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FAN NOISE AND ITS PREDICTION

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SUMMARY

When designing ventilation systems, one is often unable to obtain very specific details either of the fan to be used or of its noise as measured by the fan manufacturer under controlled conditions. A number of empirical prediction formulae have been developed in the past and are frequently used by system designers and consultants in the early stages of design. A number of these calculation methods are presented and discussed in this paper.

INTRODUCTION

The control of noise is playing an increasingly important part in the process of ventilation system design. In factory installations ventilation fans can be a serious source of community disturbance. Industrial process fans, associated with heat exchangers, dust separators, furnaces, etc., can even be a source of health hazard to personnel working in their close proximity. In this paper we are concerned primarily with the use of fans in building ventilation systems although many of our comments will be pertinent to the industrial use of fans also.

In building ventilation systems fan noise is heard generally as a low frequency "rumble" which emanates from the diffuser outlets and through the walls of the ventilation ducting. The low frequency nature of fan noise makes it a difficult and expensive problem to solve in retrospect. If the problem is recognised in the preliminary design stage, however, steps can be taken to tackle it - by such action as the provision of adequate duct lining, the insertion of sound attenuators, the reinforcement of duct walls, and in some instances by arranging to serve critical spaces by a separate low pressure fan unit.

For those of you who are not familiar with the terms used in describing noise, much of the essential information is contained in the first few pages of Chapter 31 of the ASHRAE Guide and Data Book (1). We shall refer to this document later on; it makes a number of very pertinent comments about the role of the fan in ventilation system design.

FAN NOISE

Fan noise is aerodynamic in origin. It derives from the turbulence which exists in the air flow through the fan. When turbulent flow interacts with a solid surface, or when turbulence, in the form of a wake, is generated by a solid surface, time-varying changes in the momentum field around the surface are clearly generated. Such changes require fluctuating forces, and these forces may be considered as the source of fan noise.

The aerodynamic origin of fan noise allows us to make some pertinent observations, such as:

(a) In the same sense that even the most perfect aerodynamic shape generates a turbulent wake, so all fans must generate noise.

(b) Turbulence intensity increases with flow speed and so the noise of a fan will generally increase as the characteristic flow speeds within it are increased.

(c) Any design or operating fault that results in flow "separation" or "stall", will increase the noise generated by the fan blade.

(d) The close proximity to the blading of fixed surfaces, either upstream or downstream will increase the fan noise - especially at the blade passage frequency.

(e) The vibrational properties of the fan blading play no part, generally, in fan noise. The generation of aerodynamic noise is not dependent upon the local movement of surfaces.

FAN NOISE PREDICTION

The empirical "laws" relating fan noise to size, static pressure, speed and capacity have been known for some time. These state that the overall sound power level of a class of fans, at a fixed point of rating, will vary as

$$\begin{aligned} &70 \text{ Log (size)} + 50 \text{ Log (speed)} \\ &20 \text{ Log (size)} + 25 \text{ Log (pressure)} \qquad (1) \\ &10 \text{ Log (capacity)} + 20 \text{ Log (pressure)} \end{aligned}$$

The complications arise however when we try to compare fans of different types, at different points of rating, and to look at the shape of the frequency spectrum.

The noise of a fan is often considered in two distinct parts: rotational noise, including blade passage frequency tones, and vortex noise which is characteristically broadband in nature. Whilst the frequencies at which rotational noise will occur are predictable, the levels of occurrence are highly variable, depending in many instances upon fairly detailed aspects of the fan design - such as clearance between the blade tip and cut-off in a centrifugal fan. Vortex noise, however, is more predictable and more amenable to measurement and it is with this aspect of fan noise that current prediction techniques are able to cope.

ALLEN METHOD

One of the earliest methods of predicting fan noise, and one which is still substantially useful, is by C.H. Allen and is published in Noise Control by Beranek (2). The Allen procedure expresses the power delivered to the duct on either the exhaust or the supply side of the fan by

$$\begin{aligned} PWL_B &= PWL_S + 10 \text{ Log}(Q) + 20 \text{ Log}(P) \\ &= PWL_S + 35 + 10 \text{ Log}(HP) + 10 \text{ Log}(P) \end{aligned} \tag{2}$$

Where PWL_B is the octave band power level (re 10^{-12}watt)

PWL_S is the specific power level in each octave band (re 10^{-12}watt)

Q is the fan discharge capacity (cfm)

P is the fan static pressure (inches w.g.)

HP is the rated motor horsepower

Values of PWL_S for the two major classifications of ventilation fan are tabulated in Figure 1. The accuracy of the Allen method is quoted at about $\pm$ 4 dB.

The dependence of fan noise upon the flow capacity, and upon the square of the developed pressure - which is implicit in the first line of Eq (2) - is one form of the fan noise law, and about this there is little dispute. Eq (1), however, assumes that the noise is uniquely determined by the volume flow and the pressure drop alone. It allows the geometry of the fan system to be important only insofar as a difference in spectrum shape is allowed between axial flow and centrifugal flow units.

The Allen method also requires that the fan is operated at or close to its maximum efficiency point. In many real situations ventilation systems are not optimised - especially if they are variable in speed. In such cases the specific power levels in Figure 1 should be increased - but by an unspecified amount.

Finally, the Allen method suggests that 5 dB be added to the octave band containing the blade passage frequency, to account for rotational noise. The blade passage frequency is given by

$$f = N \times S/60$$

where N is the number of blades

S is the rotational speed (rpm)

GRAHAM - CHRISTIE METHOD

In 1966 as a result of work carried out by J.B. Graham and D.H. Christie, a fan prediction technique was developed which took into much greater account the design details of the fan. The procedure (3) expresses the power delivered by the fan in the form

$$PWL_B = PWL_S + 70 \operatorname{Log}(D) + 50 \operatorname{Log}(S) - 259 \qquad (3)$$

where PWL_B and PWL_S are as before (re 10^{-12} watt)

D is the diameter of the fan wheel (inches)

S is the fan speed (rpm)

Values of PWL_S for a wide variety of fan types are given in Figure 2. This method is therefore substantially more explicit than the Allen method in allowing differences in fan geometry to be expressed as differences in noise level and spectrum shape. It is important to note that Figure 2 must not be used as an indication of the relative noise generating properties of different fan designs. Different fan types at identical speeds will have vastly different volume and pressure characteristics, and since noise levels are a function of volume and pressure, the noise levels will be quite different also.

The Graham-Christie method assumes, like the Allen method, that the fan is operating at maximum efficiency. Again it is suggested that a 5 dB correction be applied at the appropriate frequency to account for blade generated tones.

GROFF (ET AL) METHOD

Finally and more recently a still more sophisticated method of fan noise prediction has been developed. At present its use is restricted to forward and backward centrifugal fans - but these are the types most frequently used in building ventilation systems.

The Groff method (4) differs from previous methods in at least two respects:

(1) it recognises that as the operating parameters of a fan change, the nature of aerodynamic noise is such that the shape of the frequency spectrum must also change (unless a non-dimensional "strouhal" frequency is used);

(2) it is not limited to peak efficiency operating conditions - it allows operation both below and above optimum design point.

As regards blade frequency tones, however, the Groff technique is no different from its predecessors - a 5 dB correction in the appropriate octave band.

The Groff method may be written

$$\begin{aligned} PWL_B &= 10 \text{ Log } (QP^{\alpha}) + B \\ &= A + B \end{aligned} \qquad (4)$$

where PWL_B is the octave band power level (re 10^{-12} watt)

Q is the fan discharge capacity

P is the static pressure

α is a frequency dependent exponent

A nomograph for the term A is given as Figure 3. Charts for term B are produced as Figures 4 and 5, for backward curved blades and forward curved blades respectively.

IN CONCLUSION

The way in which these prediction techniques compare with each other and with some measured data is shown for a centrifugal fan in Figure 6. It is seen that there is substantial "scatter" between the prediction techniques at low frequencies and that in the mid- and high-frequency range the prediction techniques tend to overestimate the fan noise by up to ten decibels. The order of discrepancy in our experience is generally less than is illustrated here, i.e. the noise prediction equations provide a reasonably good estimate of the fan noise that will occur in practice.

The prediction techniques that we have described usefully point out some of the flow and design parameters which influence a fan's ability to generate noise. It may be of interest to conclude this paper by paraphrasing some of the guidelines presented in the ASHRAE Guide and Data Book to which I referred earlier.

1. The air distribution system should be designed for minimum resistance regardless of fan type, since fan sound generation is directly related to the required static pressure.

2. For a given duty, the frequency spectrum of sound generated by candidate fans should be compared as part of the fan selection procedure.

3. The fan should be selected to operate near its maximum efficiency point when handling the required air quantity and static pressure.

4. Connections at both the fan inlet and outlet should be designed for uniform straight air flow. Accelerated and swirling inlet air flow should be avoided particularly.

REFERENCES

1. ASHRAE Guide and Data Book, 1967.

2. Beranek, L.L. "Noise Reduction", McGraw-Hill, 1960.

3. Graham, J.B. "A Method of Estimating the Sound Power Level of Fans", ASHRAE Journal, V8, No. 12, Dec.1966.

4. Groff, G.C., Schreiner, J.R., Bullock, C.E. "Centrifugal Fan Sound Power Level Prediction" ASHRAE Transactions, 1967.

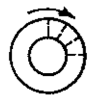
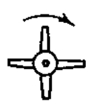
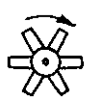
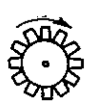
Accuracy of measurement — esp at l.f.

End Reflection losses

Inlet conditions — esp with axial

	Octave band centre frequency - Hz							
Fan Type	63	125	250	500	1000	2000	4000	8000
Centrifugal	47	50	45	40	35	30	25	20
Axial	45	49	49	48	47	44	41	37

FIG 1 BASE POWER LEVELS FOR USE WITH ALLEN PREDICTION METHOD
(dB re 10^{-12} Watt)

Fan type	Description	Applications	Octave band centre frequencies - Hz							
			63	125	250	500	1000	2000	4000	8000
	Centrifugal fan with backwardly curved airfoil blades.	1. General ventilation and air conditioning work. 2. Industrial applications where corrosion erosion or dirt is not a problem.	97	94	92	90	88	85	82	80
	Centrifugal fan with backwardly curved or sloped single thickness blades.	1. General ventilating and air conditioning work. 2. Industrial applications where corrosion, erosion or dirt is not a major problem.	97	95	93	94	93	88	84	79
	Centrifugal fan with single thickness blades with foward curved heel and radial or nearly radial tip.	1. Used principally for industrial applications where medium to high pressure requirements must be met. May be used moderately dirty applications.	108	104	94	92	90	88	89	87
	Centrifugal fan with single thickness radial blades. Blades are relatively short in direction of air flow.	1. Industrial applications where corrosion, or erosion is a problem, or dust loading is very heavy. Also used in conveying systems where material passes through the fan wheel.	103	103	96	96	93	88	85	84
	Centrifugal fan with single thickness radial blades. Blades are relatively long in direction of air flow.	1. Industrial applications where relatively small volumes at high pressure are required.	114	111	104	104	100	97	94	91
	Centrifugal fan with single thickness blades curved foward at both heel and tip.	1. General ventilation and air conditioning work for low pressure, high capacity requirements.	105	101	98	90	89	87	82	77
	Axial fan with relatively long blades and small hub.	1. Designed to meet requirements of high capacity at very low pressures.	92	93	92	91	91	88	84	80
	Axial fan where hub is about 50% of fan tip diameter.	1. General ventilation and air conditioning work. 2. Industrial applications where corrosion, erosion or dirt is not a problem.	96	93	97	96	94	90	86	85
	Axial flow fan with relatively short blades and large hub.	1. Industrial applications where requirement is for high pressure at medium capacity.	90	86	90	93	93	89	83	81

Special note - This table cannot be used for direct comparison of the noise generated by any one fan with the noise generated by any other fan in this list.

FIG. 2 BASE POWER LEVELS FOR GRAHAM PREDICTION METHOD - dB re 10^{-12} watt.

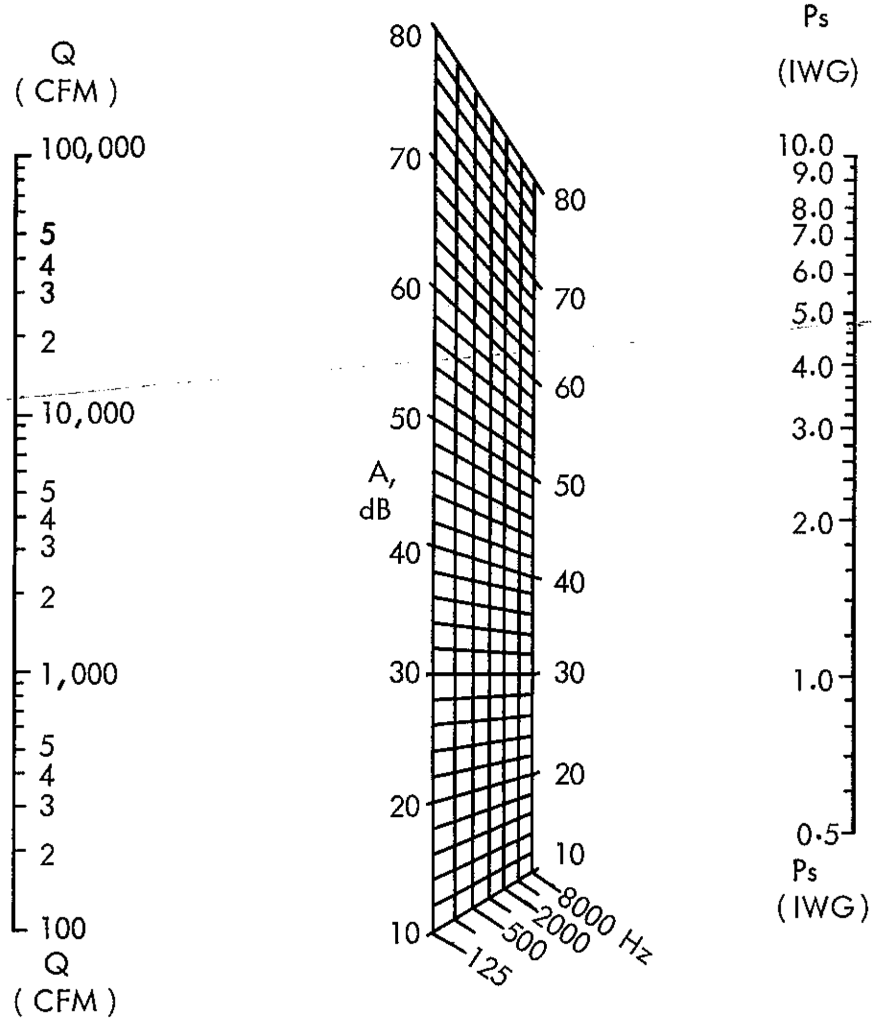


FIG 3 NOMOGRAPH FOR DETERMINATION OF 'A' IN PREDICTION METHOD BY GROFF ET AL

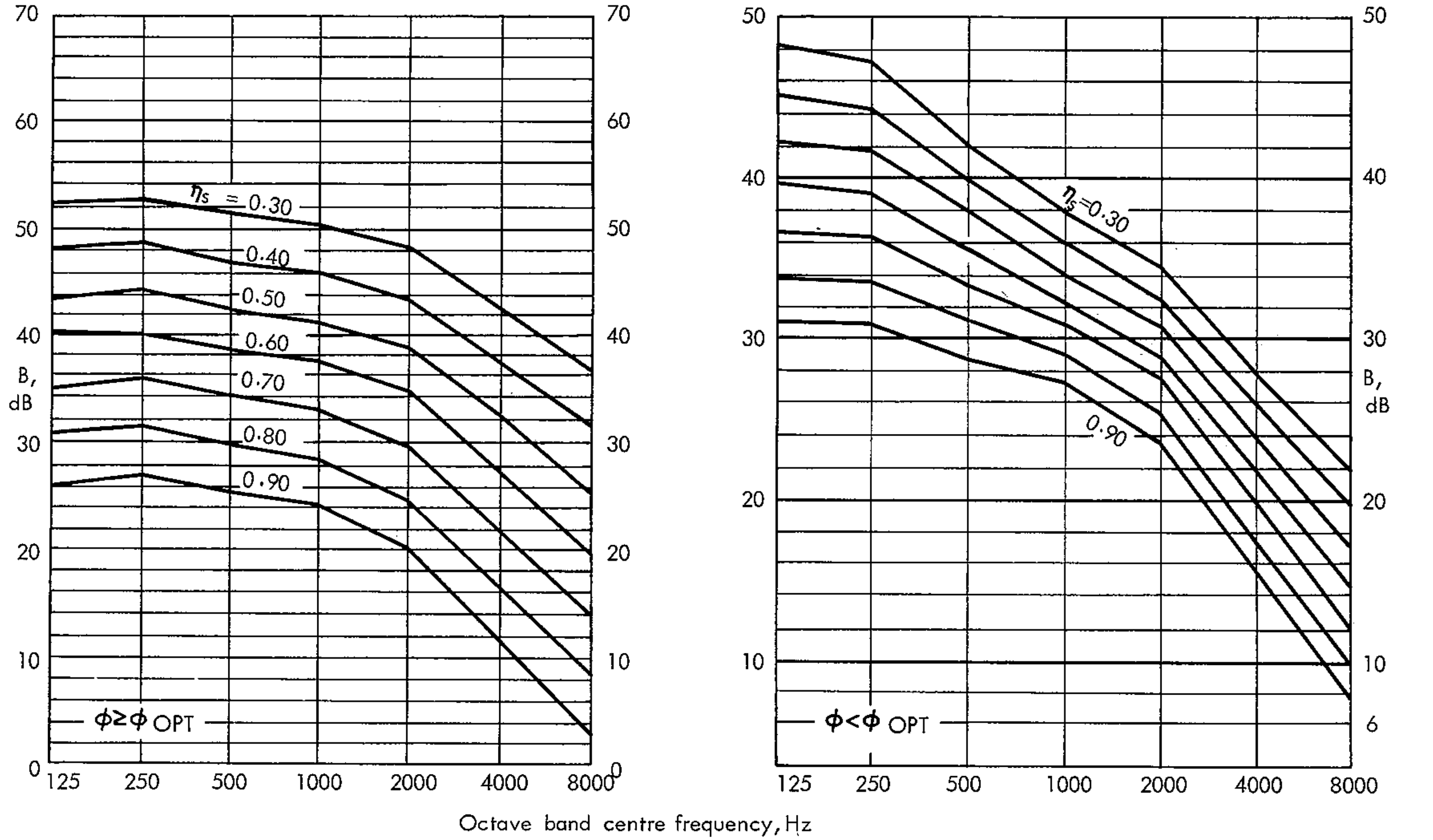


FIG 4 CHART FOR DETERMINATION OF B IN PREDICTION METHOD BY GROFF ET AL CENTRIFUGAL FANS WITH BACKWARD CURVED BLADES.

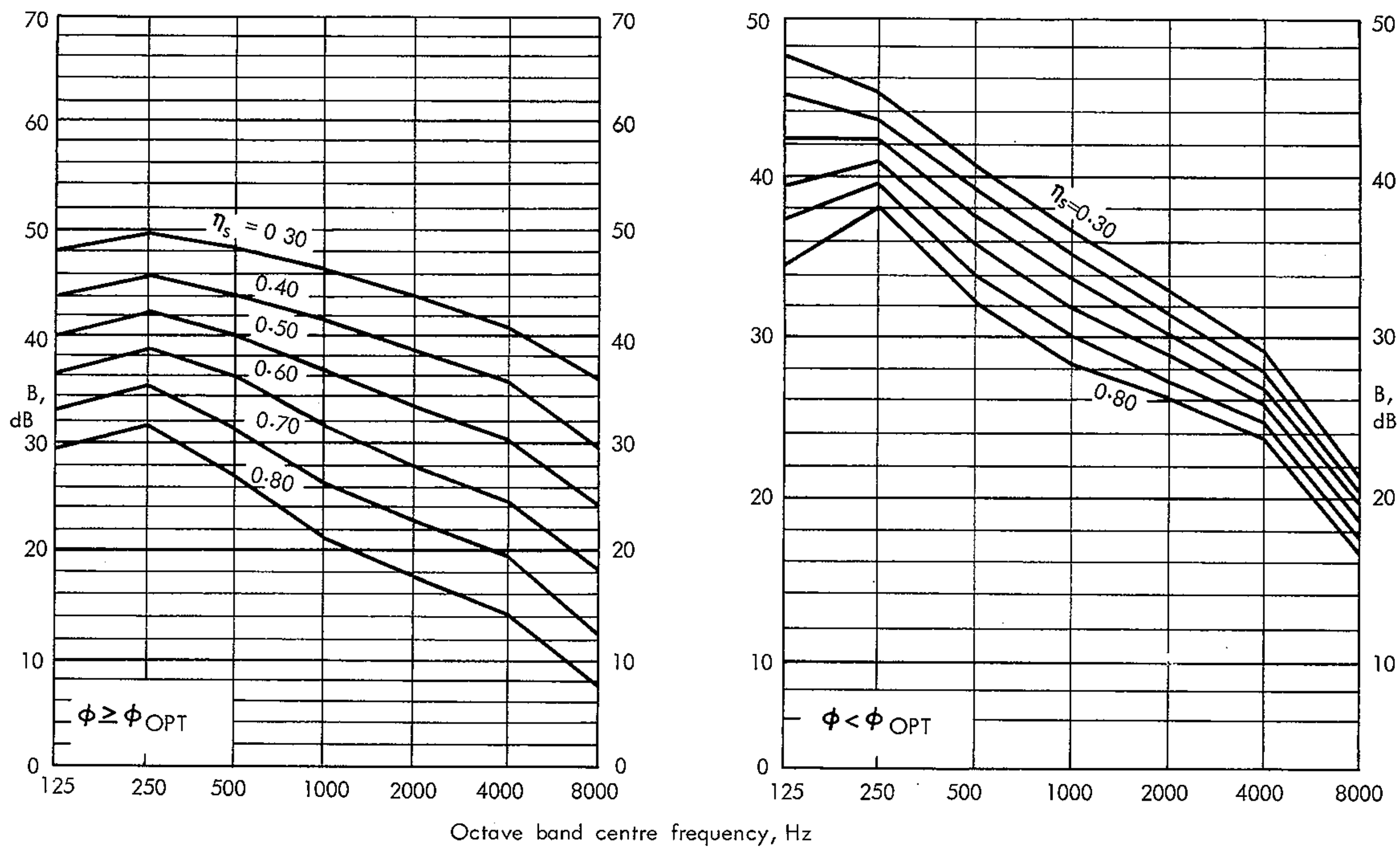


FIG 5 CHART FOR DETERMINATION OF B IN PREDICTION METHOD BY GROFF ET AL CENTRIFUGAL FANS WITH FORWARD CURVED BLADES.

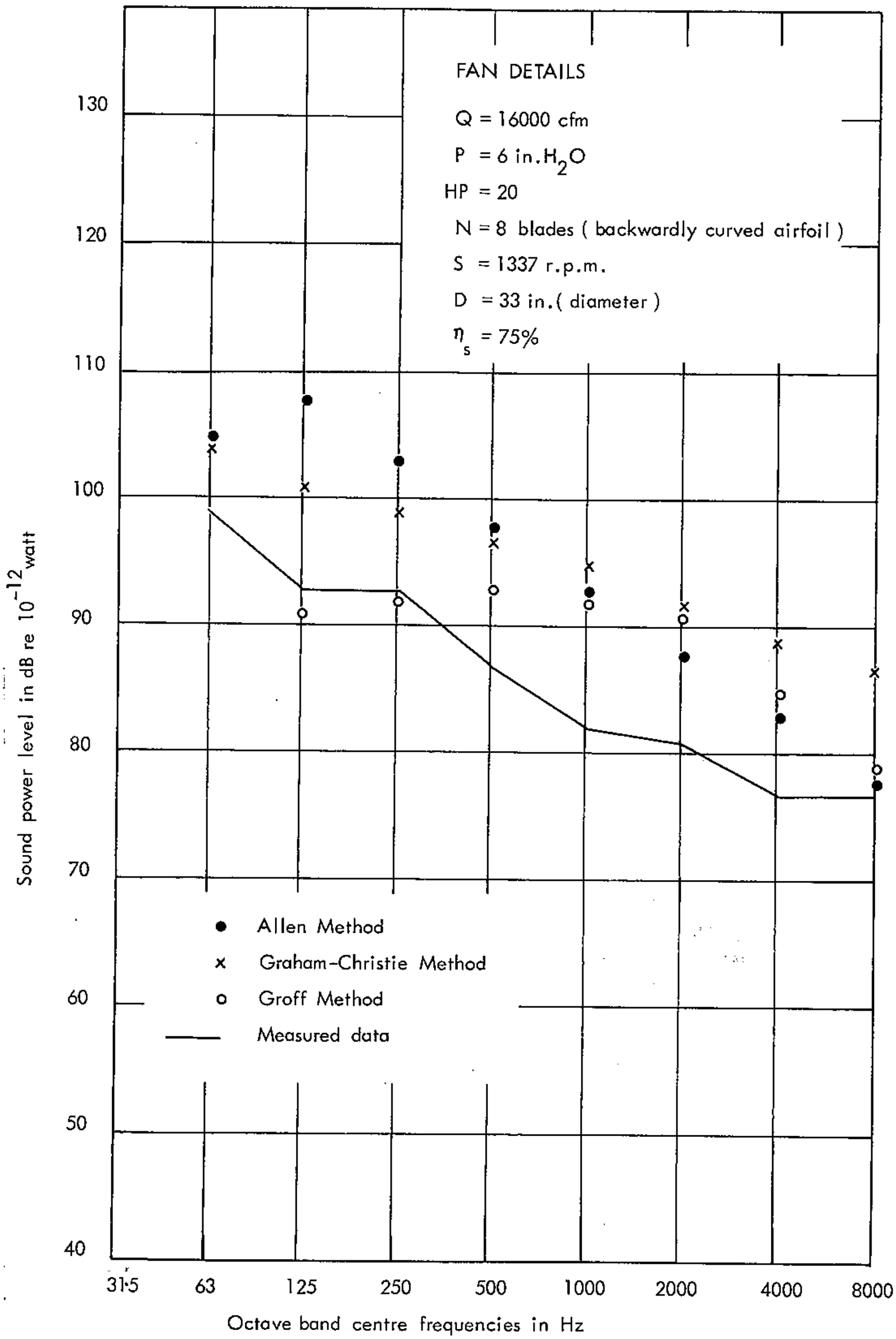


FIG. 6 COMPARISON OF PREDICTION METHODS WITH MEASURED DATA FOR CENTRIFUGAL FAN.

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