MODIFICATION AND EVALUATION OF AN AUTOMOTIVE COOLING AXIAL FLOW FAN NOISE PREDICTION MODEL

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ABSTRACT

The intention of this paper is to demonstrate the feasibility of a noise prediction model recently proposed for use in determining the sound pressure level spectrum of axial flow fans for an automotive cooling application. The predictions of the noise model, based solely on blade geometry and operating conditions, were compared with numerous empirical studies, one of which is presented here. The model is shown to be very effective in the absence of secondary sources of noise, such as blade corner details and the fan hub, while being totally ineffective when these sources are not negligible. Furthermore, in a fractional design of experiment, the model is used to predict the three most important geometrical parameters to consider in fan design. From the point of view of quietness, these parameters are overall fan radius, chord width and rotational speed.

SOMMAIRE

L'intention de cet article est de démontrer qu'un modèle de prédiction de bruit, produit dernièrement pour prédire le spectre de niveau de pression sonore d'un ventilateur employé dans une application de refroidissement dans l'industrie automobile, est réalisable. Les prédictions du modèle, établi seulement sur la géométrie des pâles du ventilateur et les conditions de fonctionnement, ont été comparées en vue de plusieurs études empiriques, dont l'une d'elle est présentée ici. On a montré que le modèle en éfficace en l'absence de bruit de sources secondaire tel que les détails des coins des ailes et le moyeu du ventilateur, cependant le modèle est totalement inefficace lorsque ces sources ne sont pas négligeables. De plus, dans une fraction du design de l'expérience, le modèle a prédit les trois paramètres les plus importants à considérer dans le design d'un ventilateur du point de vue niveau de bruit, le rayon du ventilateur, la largeur de la corde et la vitesse de révolution.

1. Introduction

Except for Gutin (1936), the most substantial theoretical investigation into aerodynamically generated sound is given by M. J. Lighthill [1]. In this historical paper, Lighthill derives a second order partial differential equation which characterizes the propagation of sound in a homogenous and isotropic medium. Many others, since then, have made significant contributions to noise theory, including Curle [2], who investigated, with respect to a sound field, the issue of solid, stationary boundaries , Morfey [3] and Longhouse [4], who researched the mechanisms of sound generation and Fukano *et al.*[5] who attempted to model turbulent noise generation. Two more recent investigators, specifically dealing with the topic of axial flow fan noise, are Quinlan [6], who discusses the application of active control as a means of reducing radiated noise and Lee *et al.*[7], who present an analytical model for predicting the vortex shedding noise generated from the wake of axial flow fan blades.

Kent Clark Bates developed a method of predicting axial flow fan sound pressure spectrums from simplified blade geometry and fan operating conditions. Based on the work in his thesis [8], a computer program has been produced at Siemens Electric Ltd. with the intention of applying Bates' noise prediction theory to engine cooling fans. The objective of this project is fan development time optimization through the integration of the computer code into the design process. It was felt that this course of action would prove to be effective through minimizing the time spent with prototypes and empirical evaluation. In order to accomplish this task, a two part plan was developed. First, a series of validation tests to substantiate the computer model. Second, a fractional factorial design of experiments (DOE) to identify key design parameters. The computer model is written in Microsoft Visual C++ 1.0, and is designed to run in a Windows 3.1 environment. It was the hope of management to be able to harmonize the technologies of computational fluid dynamics (CFD) and noise prediction to produce an economical axial flow fan that maximizes efficiency while minimizing noise.

2. Nomenclature

C_m	the m[th] complex Fourier coefficient of the radiation sound pressure relative to ambient pressure
d_m''	the m[th] complex Fourier coefficient of the second derivative of the fan displacement function
f	frequency [Hz]
f_0	fan rotational frequency [Hz]
F(f)	frequency response weighting function
G(f)	one sided mean-square pressure spectral density function $[N^2/m^4]$
$I(r_s)$	Intermediate integral in the calculation of C_m
m	Fourier coefficient index
n	number of defining fan blade cross sections
N	fan rotational speed [RPM]
Nb	number of fan blades
P_{ref}	decibel reference pressure, $20[\mu N/m^2]$
r	radius [m]
R_f,	receiver radial location [m]
s	subscript denoting the current element being considered
SPL	sound pressure level [dB]
V	relative velocity of air (m/s) ($V \approx 2\pi(N/60)r$)
w	blade chord width [m]
Z_f,	receiver axial location [m]
ϕ	angular coordinate [rad]
γ	blade pitch angle [rad]
θ	blade camber angle [rad]
ρ	blade radius of curvature [m]
ρ_0	ambient air density $[kg/m^3]$
υ	kinematic viscosity of air (m^2/s)

The basic parameters for an arbitrary fan blade cross section are below, in Figure 1.

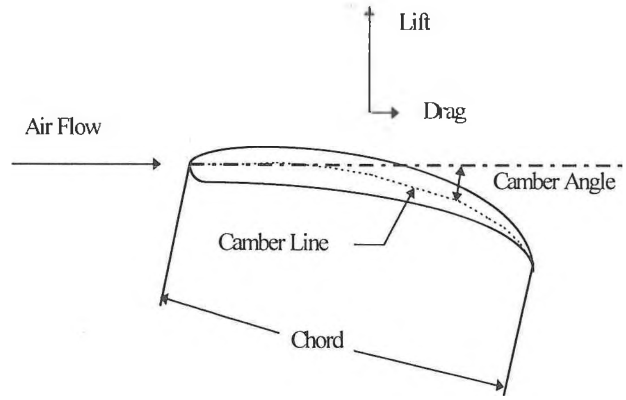

Figure 1 - basic airfoil cross section

3. Mathematical Foundation

Overview. To familiarize the reader with the basic concepts of Bates' noise prediction theory, the key equations are summarized below. The general idea of the mathematics is to calculate the non-zero complex Fourier coefficients that predict the blade passage frequency tone levels and then estimate the broadband components thereafter.

Basic Equations. It has been shown by Bates that the SPL within a frequency band having center frequency f_c, and bounded by a lower and upper frequency, f_1 and f_2 respectively, is given by:

$$(SPL)_{f_c} = 20\log_{10}\left(\frac{\sqrt{\int_{f_1}^{f_2} F(f)G(f)df}}{P_{ref}}\right) \tag{1}$$

It may also be shown that G(f) can be expressed as an infinite summation of complex Fourier coefficients, C_m, where $C_m = C_m(N, Nb, r, R_f, w, Z_f, \phi, \gamma, \theta, \rho, \rho_0)$. From the number of parameters that C_m is a function of, the reader may deduce that the main computational effort of the noise prediction model involves calculating these coefficients.

Constraints and Assumptions of the Bates' Original Theory. While the accuracy of the predictions is of the utmost importance, certain assumptions are made in an effort to reduce the mathematical complexity of the model (please note: in the modified theory used in the noise model being presented herein, the effect of some of these assumptions have been attempted to be minimized. In the following list, these shall be noted, for reference, in *italics*,

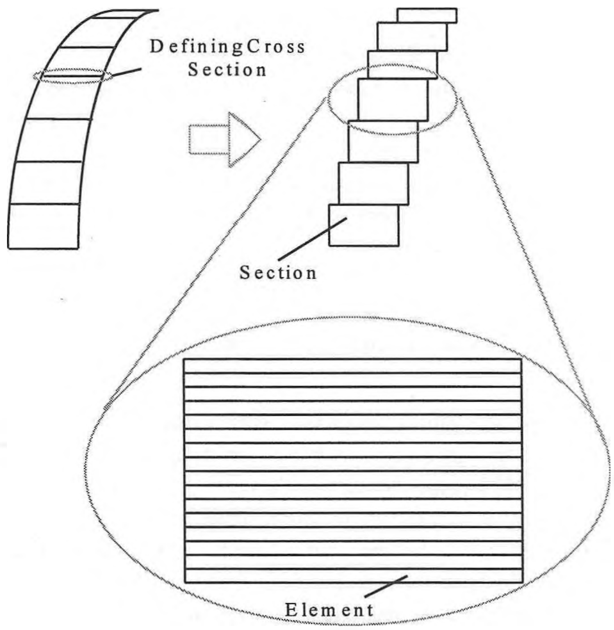

Figure 2 - schematic detailing the method in which Bates' constant fan blade assumption may be modified to more accurately resemble axial flow fans.

since they will be discussed again later). First and foremost, the noise prediction model is not wholly independent of experimental correlation. Bates neglects the significant applied force and stress source terms (the dipole and quadrupole terms respectively) of the wave equation in the development of his noise prediction theory. To partially compensate for these omissions, and in an effort to match experimentally measured autocorrelation functions, Bates adds a correctional term to his own theoretically derived autocorrelation function. The second approximation assumes a particle of air in the path of the blades is displaced only in the interval of time in which it is in contact with the fan blades. The third maintains an equivalent air displacement pattern may be generated by an infinite array of acoustical monopole sources located evenly in the plane circular band bounded by the extremities of the fan blades. *Fourth, the cross section of every blade, at any radial point, is constant. It is in the shape of a circular arc and possesses fixed values of pitch, camber, chord width and radii of curvature.* Fifth, all blades are rigidly connected to a central hub, but the effects of the hub as well as any rivets, blade thickness, blade corner detail or blade vibrations are neglected. Finally, the predicted field sound pressure is a stationary random process.

Modifications to Bates' Theory. While little may be done about many of the assumptions, a remedy exists for that of the fourth listed above . Bates' theory was modified to allow an arbitrary number of defining fan cross sections to be input and a representative fan be constructed from sections whose parameters are the average of the bounding cross sections (Figure 2).

Furthermore, Bates employs a circular arc blade cross section in his model, so its radius of curvature is readily available. However, the cross section of a fan blade at Siemens Electric is a C4 airfoil and therefore this parameter is non-existent. Nevertheless, the camber line contour equation used in computing the airfoil shape of each defining cross section is based on a circular arc and is a function of the blade camber angle. It is therefore postulated that the cross sectional shape may be modeled after this base curve. Please note: because the noise prediction model is intended for use with a fan in the design stage, the blade camber angle, for each cross section, is easily obtainable. Therefore, the camber angle substitutes as an input parameter and the blade element radius of curvature is calculated from each separate value.

Superposition of Blade Elements. The above modification naturally necessitates the need of a method for the effects of all the blade elements to be combined mathematically. Since the complex Fourier coefficients are calculated by integration in the radial direction, this allows for the superposition of the noise contributions of each fan blade section (for the purposes of the NOISE application, each section was further broken down into sixteen smaller elements). It may be shown that the mth complex Fourier coefficient may be rewritten as :

$$C_m = \frac{Nb \cdot \rho_0 \cdot f_0^2}{2} \sum_{j=1}^{n-1} \sum_{l=1}^{16} \left[r_s^2 \cdot d_m'' \cdot I(r_s) \cdot \Delta r_s \right]_{j,l} \qquad (2)$$

Using equation (1), and other principals described by Bates, the axial flow fan sound pressure spectrum may now be predicted for realistic fan blade geometry.

4. Validation Test Methods and Evaluation

Main Equipment. In addition to the regularly used noise measurement equipment, the following special items are to be noted:

- NOISE application
- the Volvo 390F-1.3.0 fan and its associated table of geometrical parameters.

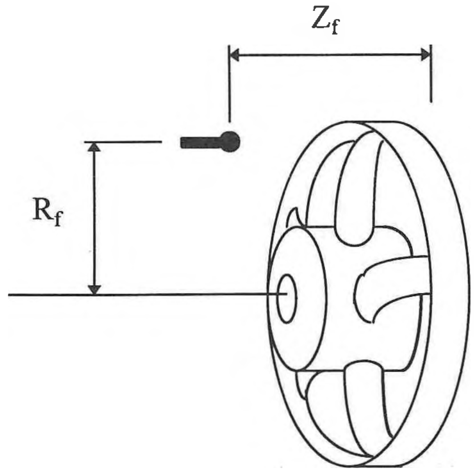

Figure 3- Typical placement of microphone, including measuring reference points on the fan

General Procedures, Experimental. ISO 3744 describes a procedure that may be used for measurements in the near field. A number of different car manufacturers make reference to this standard but generally prescribe noise tests be done at a distance of 1m (far field). However, because of the limitations imposed by the assumptions listed above, this distance, although desirous, could not be used (see section 5). The experimental procedure observed was as follows:

1. Set up the apparatus in the standard configuration for noise measurement of fans, as detailed in Figure 3.
2. Set the initial placement of both microphones in line with the fan axis of rotation at a specified radial (R_f) and axial (Z_f) distance.
3. Adjust the power supply until the desired rotational speed is set.
4. Measure both the overall noise and discrete frequency noise levels simultaneously.
5. Save all data electronically.
6. Plot the data.
7. Repeat steps 2-6 for each axial and radial position required and for each desired speed.

General Procedures, Theoretical: The theoretical procedure observed was as follows:

1. Start the NOISE application
2. Input all the geometrical parameters.
3. Input all the ambient condition parameters.
4. Guess at the values of the correlational parameters, Z and ζ.
5. Plot the 'A' Weighted SPL versus Frequency graph.
6. Compare the broad-band noise levels to that of the experimental results.
7. If the broad-band noise does not correlate closely with that of the experimental results, repeat steps 4-7.
8. Note the values of the overall noise level, the tonal frequencies and the correlational parameters.
9. Save the information to a file.
10. Repeat steps 2-9 for each axial and radial position required and for each desired speed.

Experimental Validation of the NOISE Application. The axial flow fan given above has been tested, at speeds of 1800 and 2400 rpm. Noise levels were measured at radial distances of 0 - 20 cm, in increments of 5 cm, at two experimental axial distances of 10 and 15 cm (with respect to the NOISE program, these two distances are 6 and 11 cm). The results are summarized above in Table 1.

5. Discussion

The Correlation Experiment, Comparison of Relative Error. The error calculations, relative to the experimental SPL results, show three distinctive trends. The first indicates the NOISE application's prediction accuracy increases with a decreased receiver axial distance. This trend agrees well with Bates' constraint of a near field

Speed	(Rf, Zf)	Experimental Overall SPL	NOISE v.2 Overall SPL	Relative Difference
1800	(0,.06)	83.7	-80.791	196.524
1800	(0.05,.06)	83.6	49.018	41.366
1800	(0.1,.06)	82.7	79.855	3.440
1800	(0.15,.06)	83.3	84.889	1.907
1800	(0.2,.06)	82.4	87.822	6.580
1800	(0,.11)	75.9	-97.074	227.898
1800	(0.05,.11)	75	8.491	88.679
1800	(0.1,.11)	74.8	36.524	51.172
1800	(0.15,.11)	74.5	63.575	14.665
1800	(0.2,.11)	74.2	69.816	5.908
2400	(0,.06)	91.9	-76.811	183.581
2400	(0.05,.06)	90.1	56.599	37.182
2400	(0.1,.06)	90	87.433	2.852
2400	(0.15,.06)	90.5	92.513	2.224
2400	(0.2,.06)	89.3	95.460	6.898
2400	(0,.11)	81.7	-86.001	205.264
2400	(0.05,.11)	80.9	16.090	80.111
2400	(0.1,.11)	80.9	44.205	45.359
2400	(0.15,.11)	81.3	71.282	12.322
2400	(0.2,.11)	81.9	77.547	5.316

Table 1 - Results of the validation experiments

Factor Name	Low	High	ALIAS
Number of Profiles (P)	6	11	
Stagger Angle [°] (ζ)	55	75	
Stagger Taper (T_ζ)	NONE	Increasing	
Camber Angle [°] (θ)	15	30	
Chord Width [mm] (W)	30	80	
Chord Taper (T_W)	NONE	Decreasing	
Number of Blades (B)	2	11	$P\zeta T_\zeta\theta$
Rotational Speed [RPM] (N)	2000	3000	$PT_\zeta WT_W$
Overall Radius [mm] (R)	280	460	$T_\zeta\theta WT_W$

Table 2 -- Factors and settings for the parametric study on noise

prediction model, where he states "the theoretical solution was concluded to be invalid at distances greater than approximately one fan radius..."[Bates p. 79]. It should be noted that the values used for Z_f in the NOISE simulations was not equal to the experimental value, since, as Bates states in his thesis $Z_f \neq Z_{exp}$ [Bates p. 77]. For these experiments, Z_{exp} is calculated as $Z_{exp} \approx Z_f$ + hub thickness.

The second trend implies the NOISE model is most acceptable for radial receiver locations of approximately 75% of the maximum radial distance. This does not collaborate well with Bates' results who found adequate prediction accuracy along the entire width of the fan blades. The most probable source of error in this case is the simplification wherein the blade shape is modeled as a circular arc and the effects of the blade thickness and corner details are neglected. Since this trend held true for all fans tested, and therefore different blade section geometry, the only other change is the relative velocity of the air passing over the blade. Considering the blade sections nearer to the blade tip encounter greater relative velocities than those closer to the hub, the flat plate, circular arc blade section assumption must only be valid above a certain threshold speed value.

The final trend is the accuracy of the model increases with increased fan rotational speed. It should be noted that this observation again strongly suggests that Bates' circular arc model is only valid above a certain speed, when dealing with airfoil cross sections.

Careful investigation of the Reynolds number of the fan blade indicates the flow for the innermost fan cross sections is almost certainly laminar while that of the outermost sections is likely turbulent. The Reynolds number for an airfoil is calculated as:

$$\mathrm{Re} = \frac{Vw}{\upsilon} \tag{3}$$

Since Bates maintains "acoustic source distributions...are created by both blade geometry and turbulent flow" [Bates, p. 16], it is expected the threshold relative velocity value will prove to be that at which the transition Reynolds number occurs ($Re_{transitional}$. $\approx 10^6$ [9]). This has yet to be proven.

It should also be noted that the trends exhibited by the relative error results were independent of the type of fan tested. That is to say, the relative error of the NOISE prediction model is independent of blade geometry or sweep.

6. Fractional Factorial Design of Experiment

Overview. As stated above, $C_m = C_m(N, Nb, r, R_f, w, Z_f, \phi, \gamma, \theta, \rho, \rho_0)$. From the point of view, however, of fan blade design criterion, obviously not all of these parameters may be influenced. After some discussion, nine parameters were chosen to be investigated (Table 2).

Method. The design of experiment (DOE) was carried out as an eighth replicate of a 2^9 factorial design. The design followed that recommended by J.C. Young [10]. As a 2^{9-3} design, three design parameters were aliased (see Table 2) with three extremely unlikely four factor interactions. This translates to, as described by Young, a resolution IV experiment and as such, some two factor interactions will also be confounded with some other (hopefully negligible) two factor interactions (the terms "aliased" and "confounded" are statistical terms and are meant to convey the idea that the results of the experiment could be attributed either of the factors or interactions the results are confounded or aliased with). The various levels of the parameters were estimated to represent a fair spread of realistic design parameters, as experienced at Siemens Ltd.. Table 2, below, describes the experimental set-up.

The assumptions made in the experiment were as following:

1. All interactions greater than two are unlikely, and as such are ignored.

2. All interactions with the number of design profiles are irrelevant with respect to noise generation (these will only affect the prediction accuracy), and as such are ignored.

3. Ignore the interaction between the overall radius and the rotational speed. This is reasonable since

this interaction shows the effect of tip speed, which is shown by N alone. The change in the overall radius will exhibit the effect of the hub radius (which was held constant at 150mm).

It can be proven that, from the method in which this experiment was set up, the only confounded two factor interaction is between N and R (interaction NR) aliased with the P and θ ($P\theta$) interaction. Since both interactions are being ignored for this study, this experiment becomes in reality a resolution V experiment.

All sixty-four experiments were run on the NOISE application over a course of three days. The receiver axial and radial locations were kept constant at 75% of the fan radius and 6cm respectively. This was determined to be the ideal location in terms of the accuracy of the NOISE program, as detailed above. Furthermore, the autocorrelation parameters were also selected on the basis of past experience.

7. Results

The factors, in order of importance, were found to be: R, W, N, T_W, B and $T_\zeta\zeta$. This is shown in Table 3 and Table 4:

ANALYSIS of:	Std. ERROR	95% CONFIDENCE INTERVAL	
R EFFECT	0.761	11.808	14.922
W EFFECT	0.761	10.492	13.605
N EFFECT	0.761	8.708	11.822
T_W EFFECT	0.761	-8.496	-5.383
B EFFECT	0.761	5.177	8.290

Table 3 -- Statistical analysis of the major factors

ANALYSIS of $T_\zeta\zeta$ EFFECT	Std. ERROR	95% CONFIDENCE INTERVAL	
ζ (no T_ζ) Effect	1.076	-7.995	-3.592
ζ (with T_ζ) Effect	1.076	-4.096	0.307

Table 4 -- Statistical analysis of the major two factor interaction

From the above tables, we see the need for two definitions: Std. ERROR and 95% CONFIDENCE INTERVAL. The first is an estimation of the standard error for the variable, which is a measure of the degree to which an effect varies from the mean. The last in the list is the expected range of *change*, with 95% certainty, of the overall noise level, as predicted by NOISE, at the specified position, if the factor is varied from its low level (-) to its high level (+).

8. Conclusions

Based on this experiment, from the point of view of noise reduction, and in order of preference:

1. A smaller radius is preferable to a large radius
2. A shorter chord length is preferable to a longer chord length
3. A slower rotating fan is preferable to a faster rotating fan
4. Having a decreasing chord length at greater radial distances is desirable
5. A smaller number of blades is preferable to a greater number of blades
6. A constant, high stagger angle is preferable to one that is low and increases linearly in the radial direction.

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