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A New Approach to Assess the Quality of Small High-Speed Centrifugal Fans Using Noise Measurement*

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The requirements for a special approach for the quality assessment of small high-speed centrifugal fans are outlined and a new parameter designating the noise levels from the product in comprehensive form will be discussed and described as a criterion for such quality assessment.

By applying techniques of signal processing and condition monitoring, the sources of the vibration and noise in different sections of the product can be identified, then the noise from each source from different components can be determined. Using this criterion, more aspects of the quality of the products can be assessed and suggestions to improve the quality of the products can be made. Finally, the assessment of a number of vacuum cleaner motor/fan units available in the commercial market will be presented and compared with conventional specifications. It will be shown that the new parameter provides a more useful indication of appliance quality.

Keywords: Noise measurement, Centrifugal fan, Quality assessment, Dynamics of rotating machines

INTRODUCTION

The trend in current designs of household and industrial appliances is to make the products more powerful, lighter, smaller and preferably quieter. Quality assessment of such mass-produced products should be much more than testing to make sure the machine operates properly. The quality and quantity of the vibration and noise from the products must be considered. Modern design resulting from competition has tended to lead to the use of machines of a smaller weight per unit power output and hence increased electric and magnetic loading. Especially for products like a vacuum cleaner that must fulfil a certain duty, high-speed operation must be employed to reduce the size. This leads to a relatively thin frame, higher flux densities and therefore magnetic saturation, the need for increased cooling, all leading to greater problems of noise and vibration (Yang and Ellison, 1981). In order to cope with the increasing demand for quietness of operation of electric motor/fan units, many

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countries and the international organisations have developed national and international standards dealing with noise measurement and noise limits of electric machines (Yang and Ellison, 1985). These standards can meet the general requirements to set down the limitation of noise and vibration levels for a certain type of products. For comparing and assessing the individual comprehensive characteristics for different kinds of machines, these standards do not provide any general parameters or enough information. In the market, all the products must meet the national or international standards, but their individual specifications and performance are likely to be quite different. There is a need for more technical indicators to let the designer and user judge which of the products is the best choice for their special case. For a small high-speed centrifugal fan, the aerodynamic noise will dominate. This paper will focus on this case.

Vibration and noise measurements, machinery condition monitoring and diagnostics methods have been well developed for both the detection of changes in machine condition and the detection and identification of incipient defects. By applying these techniques, the sources of the vibration and noise in different sections of the products can be identified (Chen *et al.*, 1989).

Many approaches to predict and present the noise levels for certain type of products have been developed (Neise, 1975; 1976). It is difficult and impractical to predict and present the response levels of small high-speed rotating machines and fluids to force fields that are random in time and space with customary methods and specifications. The present approach tries to define a general parameter to present the noise or vibration levels for various types of small high-speed rotating machine. This parameter will be used as a criterion to compare and assess the quality of centrifugal fans in this paper.

BACKGROUND

When installed, a centrifugal fan is required to deliver a specified airflow against a resistance provided by the fluid transfer system. Thus the fan unit's operating point (i.e. the performance point specified by the combination of the airflow rate and the static pressure rise that the unit generates) is determined by the installation. It is of interest to be able to identify the unit that generates the least noise while meeting a specified airflow pumping requirement. The aerodynamic performance of a centrifugal fan unit is usually specified in terms of its pressure characteristic: i.e., the static pressure rise the fan unit develops expressed as a function of the flow that the fan delivers. The impeller and housing design, and the fan operating parameters in turn, determine the pressure-flow characteristics.

Early papers on scaling fan noise were presented by several investigators and they performed a dimensional analysis for the fan noise and obtained a general relationship among non-dimensional parameters (Neise, 1975; Bommes *et al.*, 1995).

$$P_w = P_\infty f(\mathrm{Re}, M, \mathrm{St}, x_i/D, \varphi, k) \qquad (1)$$

where P_∞ is a suitable reference pressure. The spectral noise component P_w is defined differently for tonal noise and the random noise. For tonal noise, relatively easy and practical methods have been developed by the former researchers to remove and reduce them. For random noise, i.e. aerodynamic noise, the ratio of sound power to some relevant reference power may be expressed as following product of terms:

$$(W/p) = (k)(\mathrm{Re}^\alpha)(M^\beta). \qquad (2)$$

A rearrangement, introducing the reference values and taking ten times the base-10 logarithm of both sides of the expression, results in the following model for predicting the wideband sound power level, in decibel, from a centrifugal fan:

$$L_w = A^* + B^* \lg(M). \qquad (3)$$

The coefficients A^* and B^* in Eq. (3) are constants for a given system geometry as long as there are no changes in the air inflow conditions (such as flow distortions and turbulence) or in any downstream

flow obstructions. The product $B^* \lg(M)$ is the only term that depends on fan tip speed. The coefficient B^* is the composite Mach Number exponent β. This exponent depends on the order of the sound source and takes into account the fact that the radiation efficiency of all elementary aeroacoustic sources increases with the flow velocity in a manner that depends on the order of the radiation field.

The A^* term is the sum of all those sound power level terms that do not depend on velocity. For convenience, the non-constant terms will be grouped and designated as "specific noise area level" L_{gs}, which takes into account the size, configuration, and aerodynamic loading of the system, and "specific sound transformation level" L_{us}, which is a measure of the combined effects of airflow distortions, separation, and turbulence on the transformation of the aerodynamic energy into acoustic energy, respectively.

Much research has been carried out regarding the prediction of noise levels from the products, using the parameters of size, configuration, loading, operating speed, etc. It shows how the noise levels are related to such parameters should be dependent on the quality of design. In order to assess the quality of the centrifugal fan products, many dimensionless parameters, like efficiency, coefficient φ and ψ, have been developed. In practice, it has been found that they are inadequate for numerical evaluation of the important characteristics of the centrifugal fan. A given volume (v) and a given pressure increase (Δp) can be produced by various fans which are widely different in their dimensions and noise levels. The product having the same or smaller size and the same or larger capacity, which produces less noise, is a better one. The new approach assessing the quality of the fan products should consider all these aspects comprehensively.

STATEMENT OF APPROACH

From the former research results mentioned above, it has been shown that the noise and vibration levels from fan units are related to the properties of the medium, effect of flow field (turbulence, vortex shedding, etc), effect of size and configuration of the flow system and effect of aero-acoustic source types. From the view of assessing the quality of the product, besides the noise and vibration, only the capacity (input power, output power) and size will be of concern. In order to increase the capacity and decrease the size of the product, the rotating speed must be increased to very high values. For a small high-speed fan unit, the aerodynamic origins will play a major role in producing the noise and vibration from the product. The aerodynamic noise is related to the rotating speed, the shape of the blade and design of the flow system. To get a better product, smaller size, larger capacity, and of course quieter operation, the proper shape of the blade and configuration of the flow system must be chosen. Rotating speed and size should be optimised as well.

For convenience, the noise and vibration levels will be described quantitatively by statistical energy analysis. "SEA has been described as a point of view in dealing with vibration of complex structures, and as such it employs a series of analytical and experimental 'methods', most of which predate the identification of SEA. The view-point is statistical because the system under analysis is presumed to be drawn from populations with random parameters; energy is the independent dynamical variable chosen because, by using it, distinction between acoustical and mechanical systems disappear; and analysis emphasises that SEA is an approach to problems rather than a set of techniques as such" (White and Walker, 1982). The energy density of the noise from the product can be calculated using SEA. It is very useful to assess which type of product is better to employ by using the energy density of noise and vibration per unit of output power of unit of size, i.e. defining

$$R_n = \frac{E_n}{P_{out}/V_s} \tag{4}$$

as a non-dimensional parameter to indicate the noise level for small high-speed rotating machines. Here E_n is energy density of noise from the

product, which can be calculated with the expression $E_n = \bar{p}^2/\rho c^2$, $\bar{p}$ is pressure of acoustic wave in time average, ρ is the density of the air and c is the speed of sound; P_{out} is output power from the product, for centrifugal fan $P_{out} = \Delta p v$; V_s is volume of the product, for centrifugal fan $V_s = \pi b d^2/4$.

For a centrifugal fan $\varphi = v/u\pi d^2/4$, and $\psi = \Delta p/(\rho/2)u^2$.

Rearranging Eq. (4):

$$R_n = 2bc^2\bar{p}^2/\varphi\psi\rho^2u^3c^2 = kbp^2/\varphi\psi u^3, \qquad (5)$$

where $k = 2/\rho^2c^2$, a constant coefficient.

Investigating the desirable features of a centrifugal fan (Eck, 1973):

(1) Maximum efficiency. This requirement is satisfied by a design with a suitable value of the coefficient σ, and experience has proven the higher the efficiency is, the less the noise produced.
(2) Minimal noise generation. Maximum value of ψ and low peripheral velocity.
(3) Minimum wear in operation with dust-laden gases. Maximum value of ψ.
(4) Large capacity. Maximum value of φ.
(5) Maximum capacity with minimum size, cheapest design. The product of $\psi\varphi$ must be as large as possible.

Analysing Eq. (5), a conclusion can be reached. The higher efficiency the product has, the less noise produced, namely the pressure of the acoustical wave $\bar{p}$ is smaller. So R_n decreases when the efficiency increases. The other four desirable features should be satisfied by a design with a maximum value of $\varphi\psi$. Therefore to reach the desirable features the design must have a small value of R_n. Above all, if the products with smaller dimension (b is smaller) operating at higher speed (u increasing) produce the same noise level, R_n decreases. The parameter R_n is a comprehensive description of all desirable features of a centrifugal fan and using R_n as a criterion to assess the quality of the product agrees with the trend of the modern design in which the smaller, powerful, high-speed and quieter products are preferable.

EXPERIMENTAL WORK

An appropriate experiment and analysis system is very important for getting accurate results. In order to cover all the audible frequency range from 20 Hz to 20 kHz and minimise the influences from the transducers, the appropriate microphone and data acquisition systems were employed.

Experiment System Arrangement

In order to achieve a greater amount of technical knowledge, a PC-based data acquisition system DigiS was used to digitise and record the signals as data files in real-time. The data files are available to transfer to any other analysis software which are available in PC computer systems and in workstations. It is very convenient and important for carrying out further accurate analysis and getting more technical information from the experiments. The arrangement of this system is illustrated in Fig. 1. To avoid the influence of environmental noise and reflections of the actual noise from the motor/fan units, the experimental system was assembled in a semi-anechoic chamber.

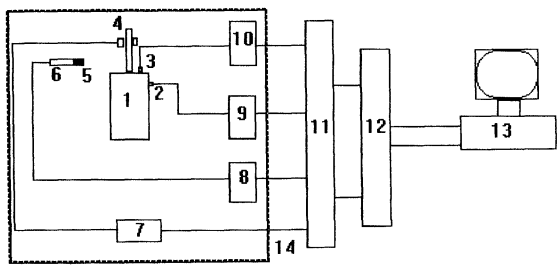

FIGURE 1 Arrangement of the experiment system where (1) tested motor; (2,3) accelerometers; (4) probe; (5) microphone; (6) pre-amplifier; (7) tachometer; (8) power supply; (9,10) charge amplifiers; (11) multiple-channel tape recorder; (12) connector box; (13) PC-computer with software; (14) anechoic chamber.

Analysis Method

A very wide range of methods can carry out analysis of noise and vibration signals. The most commonly used method of analysis is separation of the signal into its components at a range of frequencies or frequency bands. All frequency analysis eventually depends on Fourier's original approach, which starts from the basic assumption that any waveform, however complicated, whether repetitive or not, even containing discontinuities, can be

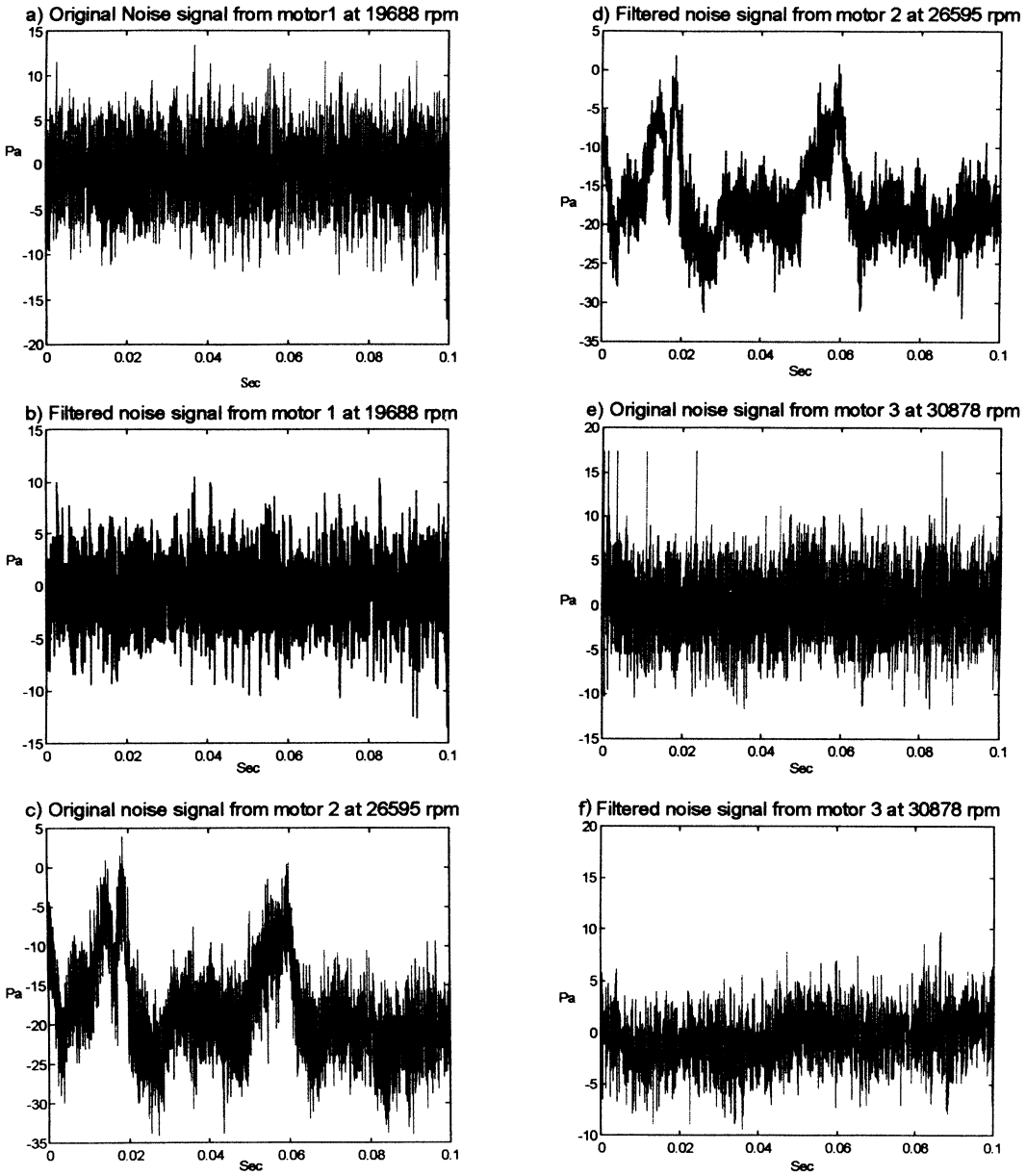

FIGURE 2 Noise pressure level signals from three different motors where (a), (c) and (e) are original noise pressure signals from motor 1, 2 and 3 respectively; (b), (d) and (f) are noise pressure signals after band-pass filter from motor 1, 2 and 3 respectively.

TABLE I The values of R_n for three motors

	No. of stages	Diameter of impeller	Width of impeller	No. of blades	Efficiency (%)	Noise level (dB)	R_n
Motor 1	2	128	10	5	30	58.81	0.1237
Motor 2	1	116	13	7	34	62.15	0.0738
Motor 3	1	109	14	9	39	65.03	0.0711

described or generated as the sum of a series of sine and cosine waves of different amplitudes, frequencies and phase. By applying the techniques of diagnosis and monitoring, the sources of the noise can be identified. Then the amount of the noise from the fan unit can be extracted from the total noise signal.

Experiment Results

The motors under test were several 1000 W, 2-pole and single-phase ac series motors, which were operated in the range 20,000–40,000 rpm. The noise signals were measured and analysed by methods described above. The original signal obtained directly from the products could be analysed to get correspondent PSD plots, then the proper frequency of the band filter can be chosen to eliminate the noise components that were not from the fan unit. In this case, a 0–5000 Hz band pass filter was employed. The experimental results are shown in Fig. 2 and the results of R_n for different fans are in Table I.

DISCUSSION AND CONCLUSION

From the above results, it will be found that the value of R_n can show the quality of the product comprehensively. Compared to conventional specification, it will provide more useful indications. When choosing a product, at first you are looking for a product that can meet the duty requirements, i.e. a certain capacity. Then the design quality that are designated by a suitable efficiency and noise level must be considered. Due to the commercial competition and requirement from the modern society, the small size of product is very important, as well. In most cases, the values of the parameters showing different features are independent and maybe conflict with each other, so it is difficult to compare the different products with such parameters. R_n combines all the desirable features and gives a quantitative value. It is possible and convenient to assess the quality of the centrifugal fan products by the value of R_n.

Acknowledgements

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NOMENCLATURE

A^*	constant for a given geometry
b	impeller width, m
B^*	a coefficient, the composite Mach number exponent β
c	speed of sound, m/s
D	impeller diameter, m
E_n	energy density of noise from the product
L_{gs}	specific noise area level
L_{us}	specific sound transformation level
K	isentropic exponent
M	Mach number
P	pressure, Pa
Δp	pressure increase across the fan, Pa
$\bar{p}$	time average value of acoustic wave pressure, Pa
P_w	spectral noise component
P_∞	reference pressure
Re	Reynolds number
St	Strouhal number
V	volume flow, m^3/s
w	wideband sound power, W

x_i	coordinates of the microphone position, m
α	Reynolds number exponent
β	Mach number exponent
ρ	density of air, kg/m^3
φ	flow coefficient
ψ	pressure coefficient
σ	speed coefficient

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