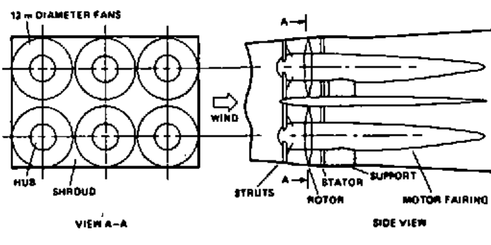

Fig. 1 Arrangement of six fan-drive units.

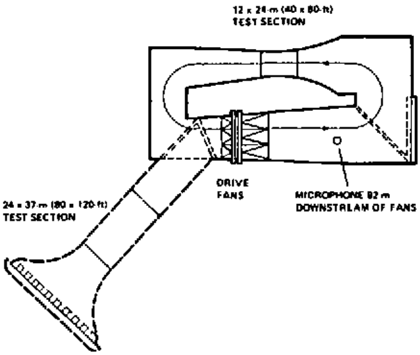

Fig. 2 Wind tunnel circuit.

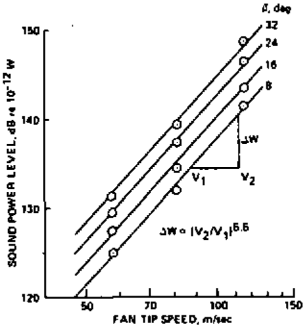

Fig. 3 Fan sound power at fixed pitch angles.

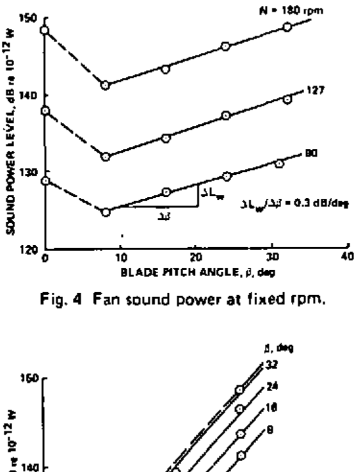

Fig. 4 Fan sound power at fixed rpm.

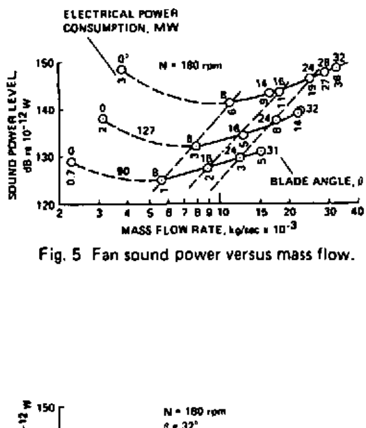

Fig. 5 Fan sound power versus mass flow.

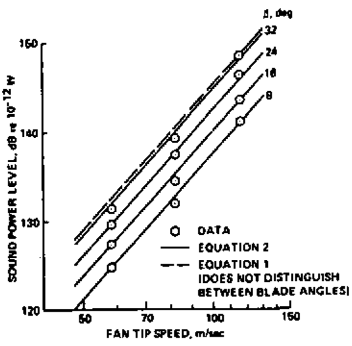

Fig. 6 Measured and predicted sound power.

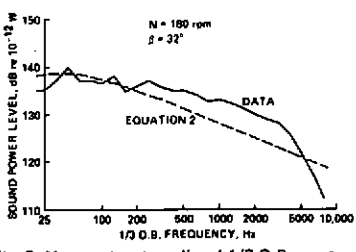

Fig. 7 Measured and predicted 1/3 O.B. spectra.