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Experimental design of a low noise centrifugal fan

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EXPERIMENTAL DESIGN OF A LOW NOISE CENTRIFUGAL FAN

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ABSTRACT

The effect of various design parameters on broadband noise generation by centrifugal fans has been studied. The parameters varied included: blade type, radial and axial inlet clearance, scroll development angle, scroll development length and cutoff clearance. Scaling laws for pressure, flow and noise were applied to allow the A-weighted sound power levels of different fans to be compared at the same operating points. That approach has allowed an optimum low noise fan design to be identified. To extend the usefulness of that design, a semi-empirical procedure for predicting the broadband sound power spectrum of any member of a family of geometrically similar centrifugal fans has also been developed. The prediction procedure allows the calculation of 1/3-octave band sound power levels based on a measurement of the sound power radiated by a single member of the fan family. By the use of that prediction procedure it is straightforward to identify optimally quiet fan designs that satisfy a range of pumping requirements.

RESUME

L'effet de divers paramètres du design des ventilateurs centrifuges sur leur génération de bruit en bande large de fréquences a été étudié. Les paramètres variés incluent: le type de pale, le débattement d'entrée radial et axial, l'angle et la longueur de développement de la roue, le débattement en sortie. Des lois de réduction à la même échelle furent appliquées à la pression, au flux et au bruit, pour permettre la comparaison, à un même point de fonctionnement, des niveaux de puissance acoustique de différents ventilateurs. Cette approche a permis l'identification d'un design optimal de ventilateur à faible émission de bruit. Pour étendre l'utilité de ce design, une méthode semi-empirique a été développée pour prédire le spectre de puissance acoustique en bande large de n'importe quel membre d'une même famille géométrique de ventilateurs centrifuges. La procédure de prédiction permet le calcul en tiers d'octave des niveaux de puissance acoustique basé sur une mesure de la puissance acoustique rayonnée par un seul membre de la même famille de ventilateurs. En utilisant cette méthode de prédiction on peut directement identifier les meilleurs designs pour des ventilateurs silencieux qui auraient à satisfaire un certain nombres d'exigences de ventilation.

1- INTRODUCTION

An experimental program has been conducted to study the influence of various design parameters on the broadband noise generated by centrifugal fans of the type used to cool large computer equipment. The design parameters considered include: blade type, radial and axial inlet clearance, scroll development angle, scroll development length, and cutoff clearance. In some respects the work reported here is similar to that of Morinushi who conducted a study of centrifugal fan noise generation that focused on impeller parameters: e.g., the blade setting angle, and the blade pitch-to-chord ratio [1]. Based on his tests, Morinushi was able to identify values for those design parameters that minimized the sound pressure level at a single point in the far field of a centrifugal fan's exit. Here we have followed Morinushi in attempting to find a combination of design parameters that minimize fan noise generation; however in the present instance it was decided to concentrate on housing design parameters rather than impeller parameters. In addition, the design optimization was based on the total sound power radiated by the fan, rather than on the sound pressure measured at a single point. Finally, it was decided to concentrate exclusively on broadband random noise since it was felt that tonal noise generation and its control were reasonably well understood. Thus in the measurements described below the blade passage tone and its harmonics were removed from the measured sound power spectra along with the harmonic components of the blower drive motor. Consequently, the results of the present work apply only to the broadband random component of the noise generated by centrifugal fans.

Here we have isolated the acoustical effects of individual design parameters by measuring the sound power radiated by each member of a series of fans that differed in no more than one or two design parameters simultaneously. However, the task of experimentally identifying the acoustical effect of individual design parameters is complicated by the fact that it is not useful to compare directly the sound power generated by different fans since any design change inevitably alters the fan's aerodynamic performance as well as its sound power. As a result, some component of the sound power change that accompanies a design change results simply from the attendant change in aerodynamic performance. A scaling procedure has therefore been developed so that measured fan sound powers could be adjusted to remove noise level differences resulting from aerodynamic performance differences. By using that procedure it has been possible to identify unequivocally the acoustical effects of design changes and, by extension, to identify a set of design parameters that results in the least noise generation. A complete description of that scaling procedure is presented here along with the optimum, low noise fan design that resulted from the extensive testing program that was conducted as part of this study.

To extend the usefulness of the optimum fan design, a prediction scheme has been developed that allows one to predict the sound power spectrum generated by any member of a family of geometrically similar centrifugal fans that includes the optimum design. It is, of course, well known that "sound laws" can be used to predict the *overall* sound power level radiated by centrifugal fans when the sound power level of a geometrically similar reference fan is known [2]. In this paper we present a method for predicting fan noise *spectra* when the sound power spectrum of a reference fan is known. In particular, it will be shown that a parameterized spectral representation proposed by Maling [3] may be combined with measured reference data to yield sound power predictions for a complete family of geometrically similar fans. Maling's approach has been significantly extended here by allowing for impeller width variations in the scaling procedure. A key benefit of the present prediction procedure is that it may be used to define "noise surfaces" that allow one to identify the design of a low noise fan that meets a given pumping requirement while generating the least amount of noise. Initial studies of those surfaces have indicated that the use of a fan having a narrow, large diameter impeller running at low speeds often results in the lowest noise levels. Finally, note that a more complete description of the work presented here is available elsewhere [4].

2- EXPERIMENTAL APPARATUS

2.1 Measurement of Aerodynamic Performance

The aerodynamic performance of each fan considered in the present study was measured by using a flow test tank built according to AMCA standard 210-85 [5]. That device allows a fan's static pressure head and volume flowrate to be measured at different operating conditions. A continuous performance curve was inferred from the measured data by fitting a polynomial curve through the measured results.

2.2 Sound Power Measurements

All sound power measurements were conducted by using the INCE plenum in a reverberation room as standardized in ANSI S12.11 1987 [6]. The INCE plenum is an acoustically transparent test chamber that allows the static pressure on the fan's discharge side to be controlled thus making it possible to measure the noise radiated by a fan at different operating points. More information on the plenum design may be found in references [6,7]. The sound power measurements were conducted using a single microphone on a rotating boom according to the substitution method. A calibrated electronic sound source (Brüel and Kjaer 4205) was employed as the reference noise source. Narrowband sound pressure spectra were measured using a Tektronix 2630 Frequency Analyser. The overall and 1/3-octave sound power levels were then computed by integrating narrow band spectra; that approach allowed pure tones to be removed from the fan spectra by a simple editing and interpolation procedure before the broadband levels were calculated. All fans were driven directly by a 4 pole, AC-motor. Hence, discrete tones resulting either from the operation of the motor or interaction of the blades and the cutoff appeared at harmonics of the fan speed. For more information on the pure tone removal, refer to reference [4]. The frequency analysis was performed in two frequency ranges (0 Hz to 2 kHz and 0 Hz to 20 kHz) so that at least 18 spectral lines would fall within each 1/3-octave band within the frequency range of interest (100 Hz to 10000 Hz).

3- DEVELOPMENT OF A SOUND POWER SCALING PROCEDURE

Any change to a fan design affects both its radiated sound power level and its aerodynamic performance. Since part of the change in sound level is directly related to the change in aerodynamic performance, it is difficult to identify the purely acoustical effect of a design change simply by comparing the noise radiated by two fans. Here, scaling laws were applied to the sound power level of

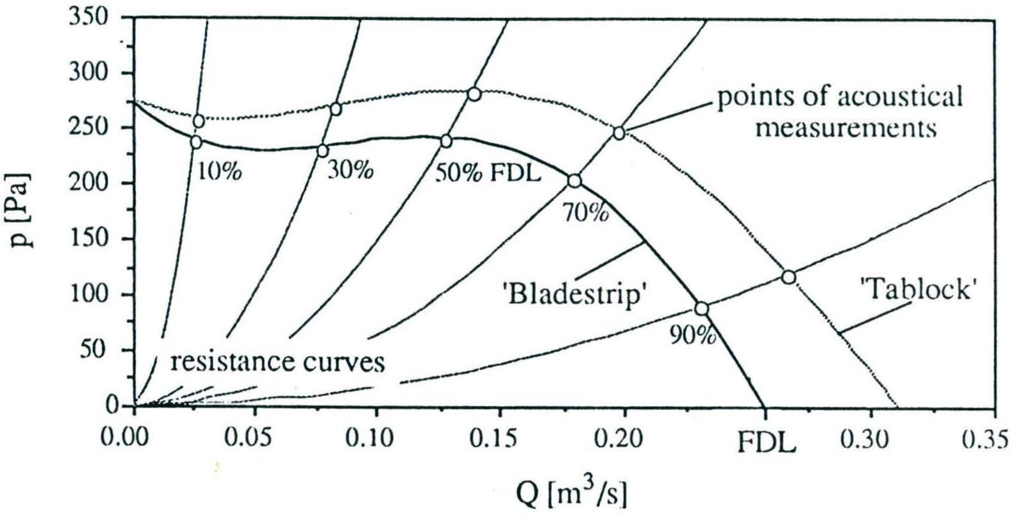


Fig. 1. Definition of operating points for the acoustical measurements.

one fan in order to predict its sound power level at an appropriate operating point on the performance curve of another fan to which the first fan was being compared. This procedure allowed the performance curve of one fan to be "mapped" onto the performance curve of another and so made it possible to identify which of several fans would meet a given pumping requirement while generating the least amount of noise.

The experimental procedure for determining the acoustical effect of design changes was based on measuring the sound power (at five operating points) radiated by each member of a family of fans that differed only in the value of one or, at most, two design parameters. The operating points were fixed by the fan having the lowest aerodynamic performance in the comparison set. That fan was tested at 10%, 30%, 50%, 70% and 90% of its free delivery flowrate (FDL). Matching operating points for the other fans in the set were chosen to be on the same system resistance curves as the operating points of the reference fan. As a result of that choice, extreme impeller geometries and speeds were avoided in the scaling as will be shown below. The overall A-weighted sound power level in the frequency range 100 Hz to 10 kHz was measured for each fan at each defined operating point. As an example of this procedure, Figure 1 shows the performance curves measured for a single fan housing operated with two different, commercially available blade types: 'Tablock' and 'Bladestrip.' It also shows the operating points at which acoustical measurements were performed.

The noise scaling procedure was always applied to the fan having the higher performance. First the operating speed and impeller diameter of a fan geometrically similar to the higher performance one, but which could operate at the same pressure, p_j, and flow, Q_j, as the lower performance fan, was determined. This transformation was achieved while holding constant the dimensionless flow coefficient, $\phi = Q/\pi D^3 N$, and the pressure coefficient, $\psi = 2p/\rho(\pi DN)^2$ [3]: i.e., the point-of-rating on the non-dimensional head-flow curve was maintained constant in this transformation. As a result, a new diameter, D_j, and speed, N_j, can be calculated according to:

$$D_j = D\left\{\left(\frac{p}{p_j}\right)^{1/2}\frac{Q_j}{Q}\right\}^{1/2} \qquad N_j = N\,\frac{Q_j}{Q}\frac{D^3}{D_j^3} \tag{1,2}$$

where: p and Q are the pressure and flowrate at the original operating point of the higher performance fan, D and N are the diameter and speed of the original fan, and ρ is the density of air. Since the operating points to be compared lie on the same resistance curve (which can be approximated by the relation $p=kQ^2$ [8], where k is a proportionality factor) the impeller diameter does *not* change in this transformation. Hence, by changing the impeller speed the higher performance fan can be run at the same operating point as the lower performance fan: i.e., when run in this way the two fans meet the same pumping requirement. Extreme scaling involving the impeller diameter fan is consequently avoided. Instead of actually altering the speed of the higher performance fan during the measurements, the following noise law [2] was used to *calculate* the A-weighted sound power at the new operating point:

$$W_{Aj} = W_A \frac{N_j^5}{N^5}. \tag{3}$$

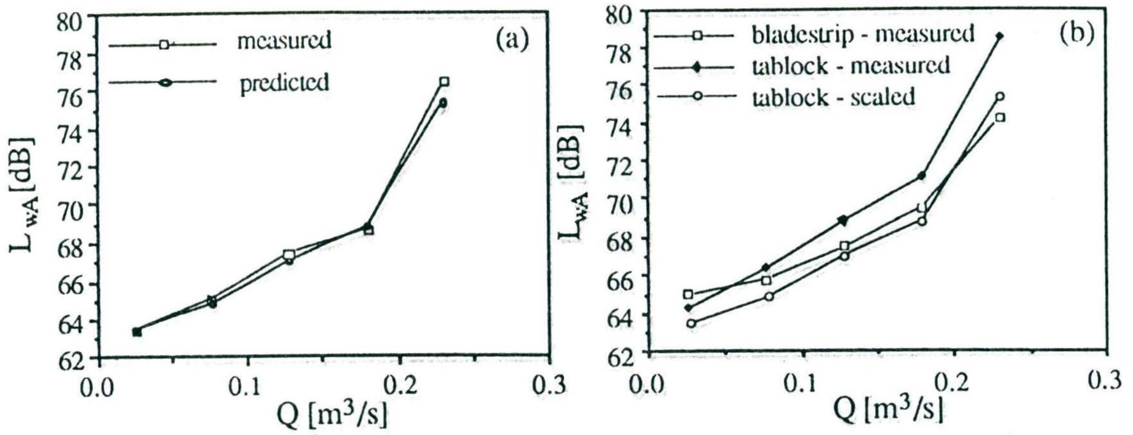


Fig. 2. (a) Scaling verification and (b) comparison of Bladestrip and Tablock fans.

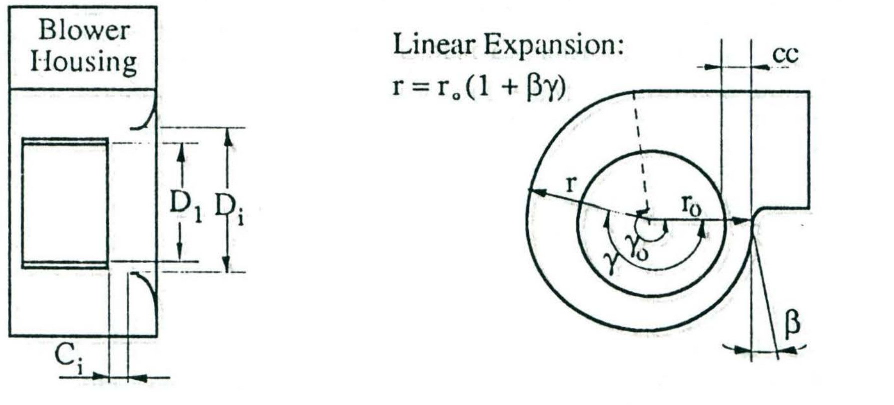
In equation (3), W_A is the measured A-weighted sound power of the original fan and W_{Aj} is the A-weighted sound power of the fan when run at the speed necessary to satisfy the *new* pumping requirement. Note that the *A-weighted* sound power was scaled.

4- VERIFICATION OF THE SOUND POWER SCALING PROCEDURE

In order to verify the scaling procedure, the speed of a Tablock fan (0.16 m impeller diameter) was reduced by using a line frequency controller so that sound power measurements could be performed at the new operating points. A comparison of the scaled, i.e., predicted, sound power levels and the sound power levels measured at the reduced speeds is shown in Figure 2(a). It can be seen that good agreement was achieved between the measured and scaled data, thus confirming that the proposed scaling procedure is reasonable.

5- IDENTIFICATION OF THE OPTIMUM LOW NOISE FAN DESIGN

To identify a low noise fan design, the noise generated by a family of fans each having an impeller diameter of 0.16 m and an impeller width of 0.1 m was studied. By using the scaling procedure described, it can be easily concluded which of several fan designs is quieter at a given pumping requirement simply by comparing the sound power levels of the various fans when scaled to yield the same performance as the fan having the lowest performance in the set. The geometrical parameters



Parameters Varied	Investigated Range	Optimum Values	
		Tablock	Bladestrip
Expansion angle (β) [deg]	6 - 12	8	8
Development angle (γ_o) [deg]	300 - 320	300	300
Cutoff Clearance (cc) [mm]	6 - 24	24	24
Radial Inlet Clearance (D_i/D_1)	0.94 - 1.08	1	1
Axial Inlet Clearance (C_i) [mm]	2.5 - 10	2.5	7.5

Fig. 3. Design parameters investigated and their optimum values.

that were varied in this investigation are summarized in Figure 3 along with the optimum values that were identified. Note that the effects of all the geometrical parameters were studied in combination with the two impeller blade types: Tablock and Bladestrip. A comparison between the two blade types indicated that the Tablock design generates less noise than the Bladestrip design at a given level of performance. Figure 2(b) shows the measured and scaled A-weighted sound power levels plotted *versus* the flowrates at the corresponding operating points for this case.

6- DEVELOPMENT OF A SOUND POWER PREDICTION PROCEDURE

It is clearly not feasible to use the above procedure to design fans for every application because of the extensive testing that would be required. Fortunately, it is possible to scale the optimum design to both smaller and larger flowrates in such a way that its acoustical properties are preserved. The procedure presented here allows the prediction of 1/3-octave band sound power levels in the frequency range 100 Hz to 10 kHz for various fans having different geometries and operating speeds, but all fulfilling a specified pumping requirement (i.e., a particular combination of pressure and flowrate). As a result, the overall A-weighted sound power level can then be calculated and plotted *versus* the corresponding point-of-rating, ϕ, and the aspect ratio, α (impeller width over diameter), to produce a "noise surface". The lowest sound power level can then be identified along with the corresponding fan geometry and required operating speed.

The procedure is based on measurements of the aerodynamic and acoustical performance of a low noise reference fan that is treated as a representative member of a family of geometrically similar fans. The aerodynamic data consist of a discrete set of operating points at which acoustical measurements were conducted using the reference fan. For each operating point, non-dimensionalized *generalized* flow and pressure coefficients, ϕ_g and ψ_g, were calculated according to equations (4) and (5) [3]:

$$\phi_g = \frac{Q}{\pi D^3 N \alpha}, \tag{4}$$

$$\psi_g = \frac{2p}{\rho(\pi D N)^2}. \tag{5}$$

When such a normalization is applied to the performance curve of the reference fan, a universal head-flow curve is obtained that characterizes the aerodynamic performance of any fan in the family [3]. A point on that head-flow curve is called a point-of-rating. Note that the conventional definition of ϕ_g has been modified here to allow for the impeller aspect ratio under the assumption that the flowrate delivered by a centrifugal fan is proportional to its aspect ratio at a constant static pressure. An experimental verification of this assumption will be presented below. The measured acoustical data consist of 1/3-octave, unweighted sound power levels for the reference fan, L_{wr}, measured at the operating points defined above. From the levels measured at a particular operating point, i.e., point-of-rating, a normalized spectrum can be derived that is characteristic of the sound power radiated by any fan in the family *operating at that point-of-rating*.

The equation that has been used to derive the normalized spectrum was proposed previously by Maling [3]: i.e.,

$$E = \rho c^2 D^3 M^3 \alpha g(\phi, s). \tag{6}$$

In equation (6), E is the sound power/Hz radiated by the fan, c is the speed of sound in air, $M = \pi DN/c$ is the impeller tip speed Mach number, and $g(\phi,s)$ is the normalized spectrum expressed as a function of point-of-rating, ϕ, and dimensionless frequency, $s = f/N$. Note that Maling's equation has been modified to account for the impeller aspect ratio by the inclusion of the factor α. The assumed dependence on α implies that the radiated sound power is proportional to the impeller aspect ratio when both static pressure and air flow per unit width are constant. An experimental confirmation of that assumed dependence will be presented below.

From measurements made using the reference fan, values of the normalized spectrum, $g(\phi_i,s_i)$, at the points of rating ϕ_i at which noise measurements have been made can easily be computed by first calculating E from the measured power levels and then solving equation (6) for $g(\phi_i,s_i)$. The values of s_i are calculated simply by dividing the 1/3-octave band center frequencies, f_c, by the fan speed at the point-of-rating being considered. To evaluate the normalized spectrum at points other than those measured, e.g., when a fan is run at a speed different than the reference fan, it is necessary to interpolate between, or extrapolate from, the measured data points. This can be achieved most easily by fitting a curve through $g(\phi_i,s_i)$ at each operating point. Since the sound power spectrum of a centrifugal fan tends to decrease by a constant number of decibels per decade of frequency, it is appropriate to fit a straight line to a doubly logarithmic presentation of $g(\phi_i,s_i)$ *versus* s_i.

Once the normalized spectra have been obtained at each point-of-rating, the impeller diameters, D_j, and speeds, N_j, can be calculated for a set of fans that all satisfy a given pumping requirement (i.e., a particular combination of p and Q) but which have different aspect ratios and are operated at different points-of-rating (the points-of-rating at which measurements were made using the reference fan). At a fixed point-of-rating, equations (4) and (5) may be used to derive relations between the reference values of D and N and new values that satisfy a particular pumping requirement: i.e.,

$$D_j = D\left\{\frac{\alpha}{\alpha_j}\,\frac{Q_j}{Q}\left\{\frac{p}{p_j}\right\}^{1/2}\right\}^{1/2}, \tag{7}$$

$$N_j = N\,\frac{\alpha}{\alpha_j}\,\frac{D^3}{D_j^3}\,\frac{Q_j}{Q}, \tag{8}$$

where the subscript j refers to the "new" fan geometry and the unsubscripted parameters are the reference values at a particular point-of-rating. In particular, note that the combination of p_j and Q_j fix the pumping requirement. The reference values of p and Q to be used in equations (7) and (8) obviously change depending on the operating point of the reference fan that is considered in this transformation. In addition, note that at each point-of-rating the aspect ratio of the new fan can be varied arbitrarily. Thus, at each operating point, pairs of impeller diameter and speed can be calculated for any given aspect ratio.

In this way ϕ_i and α_j fix the allowed values of D_j and N_j for a given pumping requirement. Given the value of N_j, it is straightforward to calculate values of s_j that correspond to standard 1/3-octave band center frequencies. The values of the normalized spectrum at these values of s_j for each point-of-rating may then be evaluated using the curve fits referred to above. Equation (6) can then be used to calculate the sound power per Hertz at each 1/3-octave band center frequency, from which, in turn, the 1/3-octave band sound power levels can be calculated. From the latter, the overall A-weighted sound power level may be computed. Finally, the noise levels calculated for different combinations of ϕ_i and α_j are then plotted *versus* these two parameters, the result being a "noise surface".

7- EXPERIMENTAL RESULTS

7.1 Reference Fan Data

The usefulness of the prediction scheme described above is obviously dependent on the identification of a low noise reference fan. Therefore, an extensive experimental program was conducted as described in the first part of this paper in order to define the optimum values of several geometrical fan design parameters. In addition, it was necessary to verify experimentally the assumptions regarding impeller width that are implicit in the scaling procedure. As noted above, the noise prediction procedure proposed here is applicable *only* to the broadband aerodynamic noise generated by turbulent flow through the fan; it specifically does not apply to pure tones generated by blade-cutoff interaction. The latter have been studied extensively elsewhere: see, for example, references [9,10].

7.2 Experimental Verification of Scaling with Respect to Impeller Width

As noted previously, the conventional definition of the dimensionless flow coefficient, ϕ_g, has been modified in this paper to account for the effect of different impeller aspect ratios on the aerodynamic performance of centrifugal fans. The modified definition reflects the assumption that the flowrate delivered by a fan is proportional to its aspect ratio at a constant static pressure. This assumption has been validated by making measurements with three fans having different impeller widths but constant impeller diameters (thus resulting in α values of 0.227, 0.436, and 0.574). The performance curves before and after normalization are shown in Figure 4; it can be seen that the proposed normalization produces an acceptable collapse of the data.

Recall that the noise equation (6) has also been generalized to account for different aspect ratios in the family of geometrically similar fans. It was assumed that the radiated sound power is proportional to the width of the impeller when both static pressure and air flow per unit width are constant. This assumption was experimentally validated by calculating the normalized spectra, $g(\phi_i,s_i)$, for three fans having different aspect ratios, but operating at the same point-of-rating. As shown in Figure 5, the collapse of the spectral data is significantly improved by the normalization procedure, thus suggesting that the proposed α-dependence is reasonable.

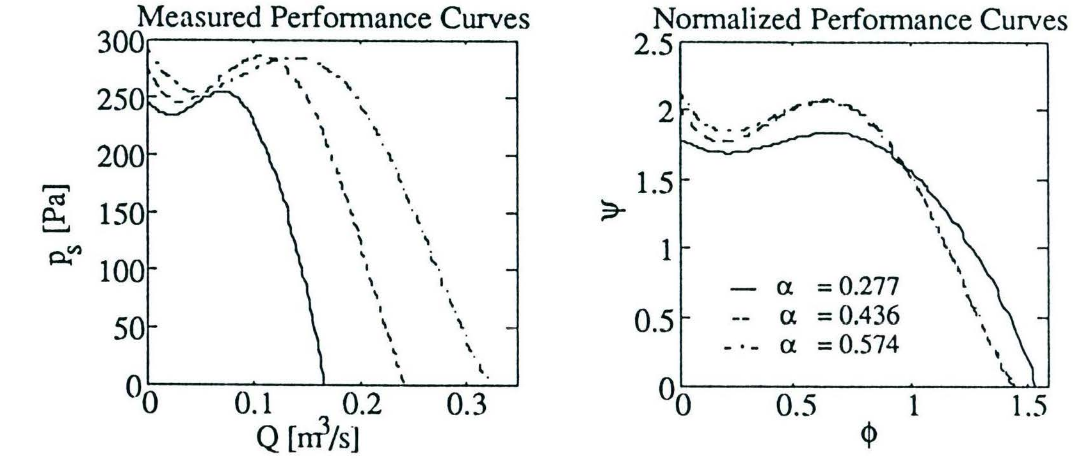


Fig. 4. A comparison of measured and normalized performance curves.

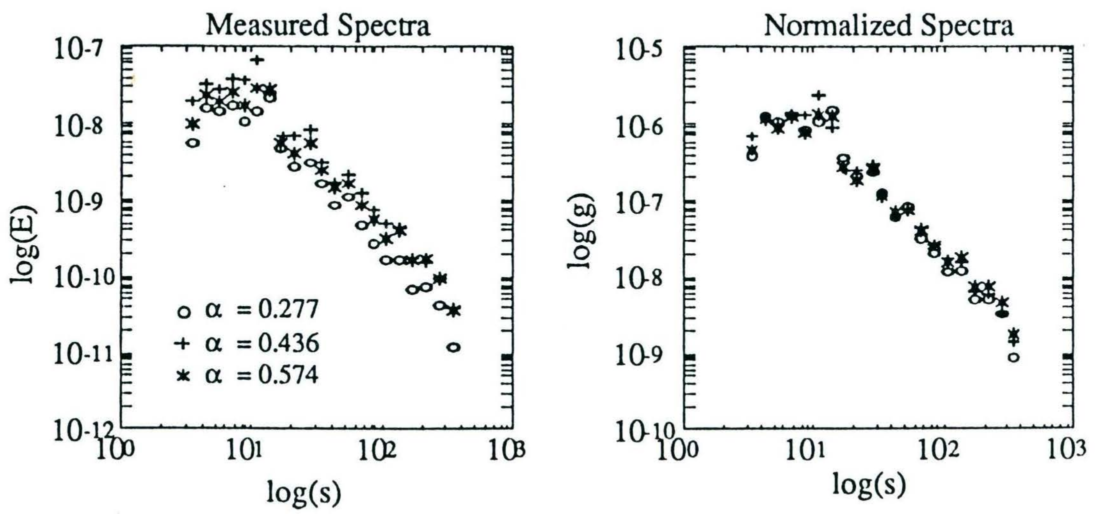


Fig. 5. A comparison of measured and normalized fan spectra.

8- CALCULATION OF NOISE SURFACES

Perhaps the most important use of the procedure described here is as a tool to aid in designing low noise fans to meet particular pumping requirements. As described above, the predicted A-weighted sound power level of all fans that are geometrically similar to the reference fan and that meet a specified pumping requirement may be plotted as a function of point-of-rating and aspect ratio. Curves of A-weighted sound power level *versus* point-of-rating and either fan speed or impeller diameter can also be generated. An example of a typical noise surface based on the reference data presented above is shown in Figure 6. That surface was calculated for an operating point characterized by a pressure of 300 Pa and a flowrate of 0.25 m^3/s. It is clear from this example that small aspect ratios are to be preferred from a noise point of view. More generally, studies of noise predictions for various pumping requirements have shown that fans having narrow, large diameter impellers running at low speeds often generate the least noise.

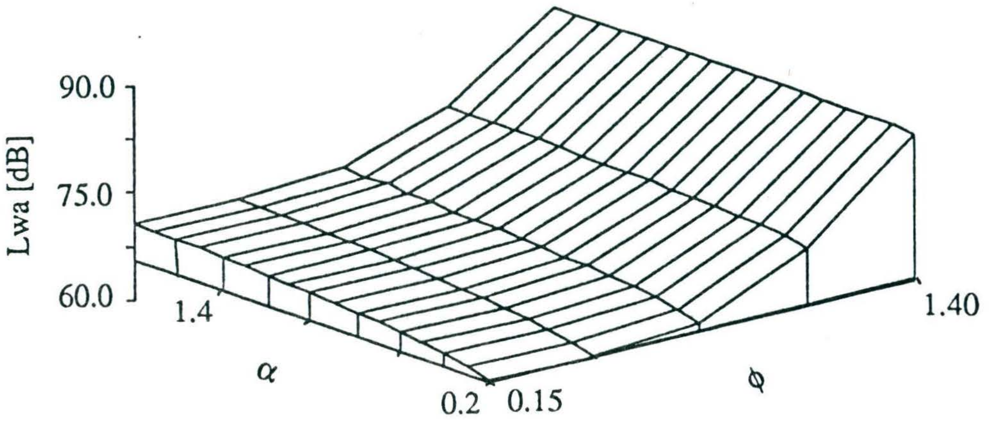


Fig. 6. A typical noise surface based on the low noise reference fan.

9- SUMMARY AND CONCLUSIONS

In the work described here, "fan laws" were used to compare the sound power generated by centrifugal fans having differing designs. In this way it has been possible to compare the noise produced by several fans and to identify unequivocally design-related noise differences. Based on this comparison scheme, a low noise fan design was identified: the optimum values of the design parameters are summarized in Figure 3. In addition, a technique has been developed that can be used to predict the 1/3-octave band sound power levels of centrifugal fans. The predictions are based on measurements of aerodynamic and acoustical performance made using a low noise reference fan identified in the first part of the present study. Given a particular pumping requirement, the prediction scheme can also be used to identify the lowest noise fan from a family of geometrically similar fans. Since the definitions for the dimensionless flow and pressure coefficient were generalized along with the noise equation to allow for impeller width variations, the prediction scheme can also account for the effects of varying the impeller width.

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