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Implementing Agreement for a Programme of Research and Development on Wind Energy Conversion Systems

18th Meeting of Experts – Noise Generating Mechanisms for Wind Turbines

Petten, the Netherlands, Nov. 27–28, 1989

Organized by:
Project Management for Biology, Energy, Ecology (BEO)
of the
Research Centre Jülich (KFA) on behalf of the
Federal Minister of Research and Technology,
the Fluid Mechanics Department
of the Technical University of Denmark

Scientific Coordination:
M. Pedersen (Techn. Univ. of Denmark)
R. Windheim (BEO-KFA Jülich)

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CONTENTS

INTRODUCTORY NOTE

NOISE GENERATING MECHANISMS OF WIND TURBINES

W. Stam
N. van der Borg

1. Introduction

The acoustic noise emitted by wind turbines can be a serious obstacle for the realisation of wind energy projects due to the annoyance that might be experienced by persons who live in the neighbourhood of these projects. This limits the total area that can be used in the site selection procedure or it might lead to limited operation time to for instance only the daily hours. To prevent these unwanted situations knowledge is required on the noise generating mechanisms of wind turbines with which new turbines can be designed, and existing turbines can be modified in such a way, that the noise emission is reduced to acceptable values. The required information must be obtained in three steps. First the physical description of the noise generating mechanisms must be found, secondly this description must be translated to mathematical models and finally these models must be validated and if necessary adjusted by information obtained from measurements.
In this note the current knowledge on the noise generating mechanisms is mentioned briefly in chapter 2. In chapter 3 the expected development of the acoustic noise from wind turbines is outlined. The available measurement techniques that can be used for the verification of the models are mentioned in chapter 4.

2. Models

The acoustic noise of wind turbines can be split up into two parts:
- aerodynamic noise, originating from the rotor blades,
- mechanical noise, originating from the nacelle (gearbox, generator, bearings and other parts).

The aerodynamic noise is generated by three phenomena:
- the variations of inflow
- trailing edge effects
- tip vortex effects

The mechanisms of aerodynamic noise have been described in mathematical models which have not yet been validated properly by experiments. The aerodynamic noise mechanisms all have a certain directi-

vity. Apart from the directivity the path of the noise to the observer is rather simple. The path of the mechanical noise to the observer, however, is in most cases complicated: the noise can leave the nacelle by ventilation holes or other gaps in the nacelle, by noise conduction through the nacelle shell or, which is the most complicated part, as contact noise via the nacelle shell and via the tower.
These sources of noise have only been described by simple semi empirical models which have not been validated. It is felt that these models are too simple to describe the mechanical noise completely.

3. Expected developments

In ref. [1] a semi empirical relation is given for the aerodynamic noise of wind turbines in dependence of the diameter and the tip speed. The relation is:

$$L_{w,a} = 10 * \log D + 50 * \log V_t - 4$$

with $L_{w,a}$ = aerodynamic acoustic source power (in dB(A))
D = rotordiamater (in m)
V_t = tip speed (in m/s)

In ref. [2] the aerodynamic source power has been calculated with this relation for 35 wind turbines ranging from 2.4 m to 100 m rotor diameter. The calculated values of the aerodynamic noise have been compared with the measured values of the total noise (see figure 1). From this it can be seen that the high aerodynamic source power values (corresponding with large rotor diameters) are about equal to the total source power values. This indicates that for large wind turbines (>30 m) the dominant noise is the aerodynamic noise. For the small wind turbines the contribution of the mechanical noise is significant. Since the future turbines will be larger than most turbines of today it can expected that the aerodynamic noise will be the dominant noise in the future.
Furthermore it can be seen from the relation above that higher tip speed values result in higher noise levels. Since the modern turbines are expected to have higher tip speed values than the turbines of today, an extra

raise in the contribution of the aerodynamic noise is foreseen. This means that for the existing turbines with diameters up to about 30 m it is required to focus on both the mechanical noise and the aerodynamical noise while for the development of new and large wind turbines the modelling of aerodynamical noise is essential.

4. Measurement techniques

The total acoustic noise emitted by wind turbine is characterised by the acoustic source power. In ref. [3] recommended practices are given for the measurement of the sound pressure level at a reference distance from a turbine. From this value the source power can be derived. Apart from some discrepancies with other recommended practices (ref. [4]), ref. [3] gives a rather straightforward procedure for the measurement. However it gives no information on the contribution of the various mechanisms to the total source power. For this purpose special measurement techniques have to be applied.
The techniques can be split up into three parts:

1. directional measurements at a distance,
2. non-directional measurements at a distance,
3. local measurements.

ad 1. Directional measurements at a distance

The aim of these measurements is to separate the contribution of sound sources with different locations such as the nacelle and the blade tips. This can be done using a synthetic acoustic antenne. In this technique a row of microphones is used. Correlation of the microphone signals gives the possibility to analyse the sound field as a function of the angle of incidence. Another technique is the use of a parabola microphone that can be directed to various parts of the turbine successively.

ad 2. Non-directional measurements at a distance

The aim of this measurement technique is to analyse the measured sound pressure of a turbine in such a way that the contribution of various sound sources can be deduced. Methods that can be used in this technique is the recognition of mechanical frequencies (such as gearbox tooth frequencies) in measured narrow band frequency spectra (see figure 2). Further information can be obtained by the possible occurrence of phenomena such as modulation (sound level fluctuations with the blade frequency) or Doppler effect (periodical shift in frequncy values of possible sharp peaks in the spectra). The experimental facilities needed for this approach and the actual measurements are not very complicated. However, the present experience with this analysis procedure is limited.

ad 3. Local measurements

Acoustic noise is normally characterised by the sound pressure. The sound pressure near a certain surface gives little or no information on the sound power that is emitted by that surface. The reason of this is that the sound pressure near the surface might be caused by incidenting and reflecting background noise or by circulating source power. For the separate measurement of all noise emitting parts of the turbine the sound <u>intensity</u> through the surface of these parts should be measured instead of the sound <u>pressure</u>. The sound intensity can be seen as the net sound power per unit area. The measurement of sound intensity requires special equipment that can be bought commercially. No experience is available yet with intensity measurements on wind turbines.

5. Conclusion

A serious bottle neck for the implementation of wind energy is the acoustic noise production of the wind turbines. To diminish this problem more knowledge on the noise generating mechanisms is needed. At the moment prediction models for aerodynamical noise are available. These models, however, have not yet been validated sufficiently. The mechanisms that generate mechanical noise are not hard to understand, but the path from the source inside the nacelle to the observer outside the nacelle is complicated. This is the reason that no complete models for mechanical noise exists. The measurement techniques that can be used for the validation of prediction models all require special instruments to be used by experts on acoustics.

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THEORETICAL MODELLING OF NOISE GENERATED BY WIND TURBINES

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J. Scott

INTRODUCTION

Theoretical models of the noise generation mechanisms of wind turbines should provide the designer with a means of assessing their likely environmental impact. The development of such models requires a thorough understanding of the physical processes responsible for aerodynamic sound generation.

This paper is principally a review of the most important generation mechanisms which have been proposed. The current level of development of predictive models is indicated, and recommendations are made for work which needs to be done to produce a quantitative model having a sound physical and mathematical basis.

The theory of sound generation by fluid motion was first put on a quantitative basis by Lighthill (1). In this theory the equations of fluid mechanics were reformulated such that they appeared as the usual equation describing sound (the wave equation) with terms on the right hand side which were interpreted as acoustic sources. Solutions of the wave equation being available, in principle, it remained to determine the source terms. This was and remains a difficult theoretical task because it would become necessary to solve the full equations of fluid mechanics.

This is not to say that the acoustic diffraction problems which result when the sources are known are easy, indeed the extensive subject of aeroacoustics is mostly devoted to their solution. Nevertheless, the main uncertainty in correctly formulated predictive models is the description of the flow.

In applying Lighthill's theory to any of the large number of subsequent "acoustic analogies" one may reason in a qualitative manner using order of magnitude estimates of the source terms. This is reasonable in many cases because the qualitative properties of the flow may be well understood. Alternatively one can assume semi-empirical forms for the sources based on experimental data in order to obtain quantitative results. However in situations where a complex turbulent flow has not been adequately characterised by experimental measurements, quantitative predictions of sound radiation will usually not be possible.

Certain general conclusions arise from the Lighthill and similar theories. The sources are acoustic quadrupoles and are relatively weak. They can however be made more efficient by the presence of nearby solid bodies, of which wind turbine blades provide an example.

A plane surface near a region of fluid in which sound is generated does little more than reflect the sound as was shown by Powell (2). The same is also true of surfaces which are slightly curved. If, on the other hand, the curvature is on a scale which is small compared to the acoustic wavelength at the frequency of interest then the radiation efficiency of sources close to the surface can be considerably increased. The extreme example of this is a sharp edge such as occurs at the trailing edge of a blade.

Another general feature of aerodynamic noise is that all theoretical models predict that it increases rapidly with flow speed. In the case of turbine blades this speed is essentially the blade speed and is proportional to distance from the rotor axis. For this reason it is to be expected that the main noise sources will be located towards the ends of the blades. This should not be taken to imply that one necessarily needs to consider the tip region (i.e. within a chord of the edge) with its complex three-dimensional flow. Just as with the calculations of lift on the blades, a two-dimensional model of the blades is simpler, more tractable and leads to results which are reasonably accurate provided that the span to chord ratio is sufficiently large.

Once sound is generated by the passage of an eddy it propagates through the air to the receiver. Because the blade is moving relative to the observer there is a Doppler shift in the frequency of the sound due to the non-zero Mach number of the blade. Other effects of non-zero Mach number are associated (Crighton (3)) with the fact that the blade moves between the emission of the sound and its reception. This has the result that the directivity of the source appears different to the observer than it does to the blade. Typically, this makes the noise appear weaker when the blade is moving away from the observer.

Other effects include ground reflection and the refraction and attenuation in sound over long distances. We shall not discuss such propagation effects further, but a complete predictive model should include them.

The two main source mechanisms for aerodynamic noise from an upwind, horizontal axis rotor are inflow turbulence noise and trailing edge noise. The former is generated primarily by the passage of turbulence in the approach flow over the leading edge of the aerofoil. The trailing edge noise is generated by turbulence in the blade boundary layer passing over the trailing edge of the aerofoil.

A NOTE ON THE KUTTA CONDITION

In classical calculations of the steady flow around a lifting aerofoil it is usually assumed that the velocity at the sharp trailing edge is finite. The assumption, known as the Kutta condition, has no real theoretical basis when the boundary layer is turbulent. It is nonetheless found to give good agreement with observations. The qualitative rationale behind it is that if it were violated, vorticity would be shed from the edge until it is satisfied.

A similar condition could be applied to unsteady problems arising from the passage of turbulent eddies over the blade. There would then be vorticity production from the trailing edge at a rate necessary to maintain finite velocity there; this condition must evidently hold if the frequency of eddy passage is sufficiently low. On the other hand it is observed (Brooks & Hodgson (4)) that there is much better agreement with measurements of boundary layer trailing edge noise if no vortical production is assumed. The question arises as to how low the frequency (or equivalently how large the eddy) needs to be before the Kutta condition should be applied.

When the Kutta condition is applied to unsteady problems the vorticity which is shed from the the trailing edge is modelled as forming an infinitely thin sheet. For this to be a reasonable model the boundary layer must be thin compared with the eddy that is causing the disturbance. It is plausible therefore to suppose that the Kutta condition should be applied to eddies for which the eddy is large compared with the boundary layer thickness h. This is true for the inflow turbulence noise but not for the trailing edge noise (as evidenced by (4)). The conclusion is that we should apply the Kutta condition to the inflow turbulence, but not to the trailing edge source. This procedure is provisional, but seems to be the best one can adopt until the basis for the Kutta condition is more precisely understood.

INFLOW TURBULENCE NOISE

One of the important noise sources for a wind turbine results from the rapidly moving blades cutting through gusts of wind. These gusts result from turbulence in the boundary layer above the ground. A blade moving at speed U, encountering a disturbance of wavenumber k, will see a temporal fluctuation at angular frequency w=Uk. In response, sound of frequency w will be radiated, principally from the blade edges as discussed above.

If we consider a blade travelling at, say, $U = 50ms^{-1}$ and a frequency range from f = 50 Hz to f = 10kHz, the eddy sizes (L = U/f) that result lie between L =1m at the lower end and L = 5 mm at the upper end of the frequency range. Although the atmospheric boundary layer contains eddies of all sizes, the dominant ones at height H are found (Hinze (5)) to have scales of order H. Since wind turbines operate at several tens of metres from the ground, the eddies which radiate in the above frequency range are not among the dominant ones: they are relatively weak by-products of the main turbulent eddies.

The above considerations indicate that a typical turbine has blades whose chord is comparable to the eddy size at the lower end of the frequency range of interest. At higher frequencies, where the eddies are small compared with the chord, it is appropriate to regard the sound source as due to the eddies interacting with the edges of the blade.

Another length scale of fundamental significance is the acoustic wavelength, λ. Since $\lambda = c/f$ (where c is the speed of sound), the ratio of the radiating eddy size to the acoustic wavelength is M=U/c, the Mach number of the blade, which is typically of the order of 0.2. Thus, the eddies are small compared to the acoustic wavelength by a factor of the blade Mach number.

Taking the frequency range as before and $c = 300ms^{-1}$ we obtain wavelengths ranging from 6m to 3cm. For blades of the order of 1m in chord it may be acceptable to regard the blade as acoustically compact (i.e. small compared with the wavelength) at the lower end of the frequency range; however it certainly is not appropriate to do this over most of the range. The reason for stressing this point is that many workers (e.g. George and Kim (6), Grosveld (7)) have assumed that the blades could be represented as acoustic <u>line</u> dipoles. This is incorrect unless the blades are compact. The reason why the assumption has often been made, aside from the simpler analysis which results, appears to be that it was adopted uncritically from the theory of rotor tones (at multiples of the blade passing frequency) for helicopters. For wind turbines, the Mach number and blade-passing frequency are lower (the latter is typically a few Hertz) and this makes the rotor tones less significant, particularly in the audible range.

The inflow turbulence source can be modelled by regarding the boundary layer as infinitely thin. The Kutta condition should be applied at the trailing edge, as noted above. A further approximation is permissible because the velocity, U, is always much greater than the wind gust velocities. This approximation, often referred to as rapid distortion theory in turbulence analysis, is that the vorticity in the gusts is passively convected by the flow that would be there in the absence of the gust. This flow can in turn be calculated, given the thinness of the blade boundary layer, by steady irrotational aerofoil theory. Using this approximation it becomes mathematically feasible to calculate the detailed source distribution for noise generation without recourse to full numerical fluid dynamics.

The inflow turbulence source was studied by Amiet (8) with a flat strip as a model of the aerofoil. He assumed that the flow was parrallel to the strip (i.e. no lift). Detailed results were given for the case when the observer is at right angles to the strip.

Glegg, Baxter and Glendinning (9) expressed the sound field in terms of the pressure jump across the strip using the Ffowcs Williams and Hawking equation. They then used the theory of Amiet (8) to determine the pressure jump. Unfortunately they appear to have taken Amiet's result for an observer at right angles to the blade and incorrectly assumed it applied in all directions. This is certainly invalid unless the blade chord is small compared with the acoustic wavelength. One result is that the source directivity given by Glegg et al is as if the blade were modelled as a line dipole. As described above, this is an assumption which can only apply at the lowest frequencies in the range of interest.

Another aspect of the inflow turbulence source considered by Glegg et al was the non-zero blade thickness. The model they use was taken from an unpublished conference presentation by Hawkings of which we have only seen the abstract. For this reason it is difficult to evaluate the model, but it appears to be assumed that the slope of the blade surface is small. This will not apply near the nose of the blade, which is probably the principal source of inflow turbulence sound.

Returning to the discussion of the quantitative modelling of inflow turbulence, we note that the eddy size is small compared with the blade chord except at the bottom of the frequency range. When this holds, the sound will be generated near the blade edges. The application of the Kutta condition at the trailing edge makes scattering inefficient there because it smoothes out the singular nature of the field which causes edge enhancement. We are therefore left with the leading edge interacting with the inflow turbulence. This is probably the main source of inflow noise.

The fact that the nose is not sharp will presumably manifest itself at frequencies such that the eddy size is comparable to the nose radius of curvature. In this case, a model including the convection and distortion of the turbulence by the flow as well as the acoustic scattering by a rounded nose will be needed. A parabolic nose profile is the simplest reasonable model for this.

In summary, the inflow turbulence source can be modelled by rapid distortion theory to describe the way the turbulence convects around the blade. An acoustic diffraction problem then needs to be solved to obtain the sound generation. Additional simplifications, such as a parabolic nose profile may be necessary to make the analysis tractable. Work along these lines is now in progress.

The atmospheric turbulence spectrum is needed as an input to the model. Since the eddies of interest are small compared to the dominant eddy size, which scales with height, it is reasonable to assume that they form part of the inertial subrange (5). It is believed that in this range the Kolmogorov spectrum applies. This was confirmed by the measurements of Boston and Burling (10) and means that the only unknown is the local energy dissipation rate of atmospheric turbulence.

Glegg et al took the Von Karman spectrum, which reduces to the Kolmogorov spectrum in the interesting range of small eddy scales (i.e. high wavenumbers). They determine the unknown constant in the von Karman spectrum by reference to the rms fluctuating velocities which are dominated by the low wavenumber parts of the spectrum. More accurate results are probably available through direct application of the Kolmogorov theory.

TRAILING EDGE NOISE

Trailing edge noise is the result of blade boundary layer turbulence interacting with the trailing edge of the blade. The eddies generate sound at a frequency dictated by their size and velocity of convection past the edge. Because the fluid in the boundary layer travels more slowly than that in the free stream, the appropriate velocity is less than U (a value of 0.6U is often quoted (Bull (11)). The typical scale for eddies in the blade boundary layer is h, the boundary layer thickness. It follows that we would expect a frequency scale for this source of order U/h.

The standard model of trailing edge noise is as follows. The edge is taken to be a semi-infinite rigid plane. It is assumed that the turbulent velocity field is the same as if the rigid plane were infinite (that is, continued past the edge). Chase (12) solved this problem without the trailing edge Kutta condition; Amiet (13) (as corrected in (14)) also solved the problem, but with a Kutta condition. As was mentioned above, measurements show agreement with the Chase version of the solution (the difference with and without the Kutta condition is of the order of 10dB). Unfortunately, most wind turbine noise models appear to use Amiet's results. This could be a significant source of error.

A further point concerns the behaviour upstream and directly in line with the plane. The theories referred to above predict that the field on the two sides of the plane are in anti-phase. It follows that if the aerofoil were finite, but remained planar, the field on the upstream continuation of the strip would be zero. This is in contrast with the result obtained by simply taking the field from the semi-infinite problem in the direction of the receiver. This is due to interference of the sound waves from the upper and lower sides of the strip. It affects the noise levels observed in a relatively narrow range of angles about the upstream continuation of the strip (for instance those in the rotor plane). The approximation of the leading edge as sharp will, of course, break down at frequencies such that the nose dimensions are comparable or larger than the acoustic wavelength. In any event further analysis is needed if it is required to model angles accurately near the rotor plane.

The trailing edge noise source model requires input from a turbulence model. What is required is the spectrum of surface pressure fluctuations under a turbulent boundary layer. Many models of this are available, but they require the boundary layer thickness at the trailing edge. This point has been considered by Chou and George (15) who provide semi-empirical formulae for boundary layer thickness for a particular aerofoil section. Boundary layer thickness both above and below the blade will be needed.

When the trailing edge is not sharp, the flow can separate generating a wake which is left behind in addition to that which would occur simply from the blade boundary layer. The standard models of trailing edge noise (see Howe (16)) assume that the boundary layer turbulence is simply convected unchanged past the edge which allows one to use measurements of boundary layer surface pressure spectra to model the acoustic sources. The presence of a blunt edge will obviously modify the source statistics near the tip in a way which theory cannot, at the moment, predict. Calculation of the sound field requires the source structure and this must be determined from experiment. As far as we are aware, there is no detailed data on the turbulent statistics of trailing edge wakes with blunt edges.

Brooks and Hodgson (4) present the results of measurements of trailing-edge noise produced by aerofoils, both with sharp and blunt edges. They compare results with predictions based on a sharp trailing-edge predictive model and obtain reasonable agreement except that the measurements with a blunt edge show a hump in the frequency spectrum due to the bluntness. This is ascribed to separation of the flow and associated quasi-periodic vortex shedding. No predictive model for the spectral hump was proposed by these authors.

Models for blunt trailing edge noise based on semi-empirical fitting to limited acoustic measurements have been described by Grosveld (7) and Chou and George (17). Grosveld claims that the hump is centred at a frequency $f=0.25U/(t+4h^*)$ for $t>1.3h^*$ and $f=0.1U/t$ for $t<1.3h^*$ (where t is edge thickness and h^* is momentum thickness). Taking, say $U=50ms^{-1}$, $t=0.01m$, and $h^*=0.25mm$, gives $f=600Hz$. If it were desired to eliminate the effects of trailing edge bluntness, the frequency of the peak could be increased to lie above the audible range. Setting $f=10kHz$ yields $t=0.5mm$.

In situations where trailing edge bluntness is important, more work will be needed, both theoretical, to determine the dominant source type, and experimental, to determine their strength as a function of frequency and trailing edge geometry.

It is instructive to compare the trailing edge and inflow turbulence sources. The former involves relatively intense eddies with small length scale h, the latter has weaker atmospheric eddies at all length scales of interest in the given frequency range. The result is that at the lower frequencies (well below U/h) sound due to inflow turbulence would be expected to dominate, while by the time we get up to frequencies of order U/h , both mechanisms contribute and the more intense eddies in the boundary layer cause the trailing edge mechanism to be dominant. A consequence of this is that we need only consider the inflow turbulence mechanism when the eddy size is larger than the boundary layer thickness. The boundary layer can then be regarded as thin for the purpose of the inflow turbulence calculations.

DISCUSSION AND CONCLUSIONS

The two mechanisms described above form the basis for understanding the important noise sources. Clearly more complex flow phenomena can be involved - for instance close to a blade tip, or during stall - and the difficulty of obtaining quantitative results for these situations is thereby increased, but the same basic principles will apply. Again, for wind turbines sited within a wind farm, the atmospheric inflow turbulence will be enhanced by turbine wake effects, with a consequential increase in noise emission. However, since it is already known that close turbine spacings cannot be used on other grounds, the enhancement in noise generation is in practice likely to be very small.

In the practical circumstances of greatest relevance - which means at low windspeeds, when masking noise is at a minimum - the modelling of inflow turbulence and trailing edge noise as proposed in this paper is expected to give a satisfactory understanding of rotor generated noise.

Work is now in progress (funded by the CEGB) to produce a model of aerodynamic noise from wind turbines based on the principles outlined in this paper.

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NOISE SOURCES ON TYPICAL DANISH WIND TURBINES

B. Andersen

1. Introduction.

During the last approximately 5 years noise from wind turbines has been a still more essential problem. Since noise is the only environmental problem to the utilization of wind power, it is normally the limiting factor when wind turbines are to be erected. During these years Danish Boiler Owners' Association and the Danish Acoustical Institute have carried out several research projects for Danish environmental and technological authorities and for manufacturers and electric power utilities etc. Since 1983 we have performed several measurements of noise emission from wind turbines using the same measurement procedure - a procedure much similar to the revised IEA recommendation from 1988 (ref. /1/). Consequently we have quite a lot of comparable measurement results.

Figure 1 displays an updated summary of the A-weighted, immission-relevant sound power level (L_{WA}) emitted from a lot of wind turbines - plotted as a function of their rated electric power.

Figure 1 shows that new wind turbines are approximately 5 dB more noiseless than older ones, but that the variance is great. The A-weighted sound power level L_{WA} is increased approximately 3 dB per doubling of rated power.

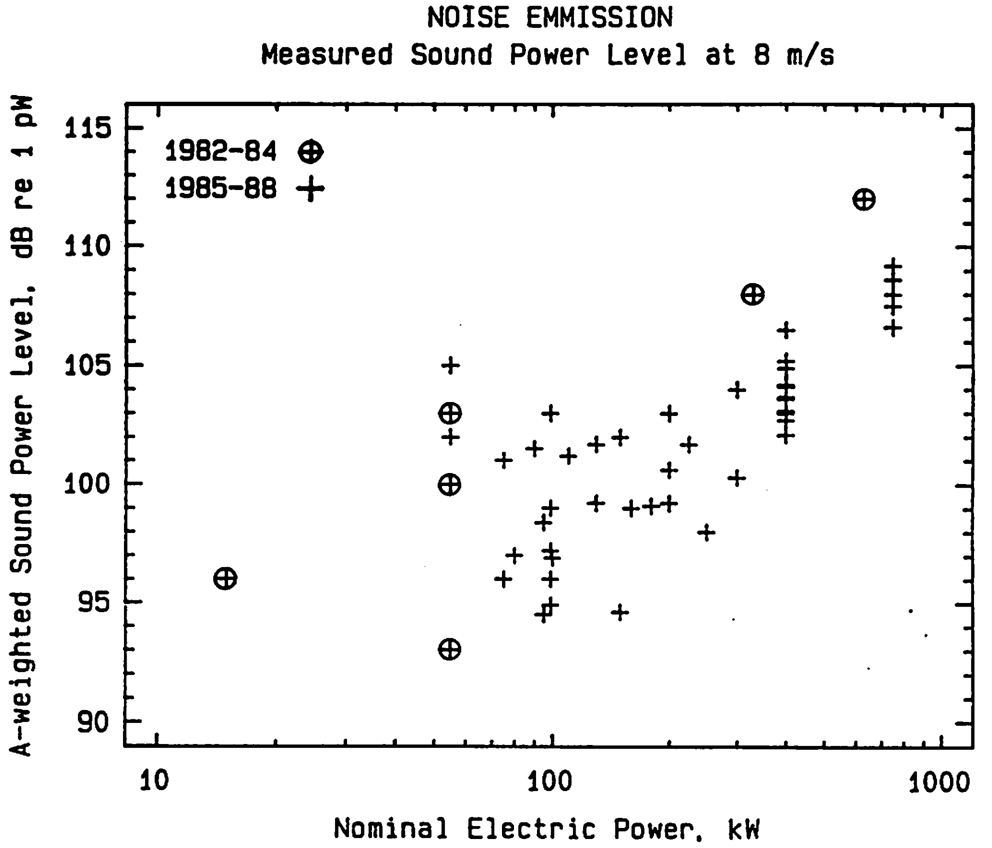


Figure 1. A-weighted immission-relevant sound power level as a function of rated electric power (windspeed at 10 m height v_{10} = 8 m/s).

2. Noise Sources.

For some typical small mass-produced wind turbines (50-75 kW) the influence of some noise sources have been investigated (ref. /2/). To illustrate this Figure 2 shows A-weighted 1/3-octaveband spectra of the total immission-relevant sound power level as well as the contributions emitted from the nacelle and the tower. The noise emitted from the nacelle was determined as the sum of contributions from ventilation openings, roof, walls and floor of the nacelle - see Figure 3.

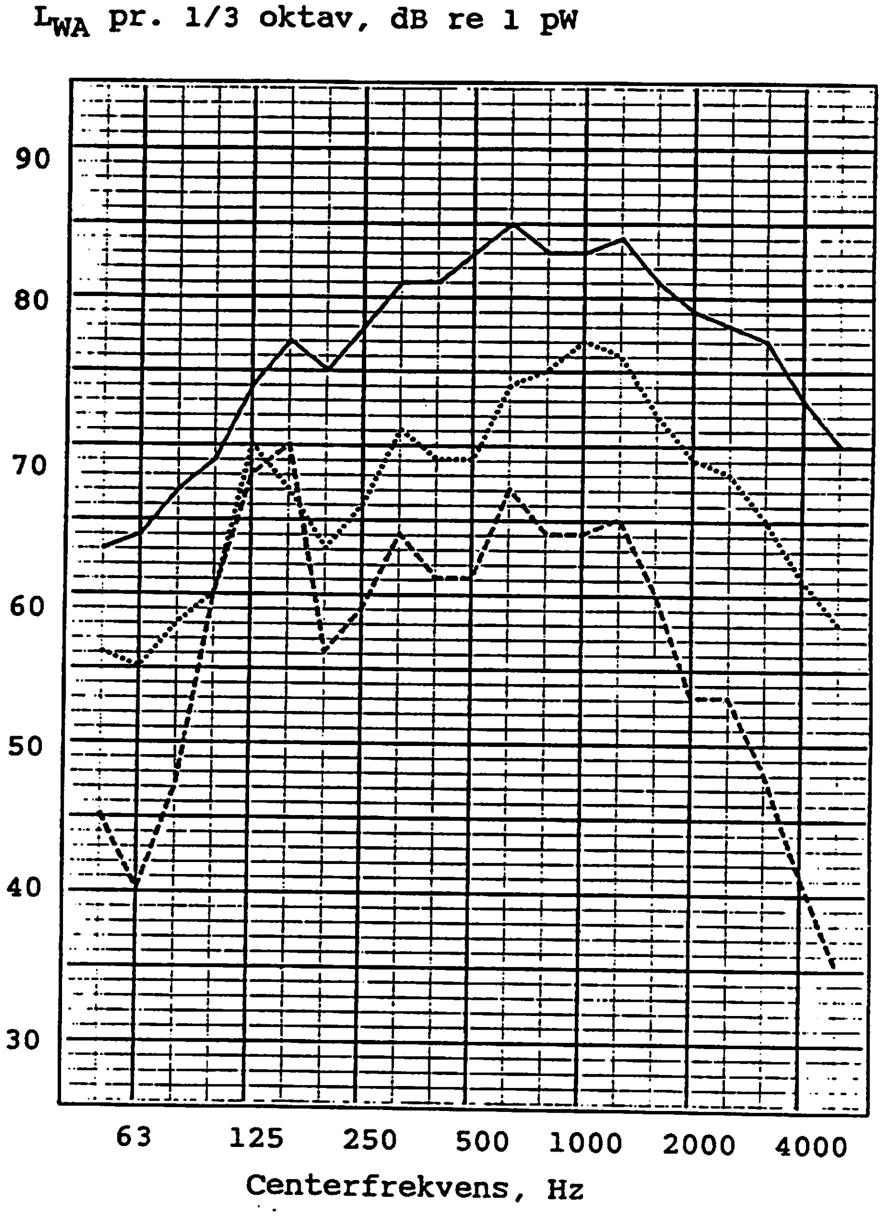


Figure 2. A-weighted sound power level per 1/3 octave

—— total immission-relevant sound power level

···· nacelle

---- tower

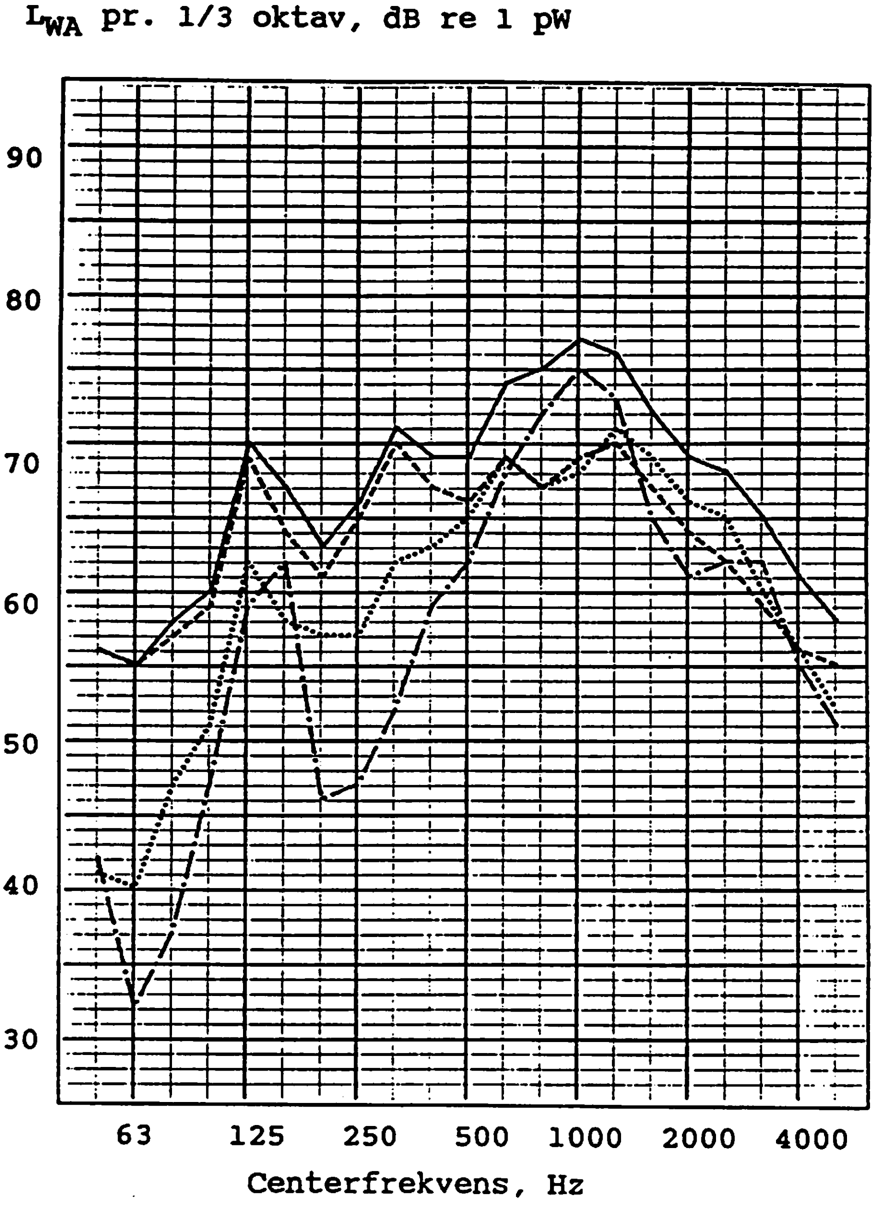


Figure 3. A-weighted sound powerlevel per 1/3 octave for noise sources on the nacelle.

.... ventilation openings

---- nacelle walls and roof

-.-. nacelle floor

—— total

3. Rotor noise.

It is seen from Figure 2 that the noise contributed from the machine components do not explain the total noise emission - the residual noise must be due to the rotor. This way of determining the aerodynamic noise contribution is of course rather uncertain, but often it seems to be the only possible method. Figure 4 displays the results obtained in the above-mentioned case.

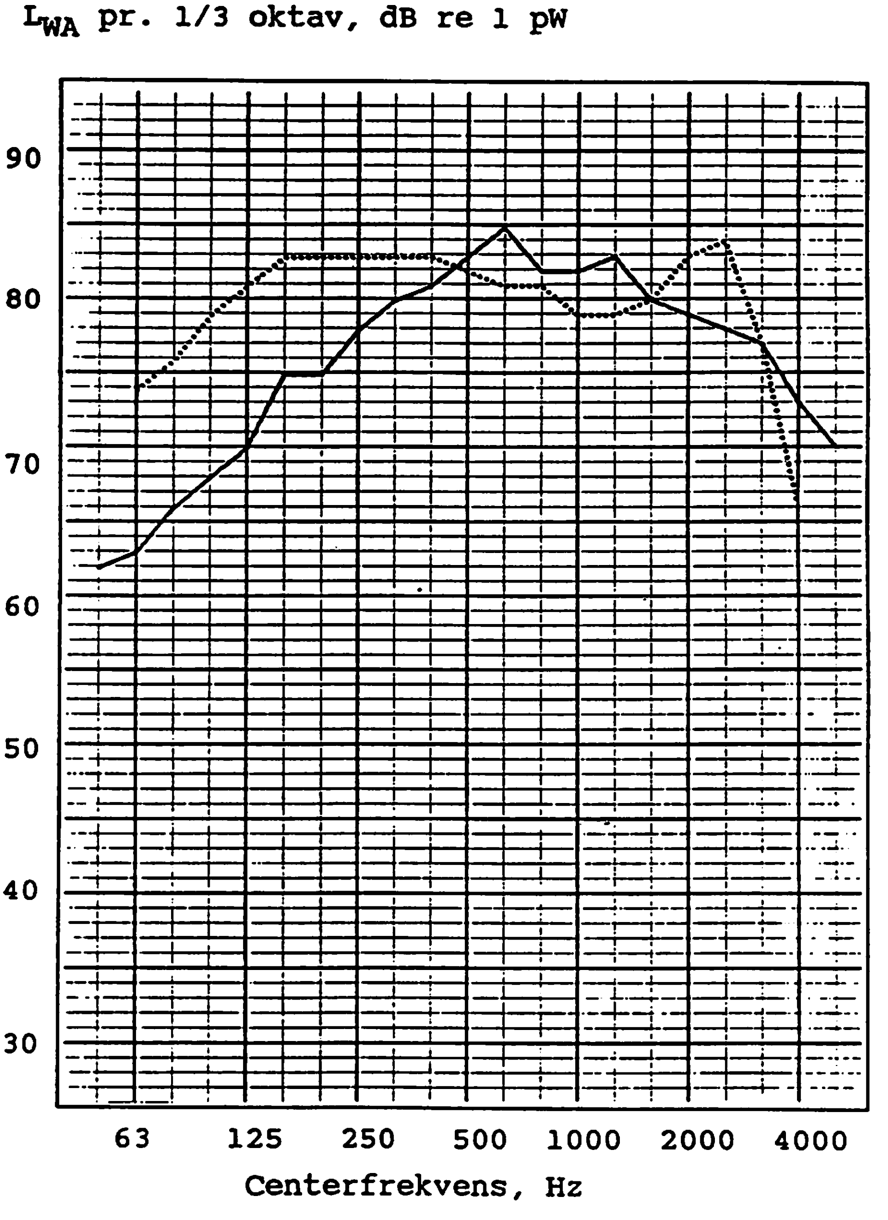


Figure 4. A-weighted sound power level per 1/3 octave for the aerodynamic noise from the rotor

—— determined as residual noise

.... predicted (ref./3/)

Figure 4 also displays a predicted noise spectrum. These computations are based on a model set up by Grosveld (ref. /3/). By personal correspondance we obtained additional information, and during 1985 the model was implemented on a minicomputer. Since then we have gathered a lot of experience from this prediction model.

4. Grosveld's prediction model.

According to the prediction model the aerodynamic noise is made up from three contributions as illustrated in Figure 5:

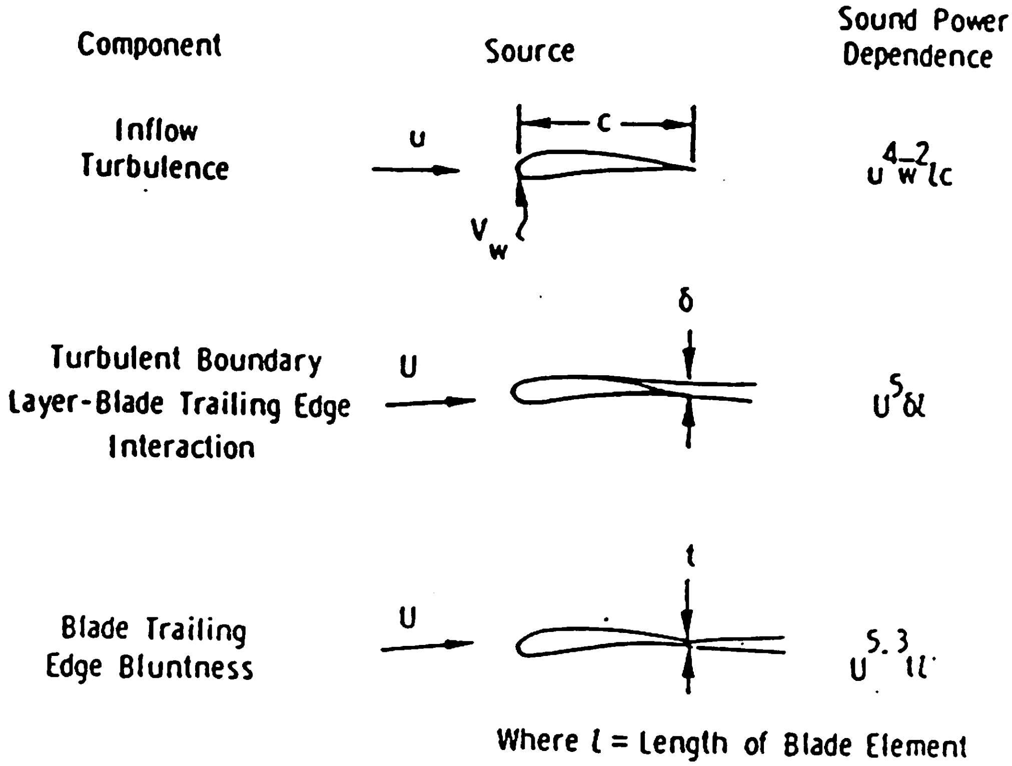


Fig. 5 Components of wind turbine broadband noise.

A) Loading fluctuations due to <u>inflow turbulence</u> interacting with the rotating blades.

B) The turbulent <u>boundary layer</u> flow over the airfoil surface interacting with the blade trailing edge.

C) Vortex shedding due to <u>trailing edge bluntness</u>.

For each of these sources a noise contribution is computed:

A: <u>inflow turbulence</u> - giving a broadband spectrum with maximum for the A-weighted octaveband level near 250 Hz.

B: <u>boundary layer</u> - giving a broadband spectrum with maximum near 1 kHz.

C: trailing edge bluntness - giving a narrow band of noise peaking at 1-5 kHz depending on the thickness of the trailing edge.

An example of computed results is given in Figure 6.

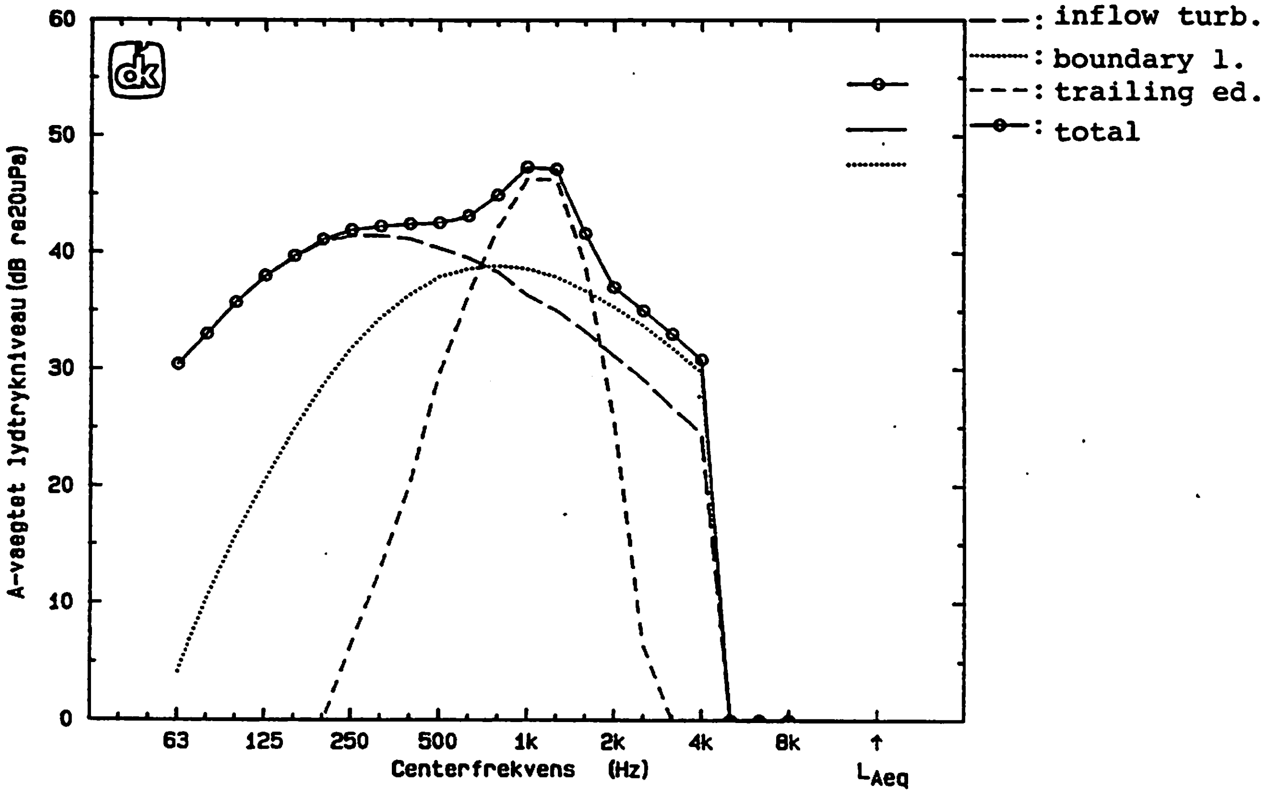


Figure 6. Contributions to the A-weighted aerodynamic noise

It is noticed that Grosveld's model does not include any noise contribution from blade-tips. Since some constants of the prediction scheme are "empirically calibrated", all modifications of the model should be thoroughly considered.

One of the most important reasons for our interest in the model was the remarkable agreement between measured and predicted noise reported by Grosveld - see Figure 7.

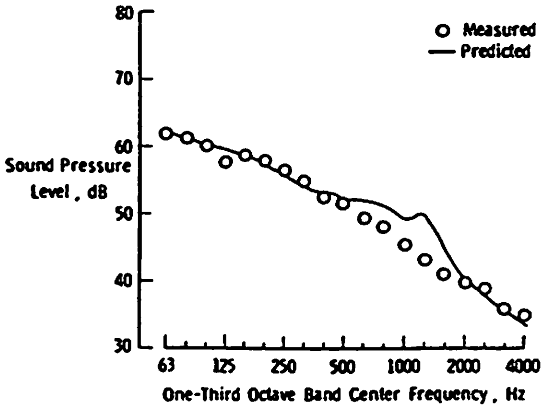


Fig. 4 Measured and predicted broadband noise spectra for the MOD-2 machine ($P = 1000$ kW, $r_o = 150$ m).

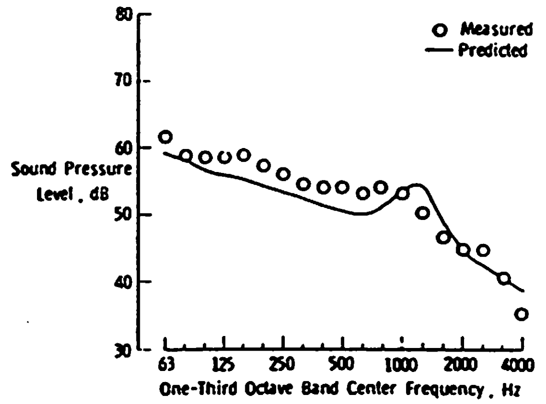


Fig. 5 Measured and predicted broadband noise spectra for the MOD-OA machine ($P = 70$ kW, $r_o = 60$ m).

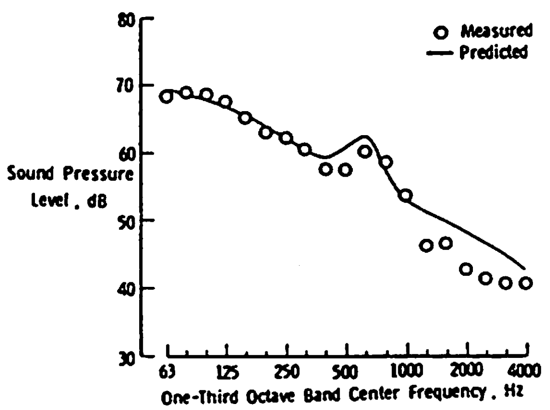


Fig. 6 Measured and predicted broadband noise spectra for the WTS-4 machine ($P = 2500$ kW, $r_o = 200$ m).

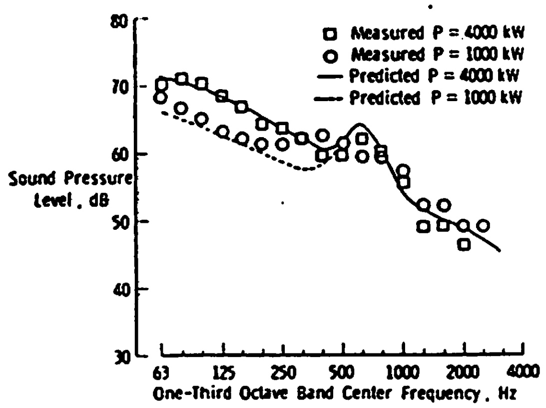


Fig. 9 Effect of output power (wind speed) on the measured and predicted broadband noise spectra of the WTS-4 machine ($r_o = 150$ m).

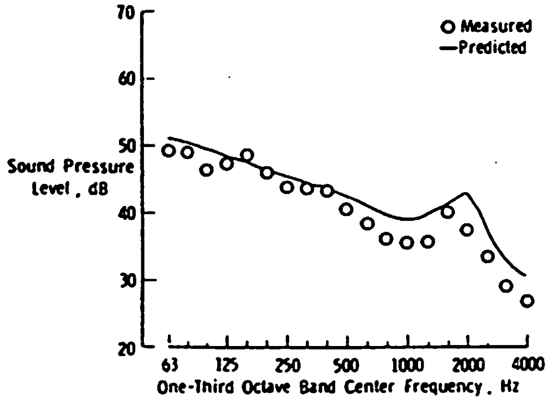


Fig. 7 Measured and predicted broadband noise spectra for the U. S. Windpower Inc. machine ($P = 25$ kW, $r_o = 91$ m).

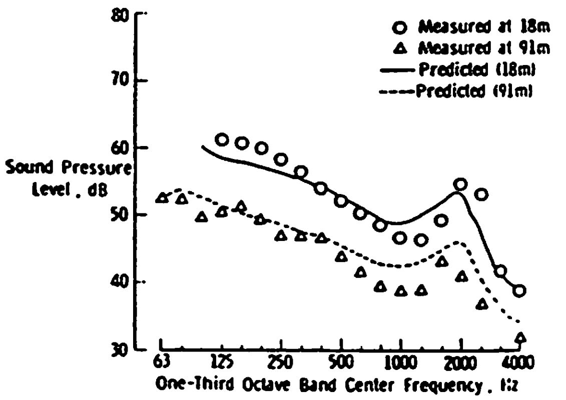


Fig. 11 Effect of distance from the U.S. Windpower Inc. machine on the measured and predicted broadband noise spectra ($P = 50$ kW).

Figure 7. Measured and predicted noise spectra from ref./3/.

Unfortunately we have not been able to reproduce this agreement in our measurements. Figure 4 shows typical great differences between measured and predicted spectra. Our experience with Grosveld's model was: the level at low frequencies (<1 kHz) is overestimated by approximately 5 dB, while the total predicted level agree reasonably well with our measurement results. It is, however, believed that the prediction of relative changes is good.

During 1988 a master's thesis from the Danish Technical University led to further correspondance with Grosveld. It then appeared that an error had occurred to our original information causing the inflow turbulence contribution to be overestimated. This may explain the low-frequency discrepancy. We also received a beta-version of a revised program for personal computers. Comparison with our original program showed however, other differences as well. The revised program seemed to be rather in-professional - so it may contain other errors. Figures 8 and 9 shows two examples of measurement results compared to predicted results using the original as well as the revised model.

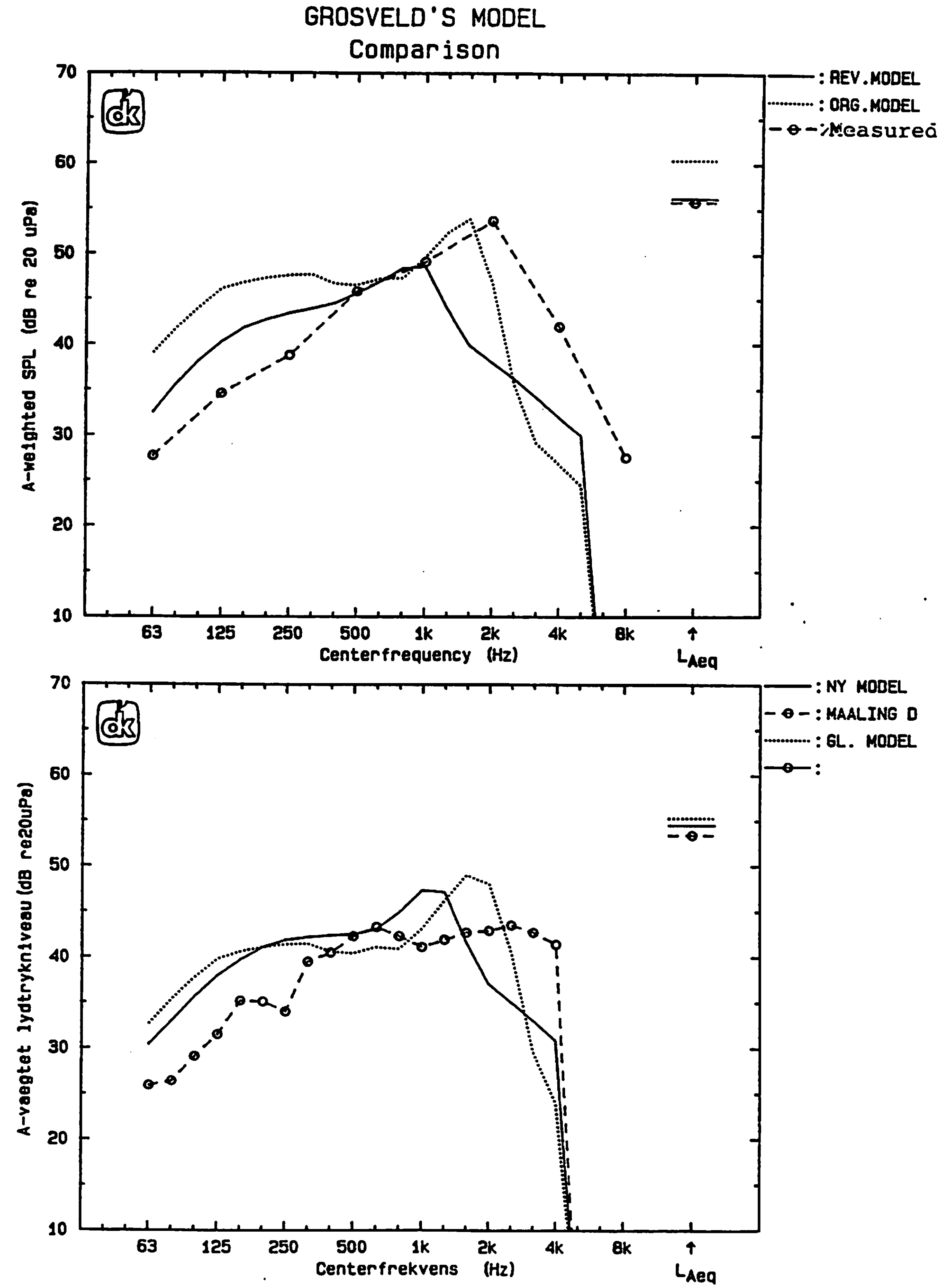


Figure 8&9. Comparison. 8:Windane 40(750kW),9:Bonus 340 kW

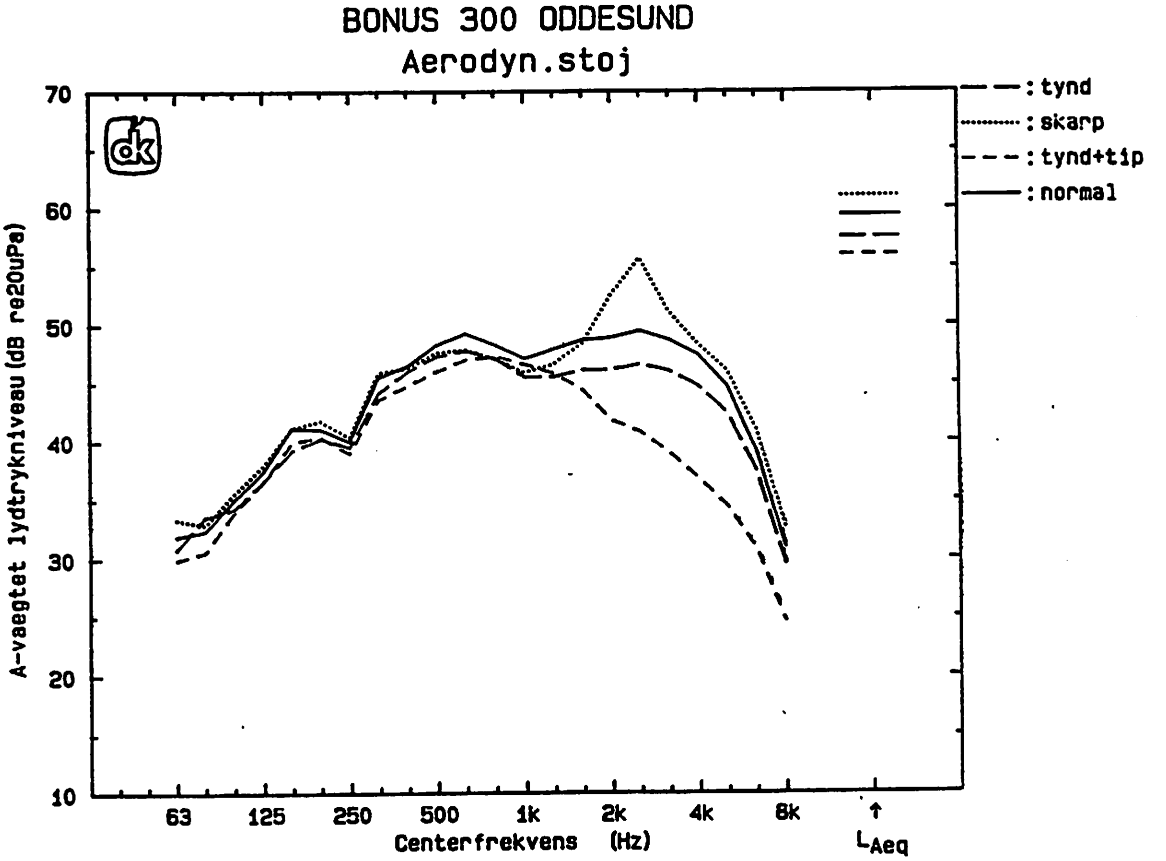
The measurement results are the total A-weighted sound power level emitted from two wind turbines for which the mechanical noise is judged to be insignificant.

5. Pilot experiment on aerodynamic noise.

A small pilot experiment was carried out on a Bonus 340 kW prototype wind turbine. The design of the windturbine is exceptional, and excessive means to prevent emission of mechanical noise have been taken. Thus elastic vibration isolators are inserted between gear/generator and tower, machine foundation, nacelle and rotor. The total A-weighted sound power level at a windspeed in 10 m height of v_{10} = 8 m/s was determined to L_{WA} = 100 dB re 1 pW. L_{WA} for the nacelle and for the tower were determined to 81 dB and 82 dB re 1 pW respectively. The noise does not contain detectable pure tone components. The rotor is a three bladed, upwind, stall regulated rotor with a diameter of 30 m, a rotational speed of 36.9 rpm, the profile series is NACA 63-200 and the hub height is 32 m. The noise was measured at a hard plate on the ground (h_m = 0) in a horizontal distance of 40 m from the tower. The trailing edge of the outer 10% of the blades were modified such that it at first was as thin as practically possible. It was then cut to a normal thickness of some millimeters but with sharp edges. Finally these edges were rounded, and the trailing edge thus obtained it's standard design.

Measurement results at v_{10} = 7.5 m/s are shown in Figure 10.

This figure also shows a spectrum measured with the thin trailing edge and with modified blade tips (a streamlined adaptor was fitted to each blade). The measurement results have not been analyzed thoroughly but agreement with the principles in Grosveld's model is obvious.



Sign.: BA

Bilag nr.: Fig. 10

Simultaneous measurements in the rotor plane showed the same trend. As expected no significant directivity is seen. The most "noiseless" blade design is however, even more "noiseless" in this direction - see Figure 11.

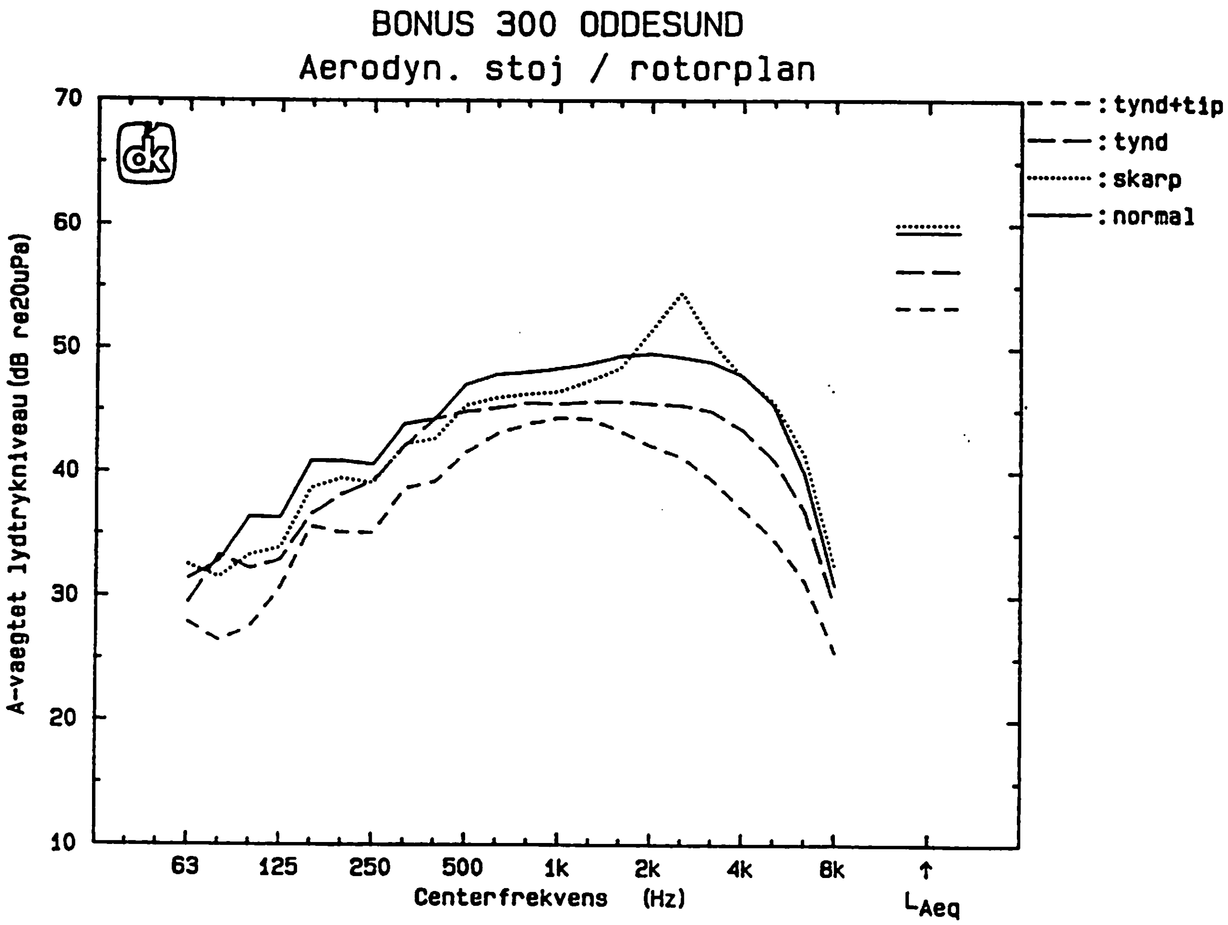


Figure 11. Measured A-weighted sound pressure level in the rotor-plane of Bonus 340 kW wind turbine (d=40m).

6. Future projects.

The recent years Danish research project have primarily aimed at attenceating the mechanical noise from the machinery of the wind turbines. In addition fundamental investigations on noise emission and on the masking effect of windnoise on the noise from wind turbines have been carried out. In the next 2 or 3 years a big collaboration project on aerodynamic noise from wind turbines will be carried out. The project is financed by the Danish Ministry of Energy and by EEC (Joule programme).

A fundamental theoretical investigation of the noise generating mechanisms will be supplemented by methodical empirical investigation on the influence of trailing edge design, tip design, tip speed, surface roughness etc. Furthermore the structure-borne noise emitted from the rotor will be examined. The aim of the project is to establish design rules for low noise wind turbine rotors.

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COMPUTATIONAL PROCEDURES FOR ROTATING BLADE NOISE PREDICTION

E. De Bernardis
D. Tarica

Summary

A computational method for the prediction of noise generated by open rotors is proposed in this paper. The theoretical procedure leading to the solution of the Ffowcs Williams-Hawkings equation is presented first. The a few results are shown, which was obtained by coding the final expression of the formal solution: it includes separate contributions from different sources of noise on the rotor blade surface.

Introduction

Recent advances in the research area of rotor Aeroacoustics are mostly related to Aeronautics. In fact current developments of propellers and helicopter rotors call for a considerable effort in order to better understand high speed effects — up to the transonic range — on the radiated noise. To face these problems a great deal of theoretical studies has been carried out in the last few years, leading to reliable computational procedures for the solution of inhomogeneous wave equation, both in the time domain and frequency domain. The expertise provided by this work allows achieving very high accuracy in the evaluation of the sound field generated by low speed rotors. Furthermore, the inherent capability of dealing with unsteady motions makes the time domain methods particularly suitable to handle such problems as yawing motion and unsteady inflow, arising in operating conditions of wind turbines. Some typical results and possible further developments are presented to suggest the application of the above-mentioned numerical procedures as a tool for predicting the main features of the sound generated by wind turbines.

Mathematical model

Starting point of the analysis is the Ffowcs Williams-Hawkings (FW-H) equation [1]:

$$\begin{aligned}\frac{1}{c_0^2}\frac{\bar{\partial}^2\tilde{\rho}}{\partial t^2}-\frac{\bar{\partial}^2\tilde{\rho}}{\partial x_i\partial x_i}=c_0^2\bar{\Box}^2\tilde{\rho} &= \frac{\bar{\partial}}{\partial t}[\rho_0 v_n\delta(f)] \\ &- \frac{\bar{\partial}}{\partial x_i}[\tilde{P}_{ij}n_j\delta(f)] \\ &+ \frac{\bar{\partial}^2}{\partial x_i\partial x_j}\left[(\rho u_i u_j+\tilde{P}_{ij}-c_0^2\tilde{\rho}\delta_{ij})H(f)\right]\end{aligned} \tag{1}$$

This is in the line of the theory of the acoustic analogy proposed, in the early 50's, by M.J. Lighthill [2] to study the problem of high speed jet noise.

The FW-H equation describes the sound generation by bodies immersed in a fluid flow: it can be easily obtained by rearranging the conservation laws for mass and momentum written in their complete form. In equation (1): $\tilde{\rho}=\rho-\rho_0$ is the density perturbation with respect to the undisturbed conditions ρ_0; $v_n=v_in_i$ is the normal velocity of a moving body whose surface is represented by the equation $f(x_i,t)=0$, v_i being the local velocity of the surface and $n_i=\partial f/\partial x_i$ its unit normal vector; u_i is the fluid velocity, with respect to a rest frame (where each point in space is denoted by a coordinate x_i); $\tilde{P}_{ij}$ represents the compressive gauge stress tensor in the fluid: $\tilde{P}_{ij}=\tilde{p}\delta_{ij}+2\mu E_{ij}$, with $\tilde{p}=p-p_0$ the pressure perturbation and E_{ij} the strain tensor. The Dirac delta function $\delta(f)$ states that the first two terms at the right-hand side of eq.(1) are nonzero only on the surface $f=0$, while the Heaviside function $H(f)$ points out that the third term only exist for $f>0$. Overbar on the differential operators denotes generalized derivatives [3,4].

Equation (1) can be simplyfied if the following assumption are made:

i) the body is the only source of disturbance within the fluid, and the perturbations generated are amall enough to allow neglecting terms that are nonlinear functions of some flow variables

ii) effects of viscosity can be neglected both in the flow and at the body surface: the fluid-body interaction can be described by the scalar field of surface pressure

iii) the flow is isentropic; the pressure perturbation $\tilde{p}=p-p_0$ is represented in the far field by its linear approximation $c_0^2\tilde{\rho}$.

In the above hypotheses the (volume) source term

$$\frac{\bar{\partial}^2}{\partial x_i \partial x_j}\left[(\rho u_i u_j + \tilde{P}_{ij} - c_0^2 \tilde{\rho}\delta_{ij})H(f)\right] \tag{2}$$

may be neglected, while the term:

$$\frac{\bar{\partial}}{\partial x_i}[\tilde{P}_{ij} n_j \delta(f)] \tag{3}$$

may be written in the form:

$$\frac{\bar{\partial}}{\partial x_i}[\tilde{p}_s \delta_{ij} n_j \delta(f)] = \frac{\bar{\partial}}{\partial x_i}[\tilde{p}_s n_i \delta(f)] \tag{4}$$

where: $\tilde{p}_s$ is the gauge pressure on the body surface.

The Ffowcs Williams-Hawkings equation can then be written in the form:

$$\begin{aligned} \bar{\square}^2 \tilde{p} &= \frac{\bar{\partial}}{\partial t}[\rho_0 v_n \delta(f)] \\ &- \frac{\bar{\partial}}{\partial x_i}[\tilde{p}_s n_i \delta(f)] \end{aligned} \tag{5}$$

where only surface source terms appear, containing linear functions of the flow variables.

Using the free-space Green's function for the wave equation, the solution of equation (5) can be written as

$$\begin{aligned} 4\pi\tilde{p}(x_i,t) &= \frac{\bar{\partial}}{\partial t}\int_{-\infty}^{t}\iiint \frac{\rho_0 v_n}{r}\delta(f)\delta(g)dV(y_j)d\tau \\ &- \frac{\bar{\partial}}{\partial x_i}\int_{-\infty}^{t}\iiint \frac{\tilde{p}_s n_i}{r}\delta(f)\delta(g)dV(y_j)d\tau \end{aligned} \tag{6}$$

where x_i and y_j represent observer and source positions respectively while t and τ are the corresponding time variables; besides: $g = \tau - t + r/c_0$, where $r = |r_i| = |x_i - y_i|$ is the distance between source and observer.

The delta functions in the integrals at the right hand side of the equation (5) can be treated to transform the formula to a surface integral expression. If the source surface is rigid then a surface-fixed frame of reference can be defined such that each surface point is denoted by a time independent coordinate, say η_j. Using this representation equation (5) can be turned into the form [5]:

$$\begin{aligned} 4\pi\tilde{p}(x_i,t) &= \frac{\bar{\partial}}{\partial t}\iint_S \left[\frac{\rho_0 v_n}{r|1-m_r|}\right]_{\tau=\tau^*} dS(\eta_j) \\ &- \frac{\bar{\partial}}{\partial x_i}\iint_S \left[\frac{\tilde{p}_s n_i}{r|1-m_r|}\right]_{\tau=\tau^*} dS(\eta_j) \end{aligned} \tag{7}$$

In the above equation τ^* is the emission (retarded) time, it is obtained — for each η_j — from the solution of the equation

$$|x_i - y_i(\eta_j,\tau)| = c_0(t-\tau) \tag{8}$$

while $m_r = m_i\hat{r}_i$ where $m_i = v_i/c_0$ and $\hat{r}_i = r_i/r$ is the unit vector in the source–observer direction.

Equation (7) describes two main effects in noise generation from subsonic rotor. The first term accounts for the effect of fluid displacement due to the body motion (thickness noise): calculating this requires knowledge of the geometric and kinematic feature of the source motion. The second term (loading noise) comes from the pressure distribution on the boby surface: then the aerodynamic problem is to be solved in order to provide data required by this noise prediction method.

Application of Farassat's formulation 1

Looking at equation (7) the derivatives appearing before the integrals is to be treated numerically with cure. It is possible to have a simpler form in the application of these methods, leading to a different formulation of rotor noise; it is based on the property of the fundamental solution of the wawe equation espressed by:

$$\frac{\bar{\partial}}{\partial x_i}\left[\frac{\delta(g)}{r}\right] = -\frac{1}{c_0}\frac{\bar{\partial}}{\partial t}\left[\frac{\hat{r}_i\delta(g)}{r}\right] - \frac{\hat{r}_i\delta(g)}{r^2} \tag{9}$$

A trasformation of equation (7) can be carried out which leads to the following expression for the sound field [6]:

$$\begin{aligned} 4\pi\tilde{p}(x_i,t) &= \frac{1}{c_0}\frac{\bar{\partial}}{\partial t}\iint_S \left[\frac{\rho_0 c_0 v_n + \tilde{p}_s n_i\hat{r}_i}{r|1-m_r|}\right]_{\tau=\tau^*} dS(\eta_i) \\ &+ \iint_S \left[\frac{\tilde{p}_s n_i\hat{r}_i}{r|1-m_r|}\right]_{\tau=\tau^*} dS(\eta_i) \end{aligned} \tag{10}$$

A relevant feature of this expression is the splitting of the loading noise in two contributions, showing the dipole nature of this source which gives rise to the appearance of a far field (represented by the terms involving $1/r$) and a near field (involving $1/r^2$).

Finally some numerical results are showed: they have been obtained applying equation (10) to a rotating blade resembling the geometric configuration and kinematical features of a wind turbine.

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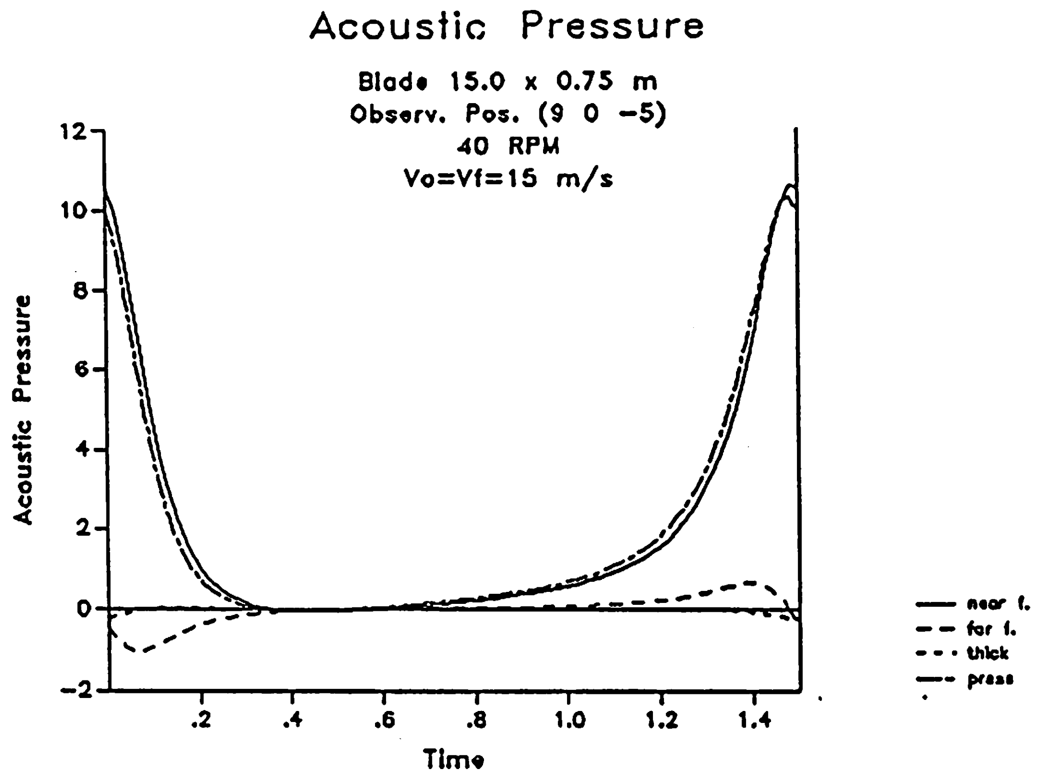
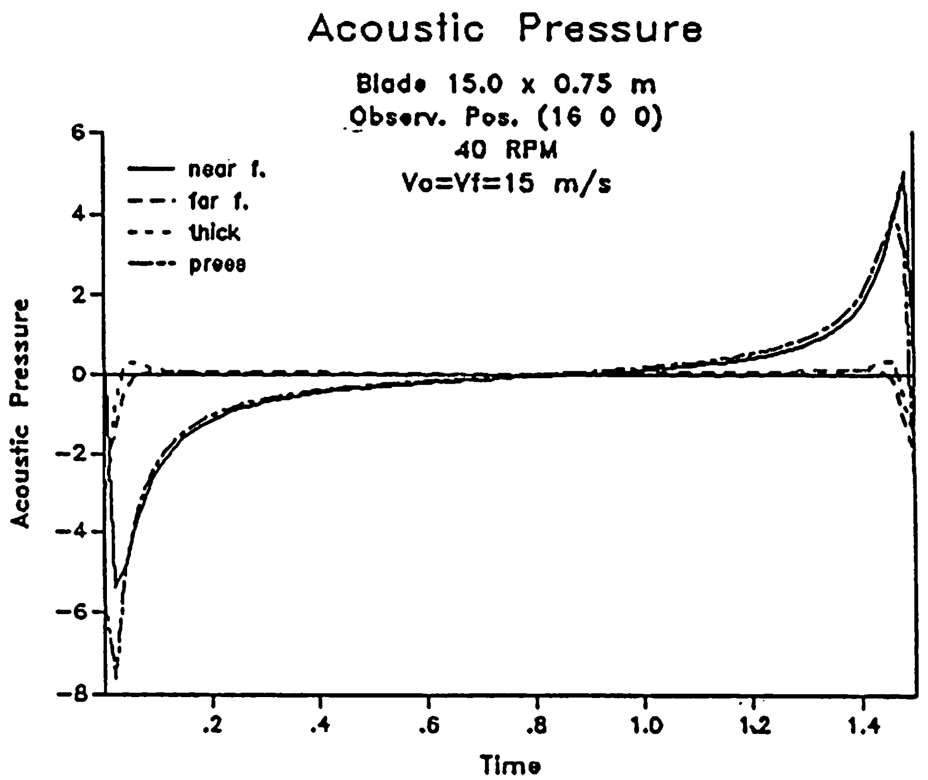


Figure 1:



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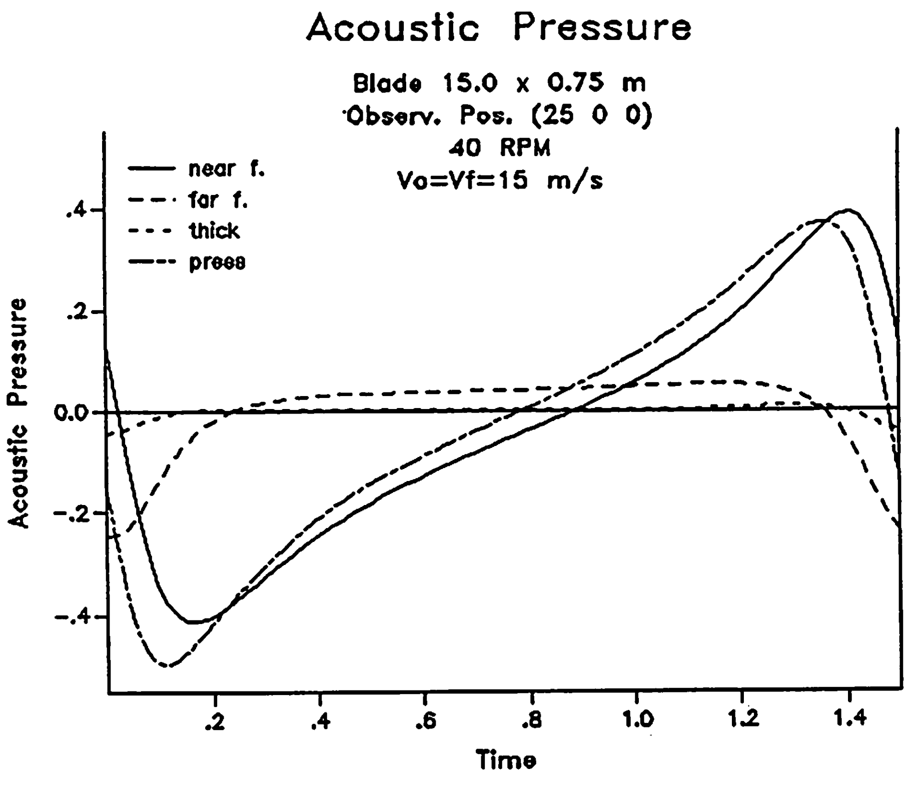

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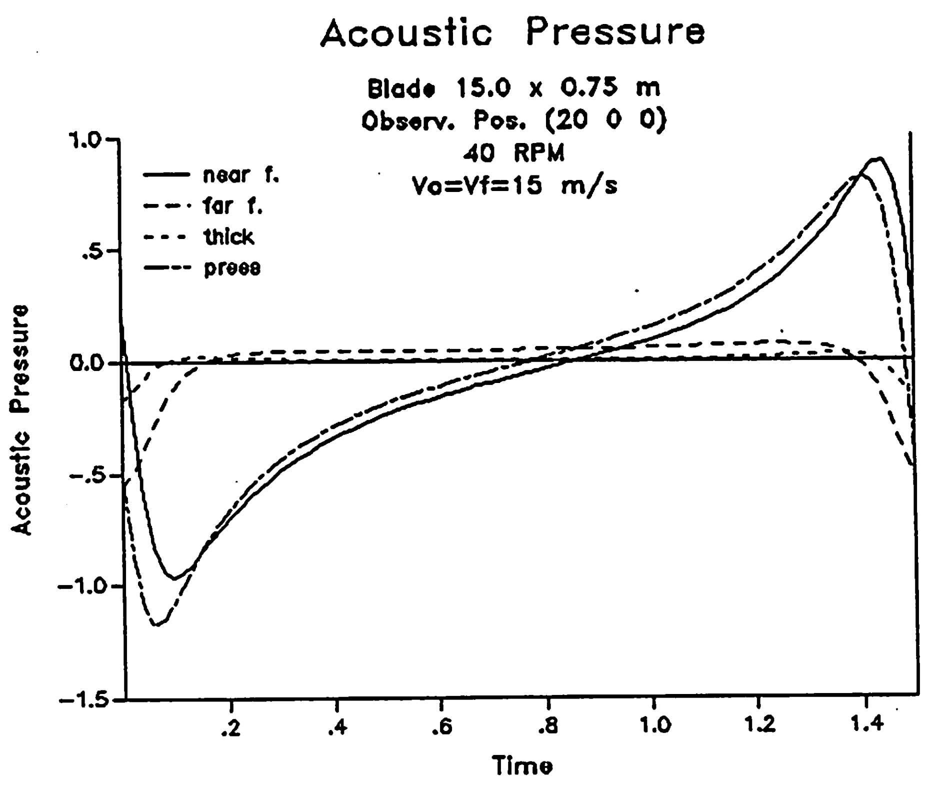


Figure 4:

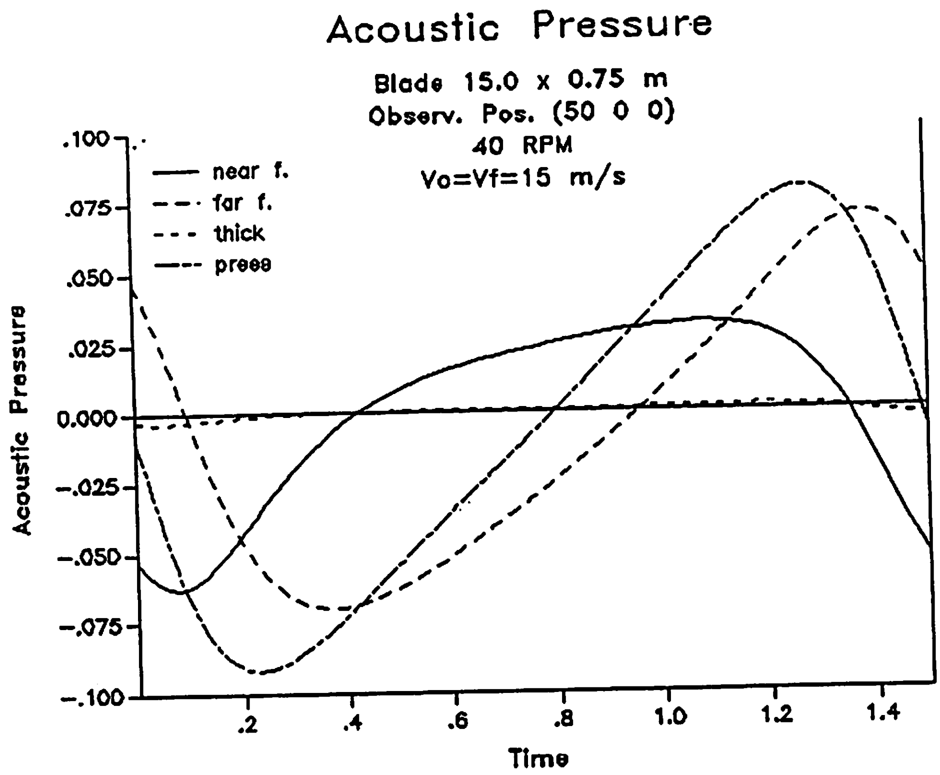


Figure 5:

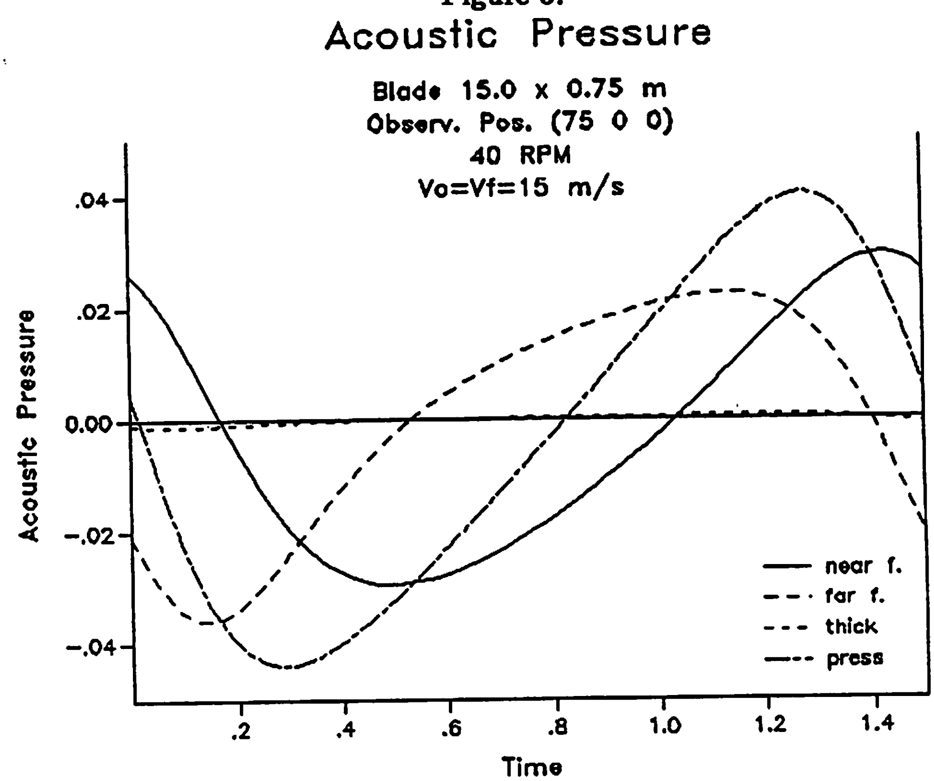


Figure 6:

TONALITY AND IMPULSIVITY OF THE ACOUSTIC NOISE FROM WIND TURBINES

N. van der Borg
M. Lendi

1. Introduction

The acoustic noise originating from wind turbines causes restrictions in the realisation of wind energy projects. For the Netherlands limits have been defined for the sound pressure levels immitted into the build-up area by industrial activities. The limits are dependent of the time of day and also of the type of noise. In case the noise is audible impulsive and/or tonal, the limits are 5 dB(A) lower than in case no tonality or impulsivity can be heard.
This causes extra restrictions in the realisation of wind energy projects because the noise from wind turbines is very often tonal and impulsive. In this paper various types of wind turbine noise, as measured by ECN, are described and the mechanisms that generate the observed tonality and impulsivity are discussed in chapter 2 and 3 respectively.

2. Tonality

In the sound pressure spectrum, measured near a wind turbine, in most cases sharp peaks can be seen with a significant contribution to the total sound pressure which is an indication that the sound is tonal. The possible mechanisms that can cause tonality are:
- misalignement,
- tooth engagement of the gear wheels,
- aerodynamical whistle from the rotor blades,
- electro-magnetic forces in the generator and
- resonances in the construction.

These possible mechanisms are discussed in the next paragraphs.

2.1. Misalignment

The drive train of a wind turbine is made of the rotor, the main shaft, the gear box, the high speed shaft and the generator. In case the drive train is not properly aligned or a shaft is bent, a periodical force is introduced which causes vibrations in the construction. The construction will radiate acoustic noise with a frequency equal to the rotation frequency of the concerned shaft and possibly also higher harmonics. In figure 1 a sound spectrum (in 1/24 octave bands) is given as measured near a wind turbine with a badly aligned drive train. In the spectrum the frequency of the high speed shaft (25 Hz) is clearly visible and also many higher harmonics (50, 75, 100, 125 Hz etc).

2.2. Tooth engagement of the gear wheels

In most cases a wind turbine has a gear box with (two or) three stages. Each stage consists of gear wheels of which the teeth catch each other with a certain frequency. This results in a periodical vibration which is far from sine-shaped. The concequence of this is the generation of acoustic noise with a frequency equal to the tooth engage frequency and many higher harmonics. This effect is observed in almost all measured spectra. Examples are given in figures 2 and 3. The calculated tooth engage frequencies of the turbines in these examples (rotational frequencies of the gear shafts multiplied by the corresponding number of teeth) are indicated in the figures. These frequencies and also the higher harmonics (integer multiples of the frequencies) are clearly present in the spectra. The tooth engage frequencies of the first stage are not visible in the spectra but the higher harmonics of these frequencies can be seen. The tooth engage frequency of the third stage in the example of figure 3 cannot be seen because it coincides with a significant contribution of aerodynamical noise to the total noise. However, the highest peak in the spectrum has a frequency equal to three times the tooth engage frequency of the third gear and is consequently caused by the third stage.

2.3. Aerodynamical whistle from the rotor blades

A rotating blade can cause a wisthling sound due to slits or cavities in the blade (e.g. from aerodynamical brakes or water-outlet openings).
This source of acoustic noise can easily be recognised by listening

to the sound because of the Doppler-effect that is introduced by the movement of the source. As an example of this the sound spectrum at a fixed position in the rotor plane of a turbine with one wistling blade has been analysed for four different rotor positions (45, 135, 225 and 315 degrees with respect to an arbitrary reference position). The part of the spectra (in 1/12 octave bands) around the frequency of the observed wistle is given in figure 4 for the four mentioned rotor positions. From the figure it can be seen that the frequency varies between 3070 Hz and 3650 Hz. The tip speed of the turbine was about 33 m/s which gives a calculated variation in the frequency of about 22%, assuming that the sound source was located at the tip of the wistling blade. This corresponds rather good with the variations in the frequency (fig. 4) of about 19%.
Another aerodynamical mechanism that causes (rather broad) peaks in the spectra is the noise due to the blunt trailing edge of the blades. This effect, however, causes a peak that is not narrow enough to call the noise tonal. As an example of this the sound spectrum of a turbine with a blunt trailing edge is given in the figures 5 and 6. After the measurement of the spectrum of figure 5 the turbine manufacturer made a spoiler near the trailing edge of each blade in order to diminish the total noise. This changed the sound spectrum into the spectrum of figure 6 from which can be concluded that the broad peak that shifted from about 1.4 kHz to about 550 Hz was cauded by the trailing edge of the blades.

2.4. Electro-magnetic forces in the generator

When the generator of the wind turbine is producing energy the electric current in the generator causes periodical electro-magnetic forces that can introduce vibrations in the generator and thus acoustic noise. The frequency of this noise can be the frequency of the electric grid (in Europe 50 Hz) and the higher harmonics. An extra contribution to the third higher harmonic can be expected because the generators are in most cases three phase machines. Furthermore the forces might trigger mechanical resonances in the generator. Noises due to electro-magnetic forces with frequencies of 50 or 150 Hz, however, have not (yet) been observed clearly by the authors. This does not necessarily mean that the phenonemum does not exist because its possible contribution to the 50 Hz and 150 Hz in the spectrum can not be distinguished from the harmonics of the high speed shaft frequency (see paragraph 2.1). The possible resonances due to electro-magnetic forces are not expected to exist during idling of the generator (e.g. during the start-up procedure or during low wind periods). The authors have not (yet) observed a peak in the sound spectrum that rises suddenly at the moment of grid connection which means that resonances in the generator due to electro-magnetic forces have not (yet) been observed.

2.5. Resonances in the construction

A turbine construction can have vibration modes with eigenfrequencies that are in the audible range (e.g. local deformations in the nacelle or tower). It is possible that a vibration mode is triggered by forces in such a way mechanical that resonances start in part(s) of the construction. This results in acoustic noise with a frequency equal to the eigenfrequency of the concerned vibration mode. Possible peaks in the sound spectrum that are caused by this phenomenum can be distinguished from gear box noise (see paragraph 2.2) by measuring a series of sound spectra during the start-up procedure of the turbine. Peaks with frequencies that are independent on the rotational frequency of the turbine are introduced by resonances in the construction in contrast with gear box noise. The authors have not (yet) observed significant peaks in the measured sound spectra that are caused by resonances in the construction.

3. IMPULSIVITY

The total sound level or certain part(s) of the sound spectrum, measured at a fixed position near a wind turbine can have values that vary periodically in time with a frequency equal to the rotational frequency of the turbine rotor or integer multiples of this frequency. This effect is called impulsivity. Impulsivity is almost always observed in the acoustic noise from wind turbines. The impulsivity of aerodynamical noise and of mechanical noise are discussed in the paragraphs 3.1. and 3.2. respectively.

3.1. Impulsivity of aerodynamical noise

The contribution of aerodynamical noise to the total sound level can be estimated by assuming that the broad frequency part of the sound spectrum is caused by aerodynamical noise only. As an example of impulsivity of aerodynamical noise the total sound level has been measured near a turbine that generated a sound spectrum with only aerodynamical noise. The total sound level has been measured during short intervals (1/8 of the rotor revolution time). The sound level as a function of time is presented in figure 7 in which can be seen clearly that the sound is impulsive. The modulation frequency is equal to the rotor revolution frequency multiplied with the number of rotor blades. This can be due to variations in the source power level (wind shear, tower passage) or due to the directivity of the aerodynamical noise in combination with the rotating sound source. These possible mechanisms could not be distinguished because no simultaneous measurements at different positions have been performed.

3.2. Impulsivity of mechanical noise

The contribution of mechanical noise to the total sound level can be recognized by sharp peaks in the sound spectrum (see chapter 2). As an example of impulsivity of mechanical noise the level of a tooth engage peak in the sound spectrum near a wind turbine has been measured during a series of short measurement intervals (1/8 of the rotor revolution time). The sound level in this discrete (1/24 octave

band) peak as a function of time is presented in figure 8 in which can be seen clearly that the sound is impulsive. The modulation frequency in this example is equal to the rotor revolution frequency multiplied with the number of rotorblades but measurements near other wind turbines have shown that modulation frequencies equal to the revolution frequency can occur as well. The impulsivity presented in figure 8 has been observed at a up-wind position of the turbine. Under the same conditions half an hour later impulsivity has been observed in the rotor plane at the same distance from the same turbine (see figure 9). The modulation frequency is equal for both positions but the pattern is not. The exact reason of the observed impulsivity and the reason of the different modulation patterns at different positions is not (yet) known. The reason of this impulsivity could be that the blades act as a sound board that radiate the mechanical noise with a certain directivity or that the blades shield the noise radiated by the nacelle periodically (variations in the sound path). Another reason could be that the source power varies periodically (variations in the source).
For checking this last possible effect an in-door experiment has been performed using the rotor shaft driving facility of ECN (called the RAAF) connected to the gear box of a wind turbine. The mechanical vibrations at the surface of the gear box have been measured with a piezo-accelerometer during short measurement periods. The level of the peak in the spectrum of the surface velocity (obtained by integrating the accelerometer signal) at a tooth engage frequency as a function of time is presented in figure 10. In this figure can be seen that the vibration level is modulated with the revolution frequency of the primary shaft. In this experiment the impulsivity can only be caused by variations in the source. From this it can be concluded that impulsivity of the mechanical noise from wind turbines can be caused by variations in the source. This conclusion does not exclude the possibility that also variations in the sound path causes impulsivity.

4. CONCLUSION

The phenomena tonality and impulsivity of wind turbine noise can form serious restrictions in the realisation of wind energy projects because both phenomena are certainly no exception. In most cases tonality originates from the tooth engagement of the gearbox. Impulsivity is very often observed in both aerodynamical noise and mechanical noise. In both cases the impulsivity can be caused by periodical variating in the sound path and also by periodical variations in the sound source power.

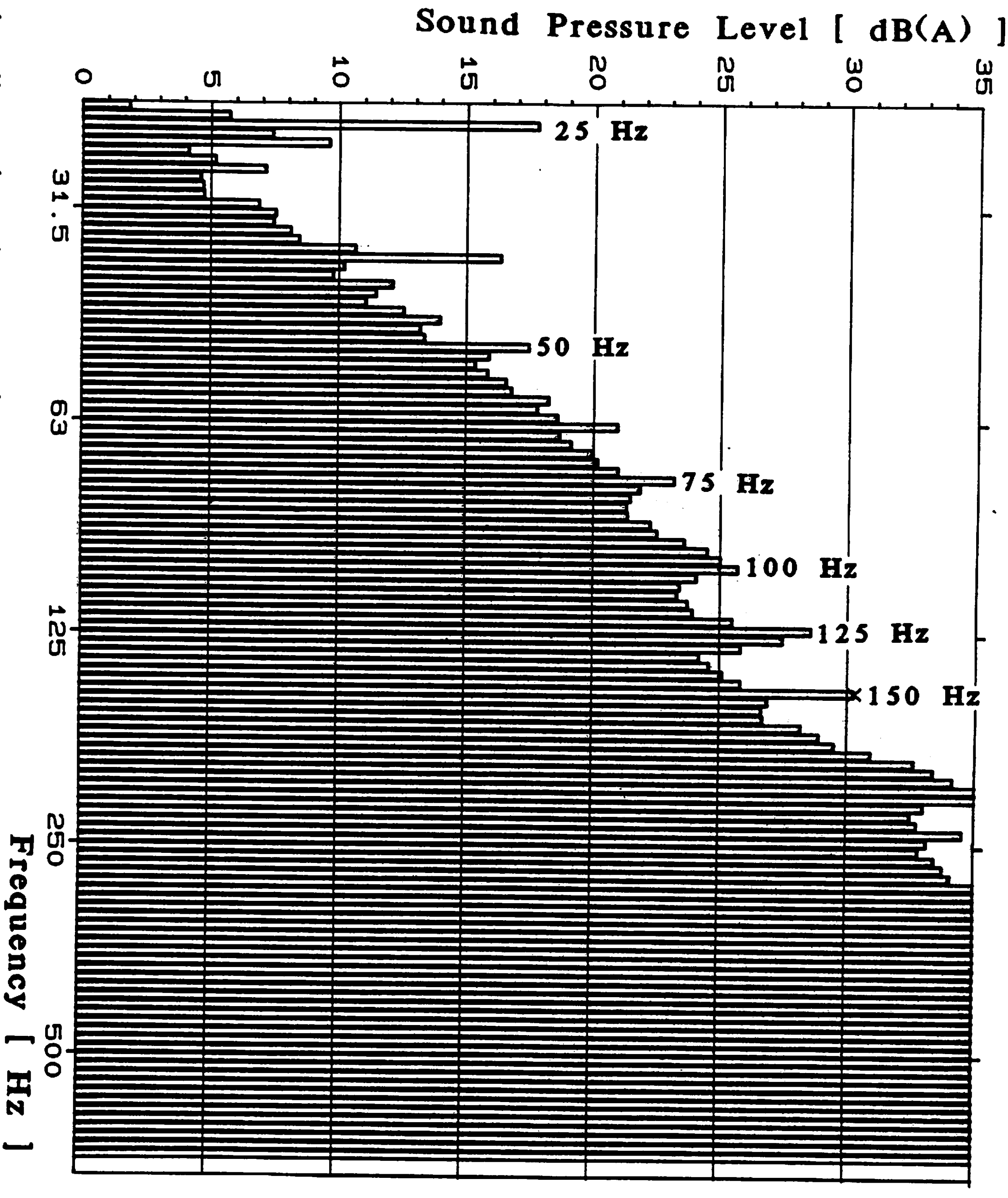


Fig. 1. Measured sound spectrum showing frequencies due to misalignment.

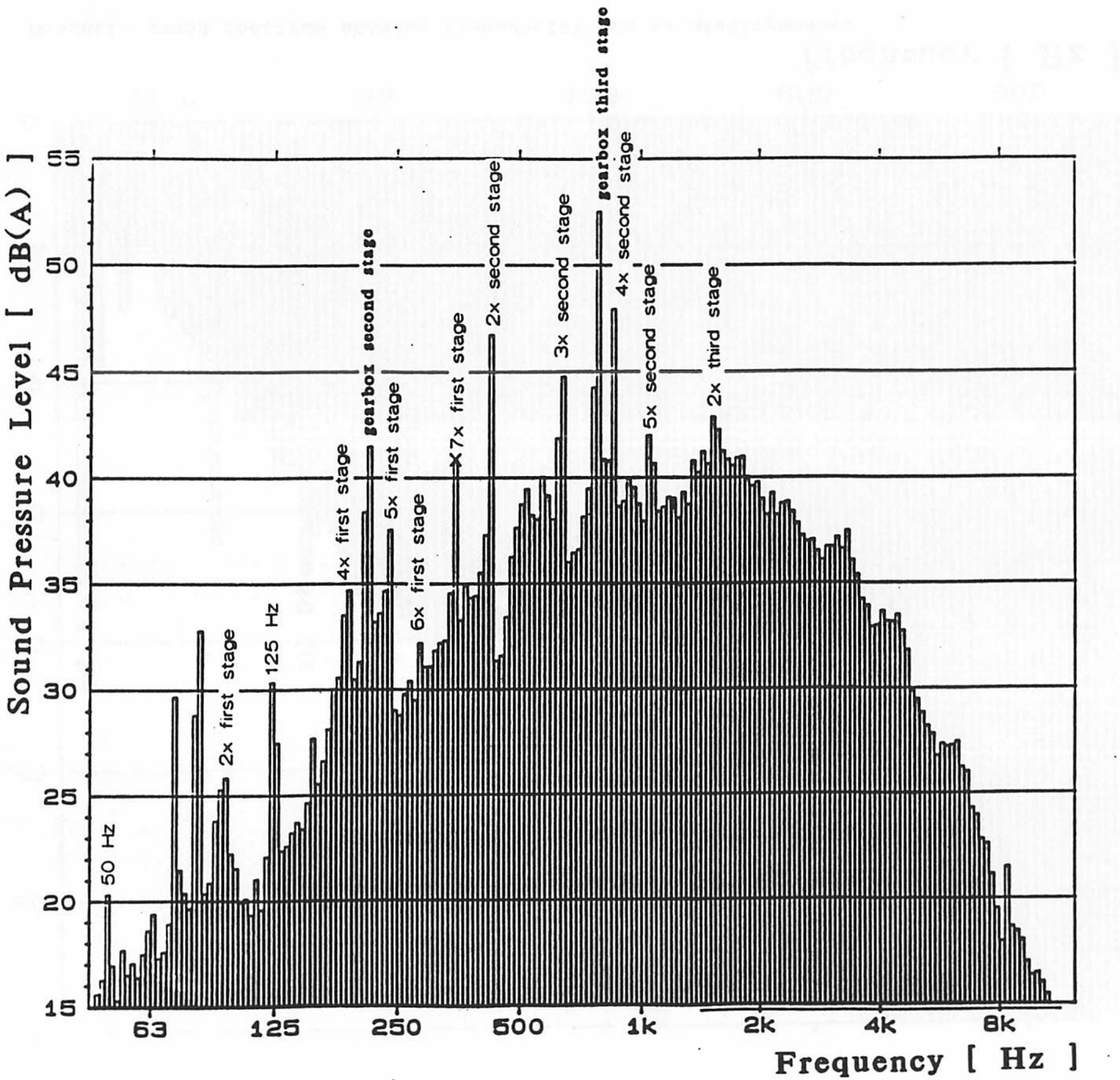


Fig. 2. Measured sound spectrum showing tooth engage frequencies.

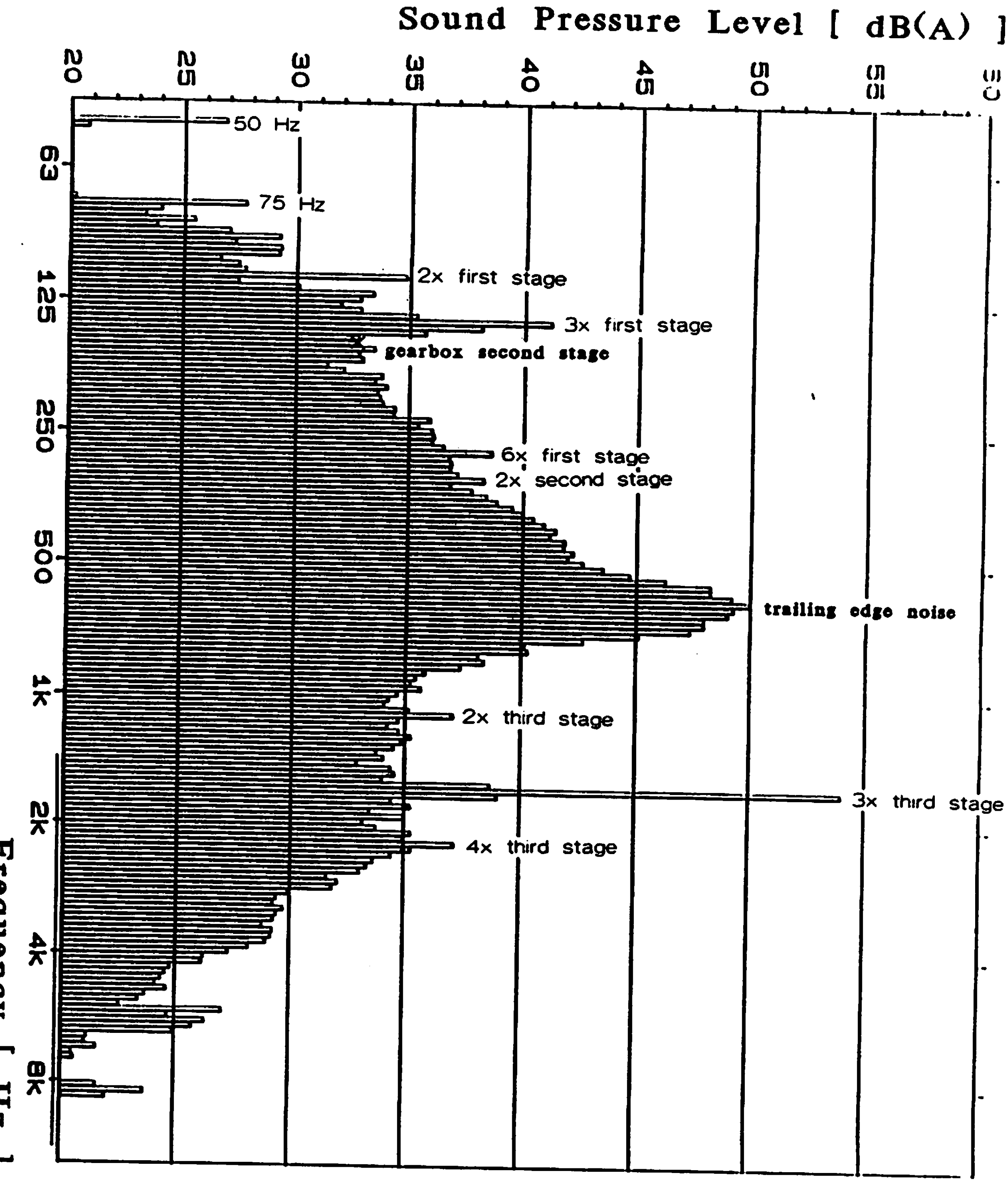


Fig. 3. Measured sound spectrum showing tooth engage frequencies.

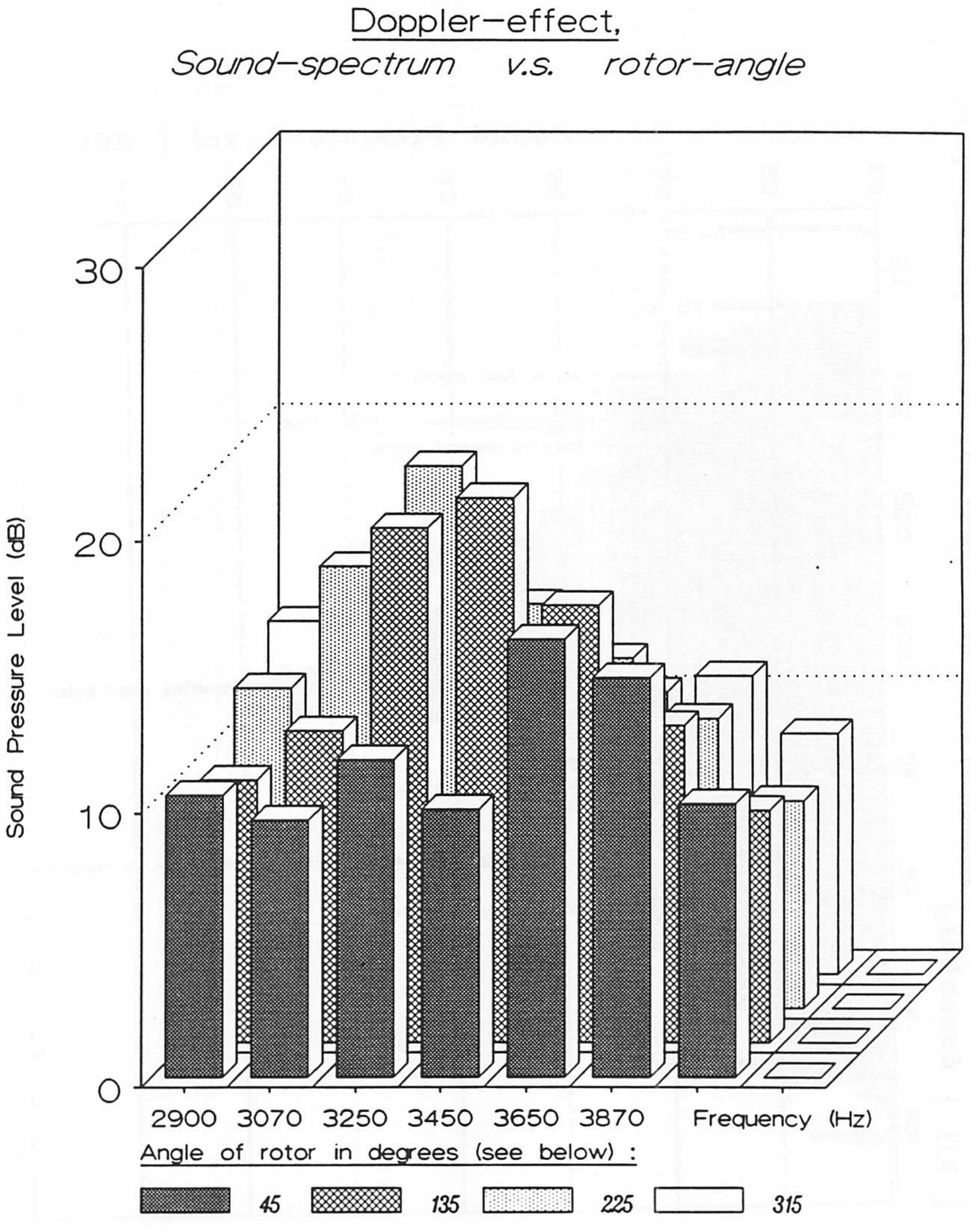


Fig. 4. Measured sound spectra during four different rotor positions showing Doppler effect.

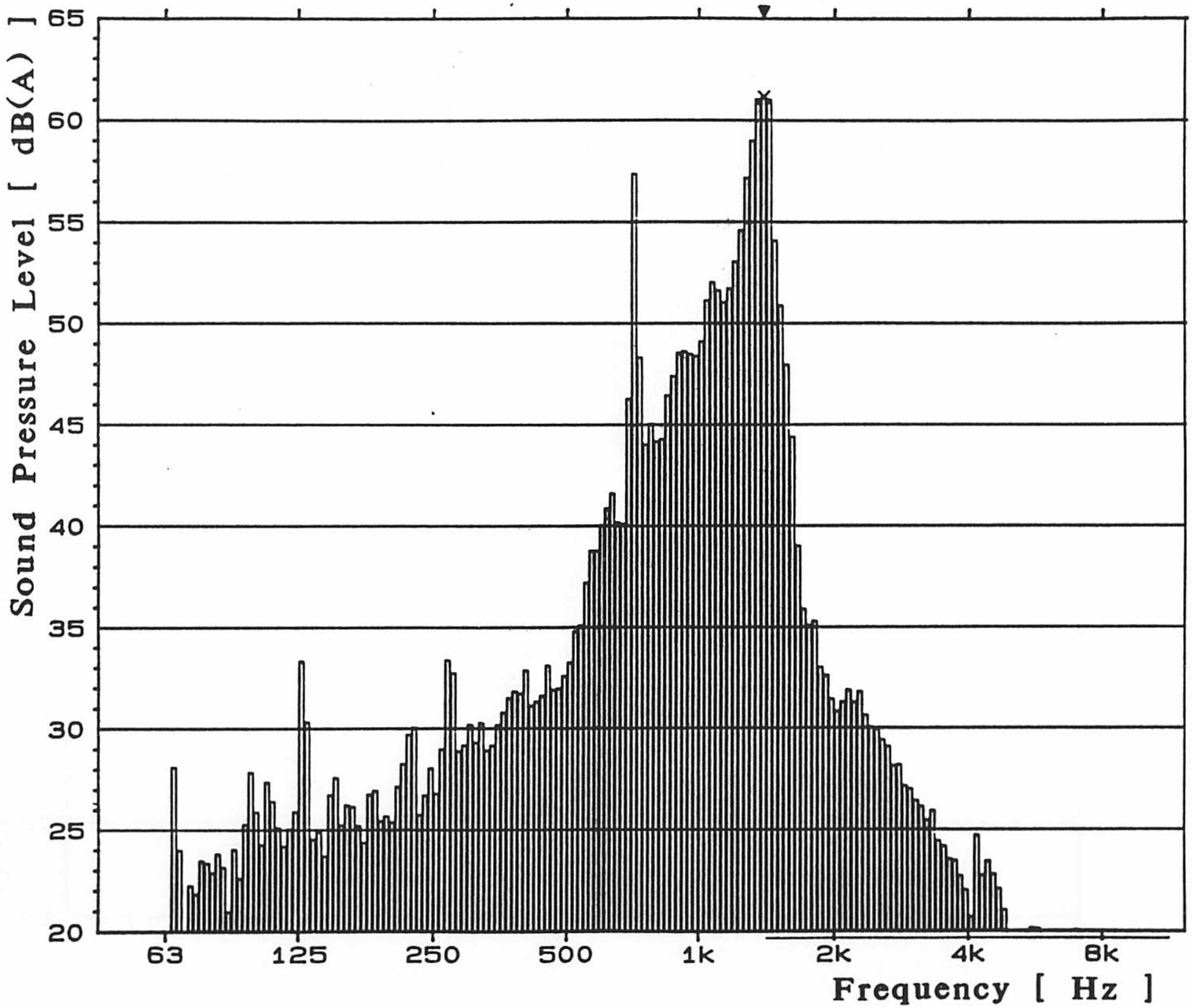


Fig. 5. Measured sound spectrum showing noise due to the blunt trailing edge (without spoiler).

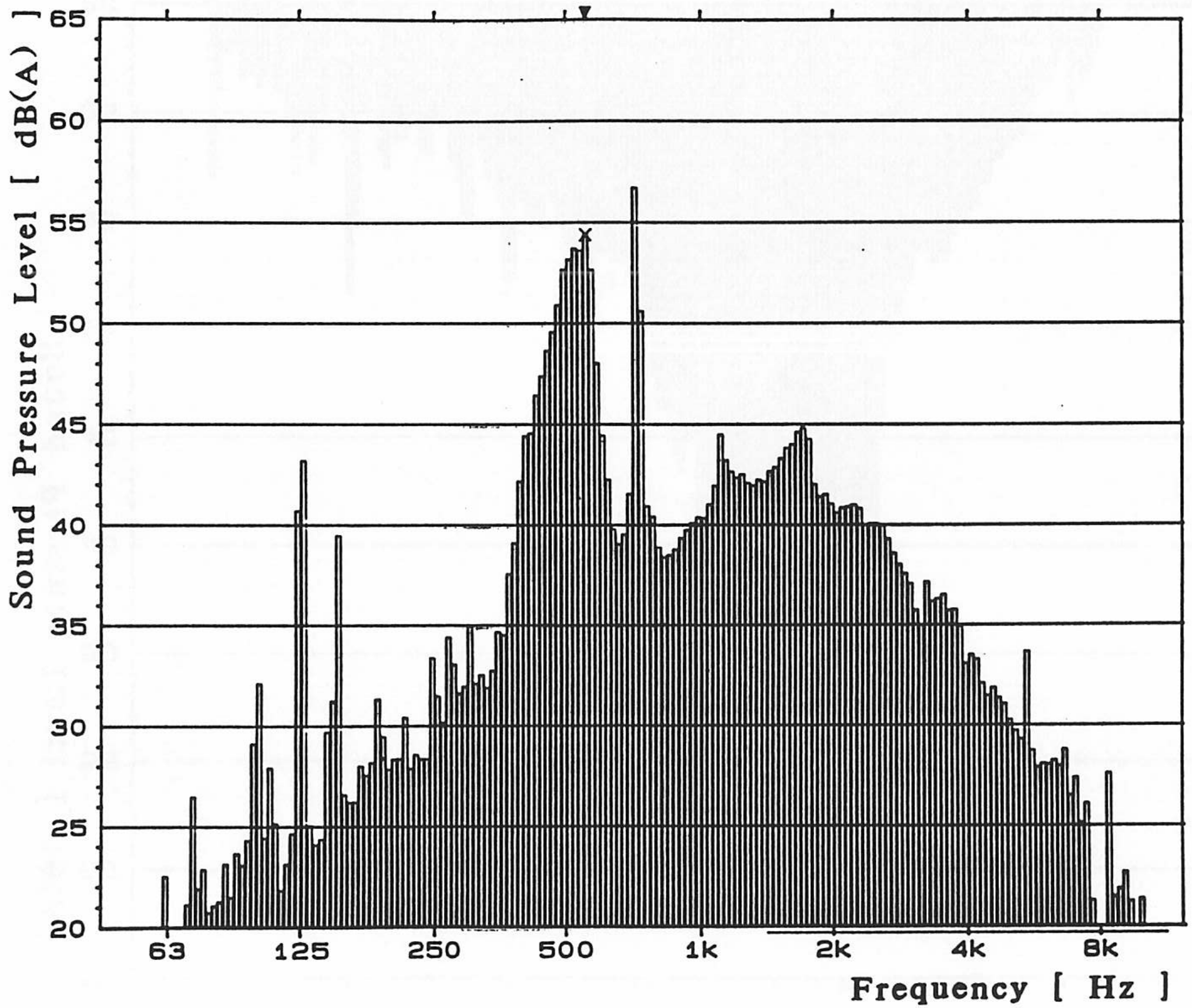


Fig. 6. Measured sound spectrum showing noise due to the blunt trailing edge (with spoiler)

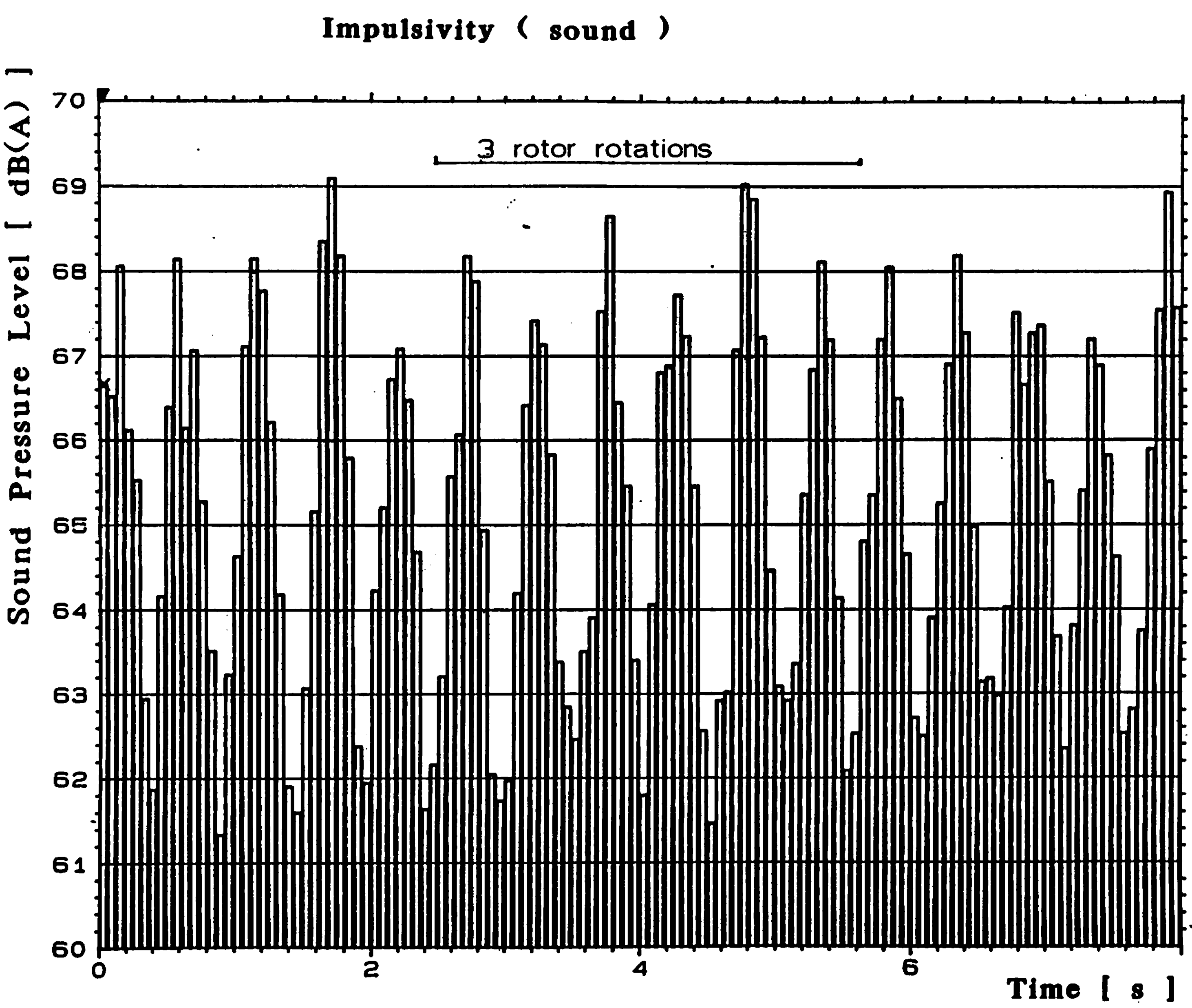


Fig. 7. Measured total sound level as a function of time (only aerodynamic noise).

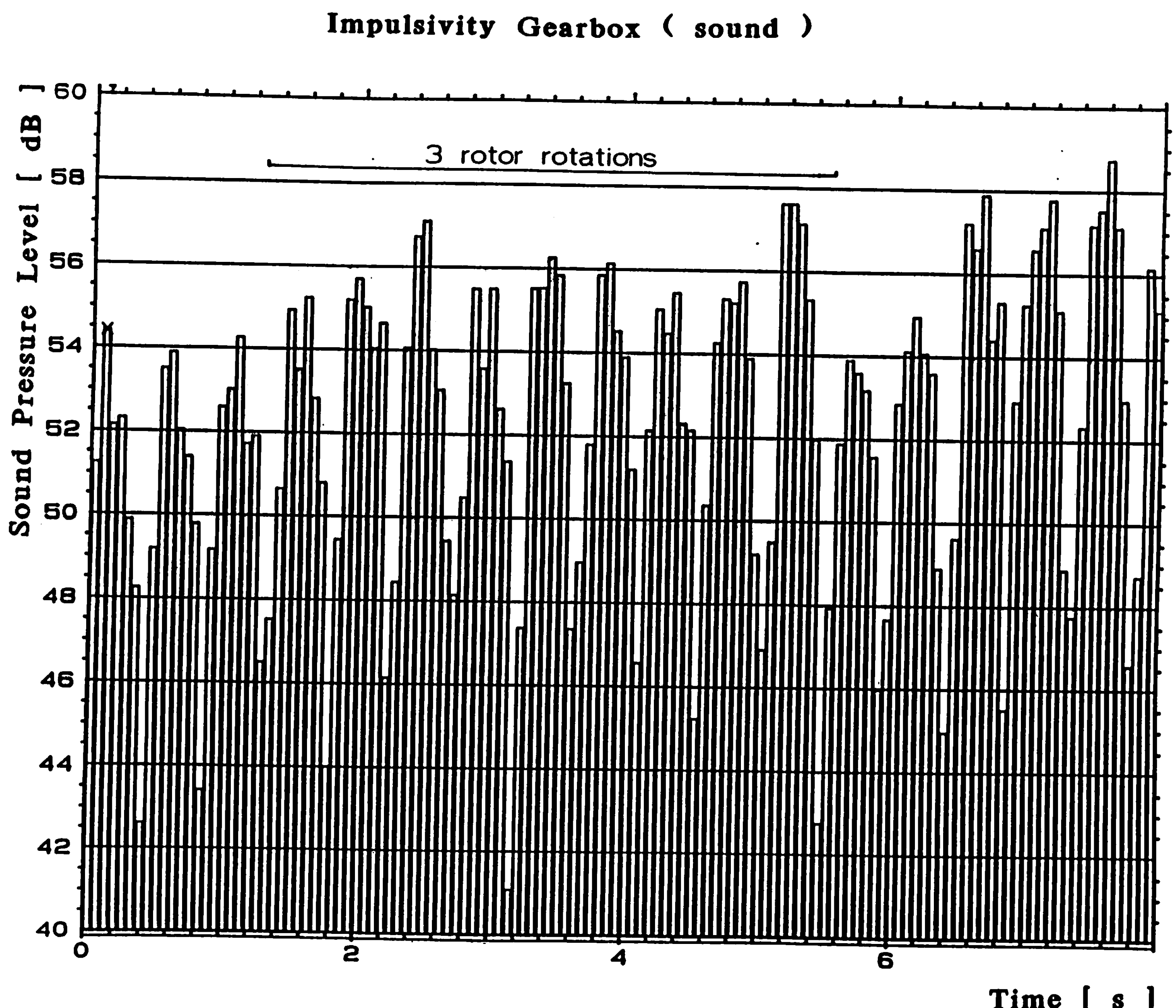


Fig. 8. Sound level at a tooth engage frequency as a function of time (measured up-wind).

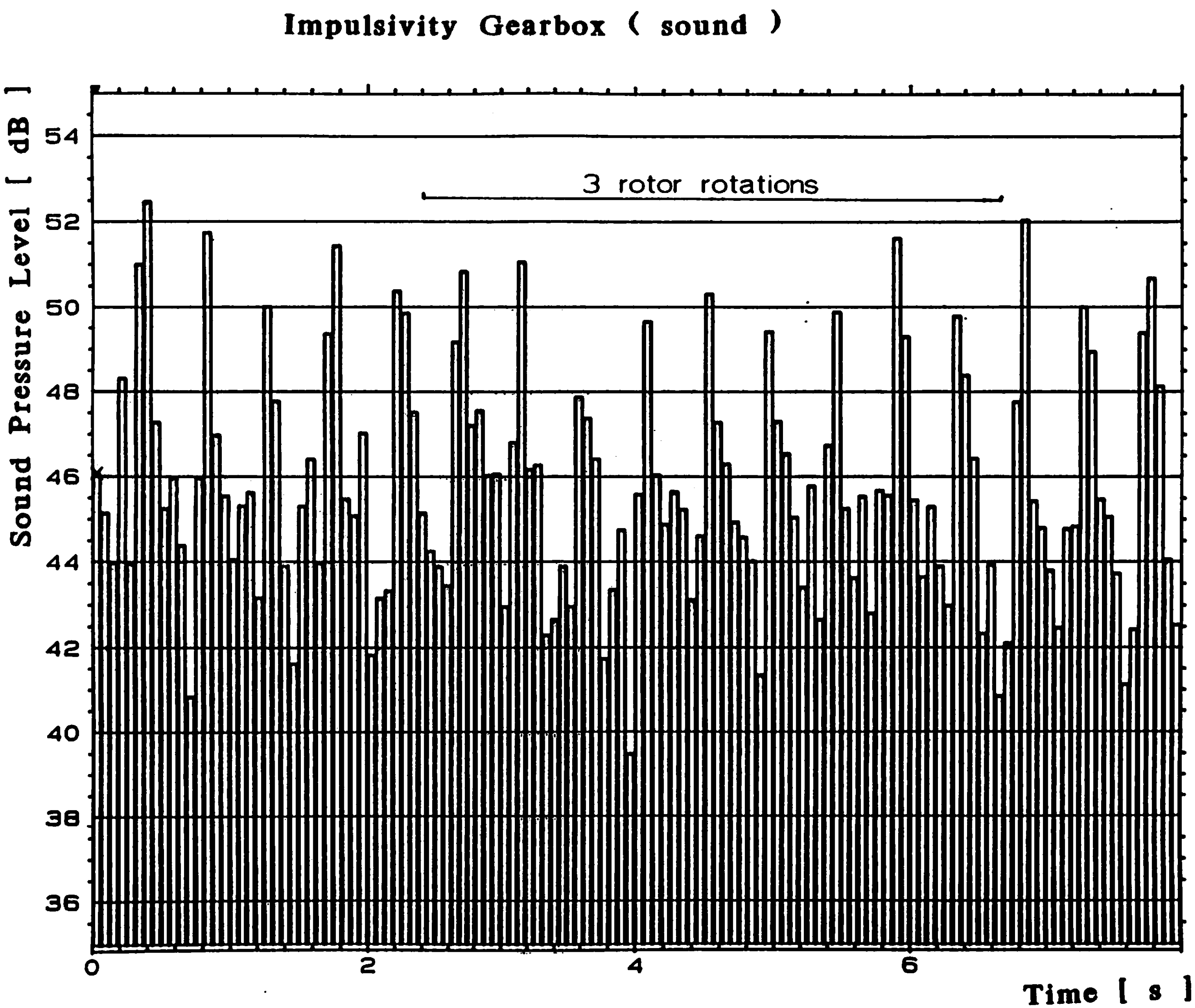


Fig. 9. Sound level at a tooth engage frequency as a function of time (measured in the rotor plane).

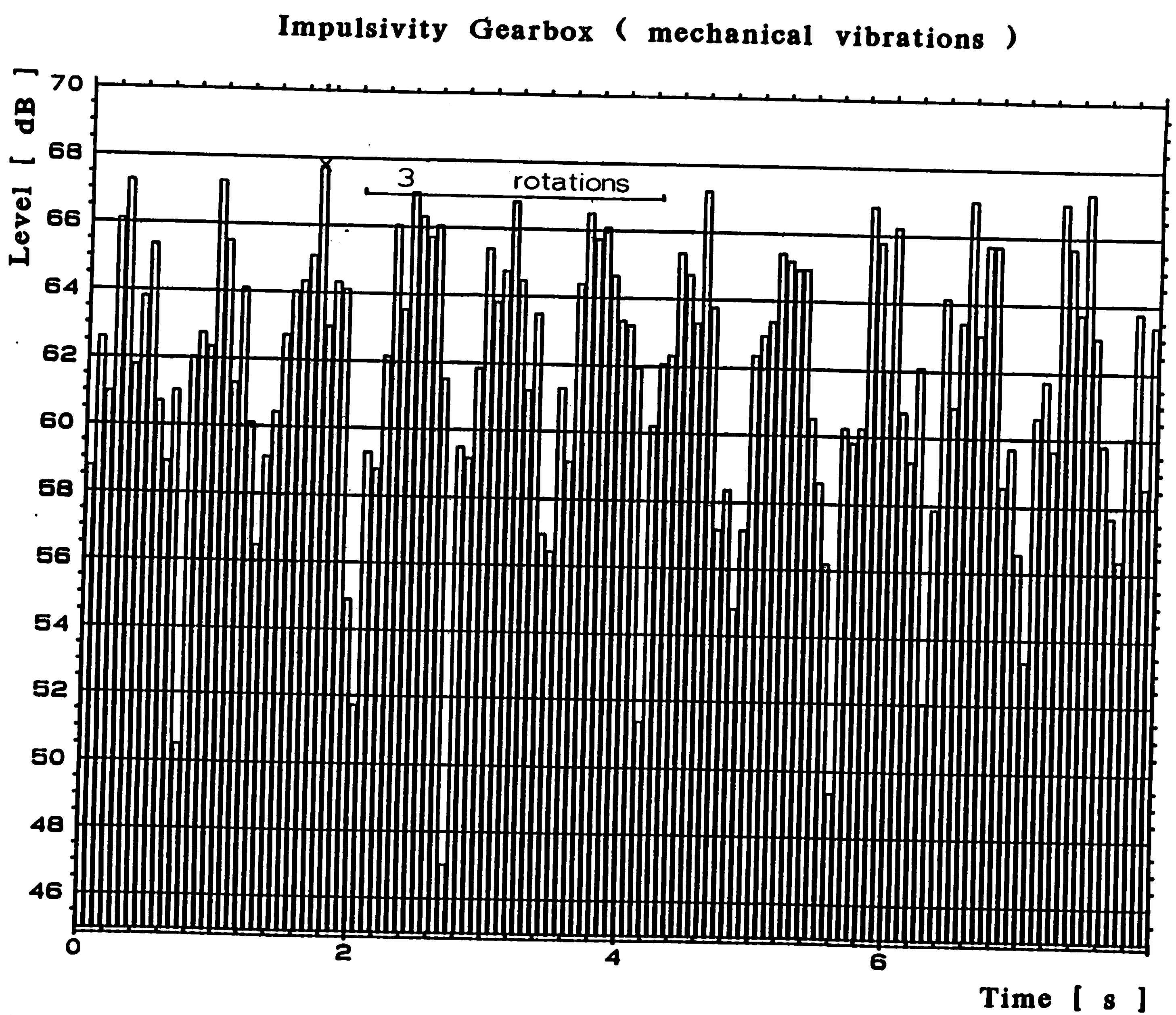


Fig. 10. Intensity of the velocity at a tooth engage frequency measured at the surface of a motor driven gear box (in arbitrary units).

A COMPARISON OF MEASURED AND PREDICTED BROADBAND NOISE FROM THE W.E.G. 20 M WIND TURBINE

A. Glendinning

SUMMARY

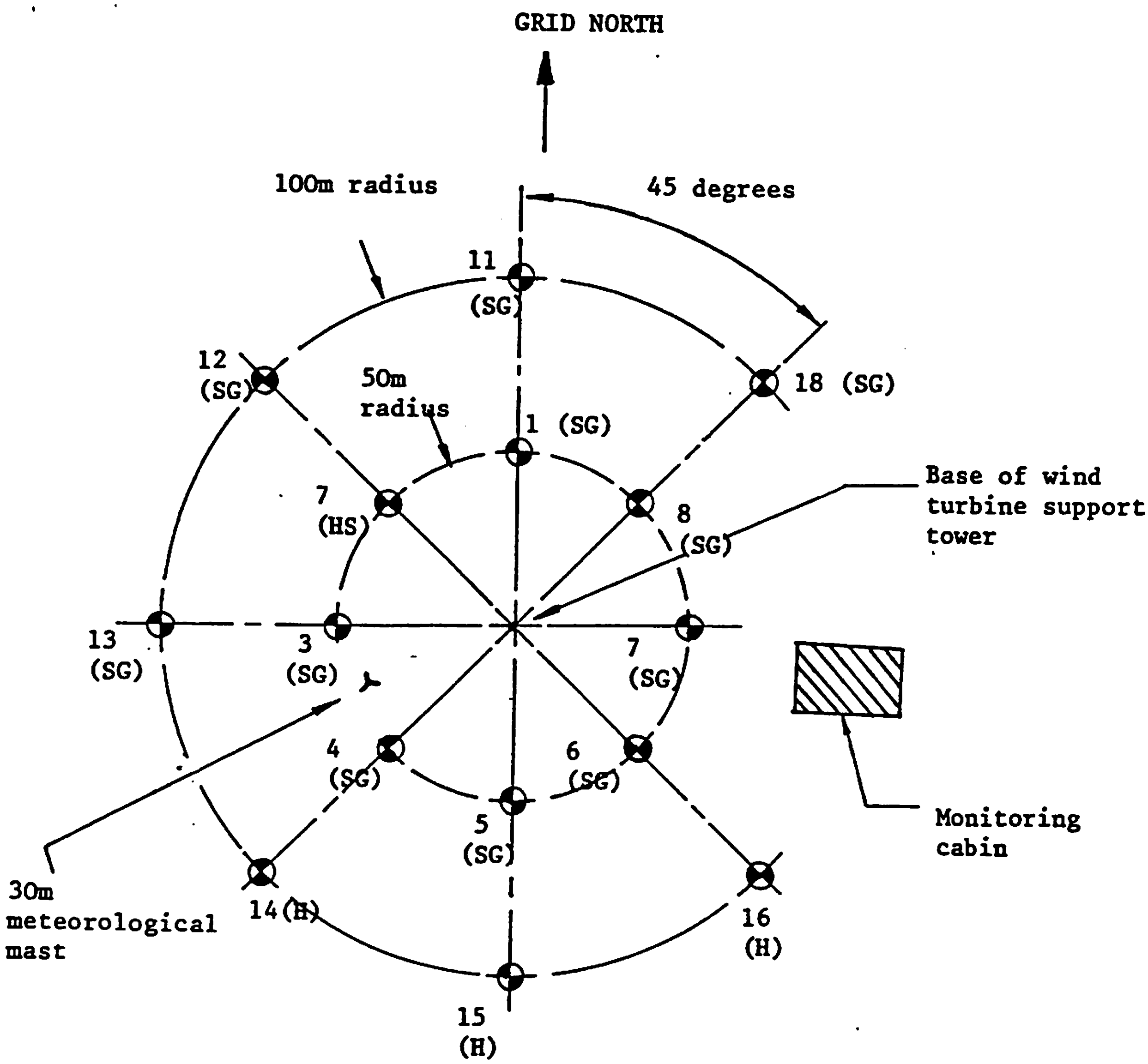
THE RESULTS OF A NOISE SURVEY ON THE W.E.G. 20 M WIND TURBINE LOCATED ON BURGAR HILL, ORKNEY ARE PRESENTED. THEY ARE COMPARED WITH THE OUTPUT OF A BROADBAND NOISE PREDICTION SCHEME, AND A DISCUSSION IS INCLUDED OF THE SOURCE MECHANISMS THOUGHT TO BE DOMINANT ON THIS TYPE OF ROTOR.

APPROACH

1. NOISE SURVEY
   - MEASURING TECHNIQUES
   - RESULTS

2. PRODUCTION OF BROADBAND NOISE
   - SOURCE MECHANISMS
   - EFFECTS OF SOURCE - RECEIVER GEOMETRY

3. COMPARISON OF MEASURED NOISE LEVELS WITH PREDICTION

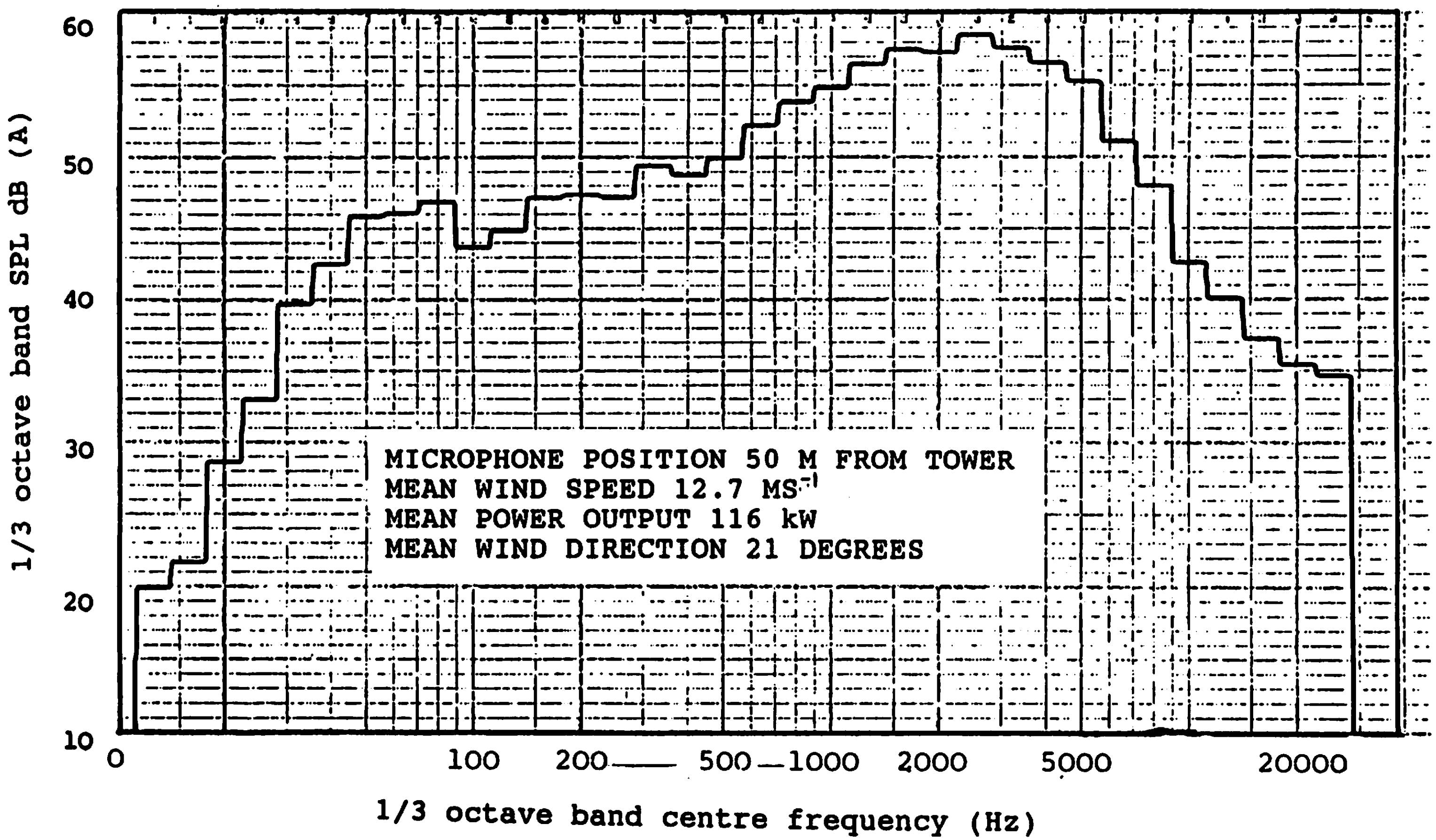


Bracketed symbols indicate ground cover: HS = hard shale
SG = short grass
H = heather

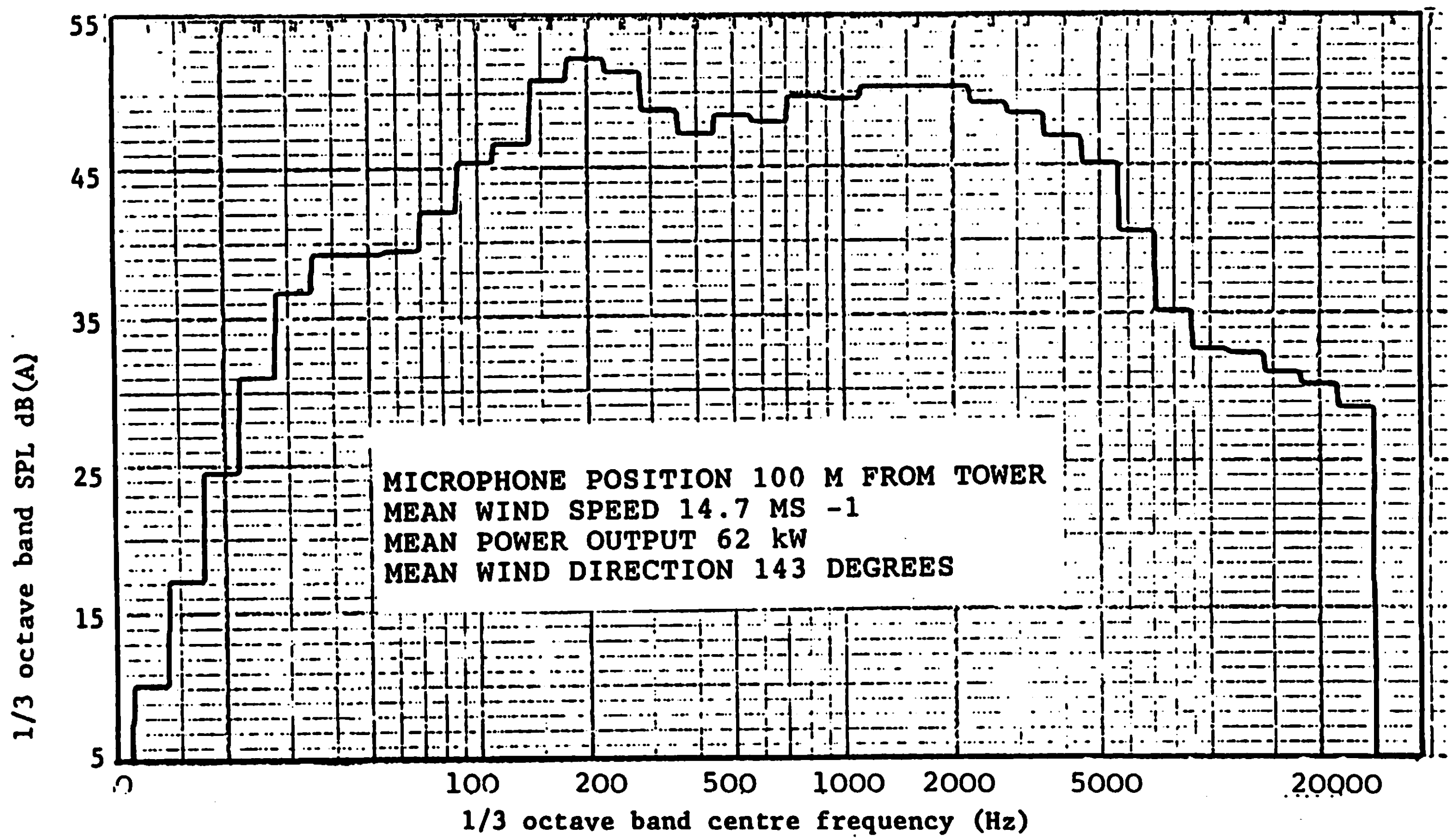
LOCATION OF MEASURING POSITIONS

DATA REQUIRED

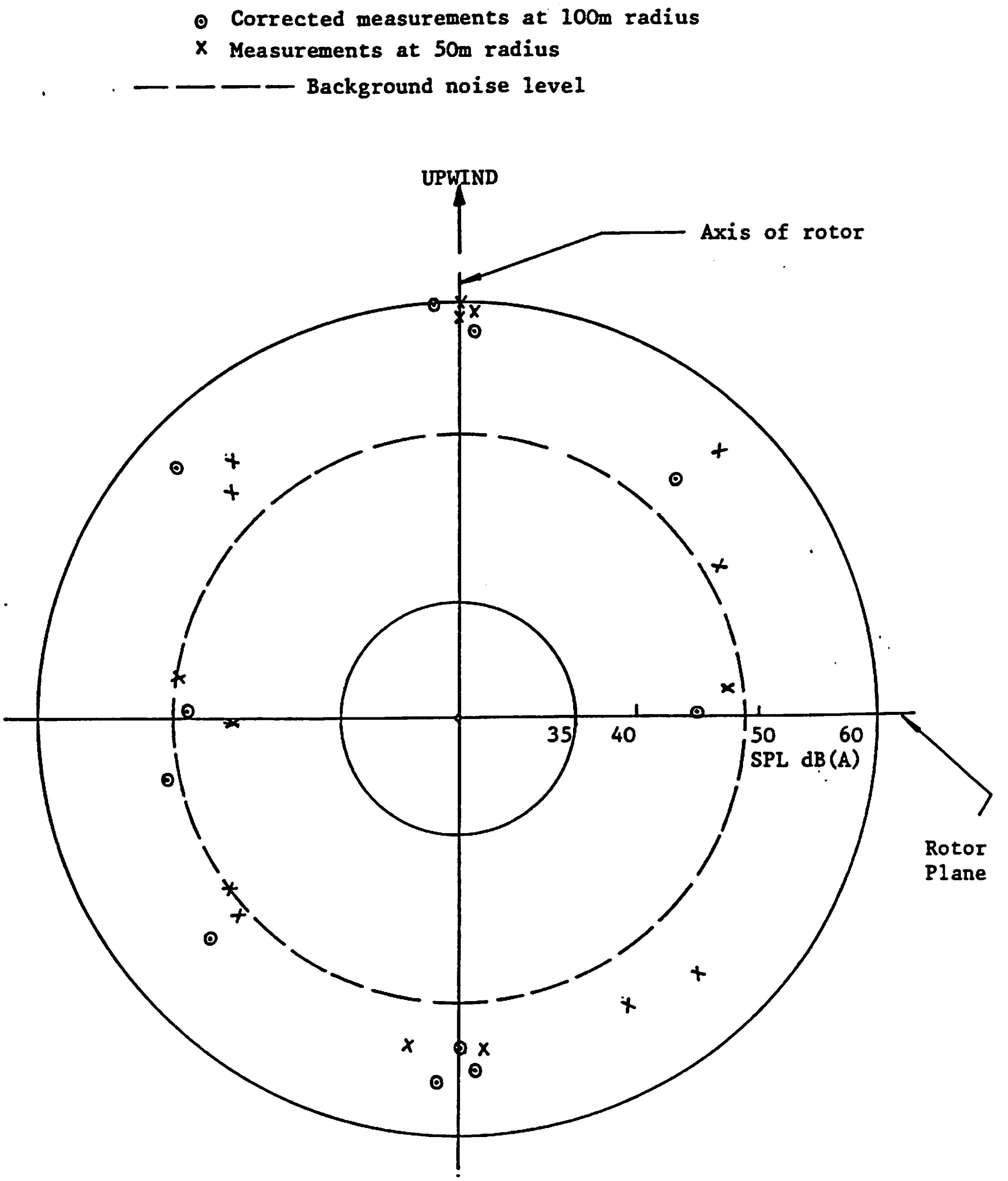
1. "A" WEIGHTED SOUND PRESSURE LEVEL.
2. ROTOR POSITION.
3. NACELLE AZIMUTH ANGLE.
4. BLADE TIP ANGLE.
5. POWER OUTPUT (kW).
6. TEETER ANGLE.
7. WIND SPEED.
8. WIND DIRECTION.



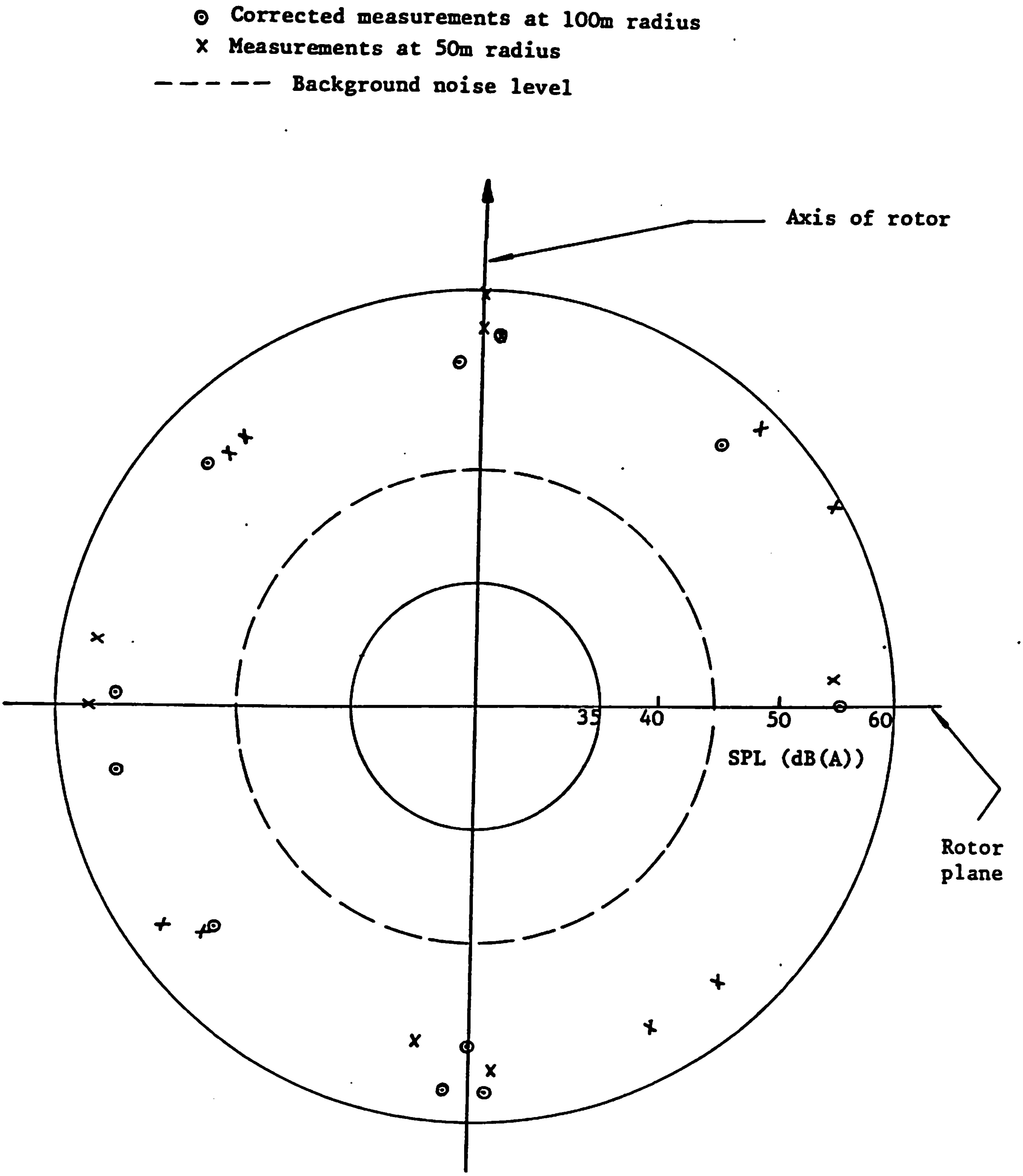
Power Spectrum of Noise Emission in Plane of Rotor



POWER SPECTRUM OF NOISE EMISSION ON ROTOR AXIS, UPWIND



DIRECTIVITY PLOT OF 200 Hz 1/3 OCTAVE BAND SOUND PRESSURE LEVELS DB (A)



DIRECTIVITY PLOT OF 2 KHZ 1/3 OCTAVE BAND SOUND PRESSURE LEVELS DB (A)

SOURCE MECHANISMS

1. Noise from Inflow Turbulence.
   - unsteady lift
   - unsteady thickness

2. Noise from Trailing edge.

3. Noise from Bluff Bodies.

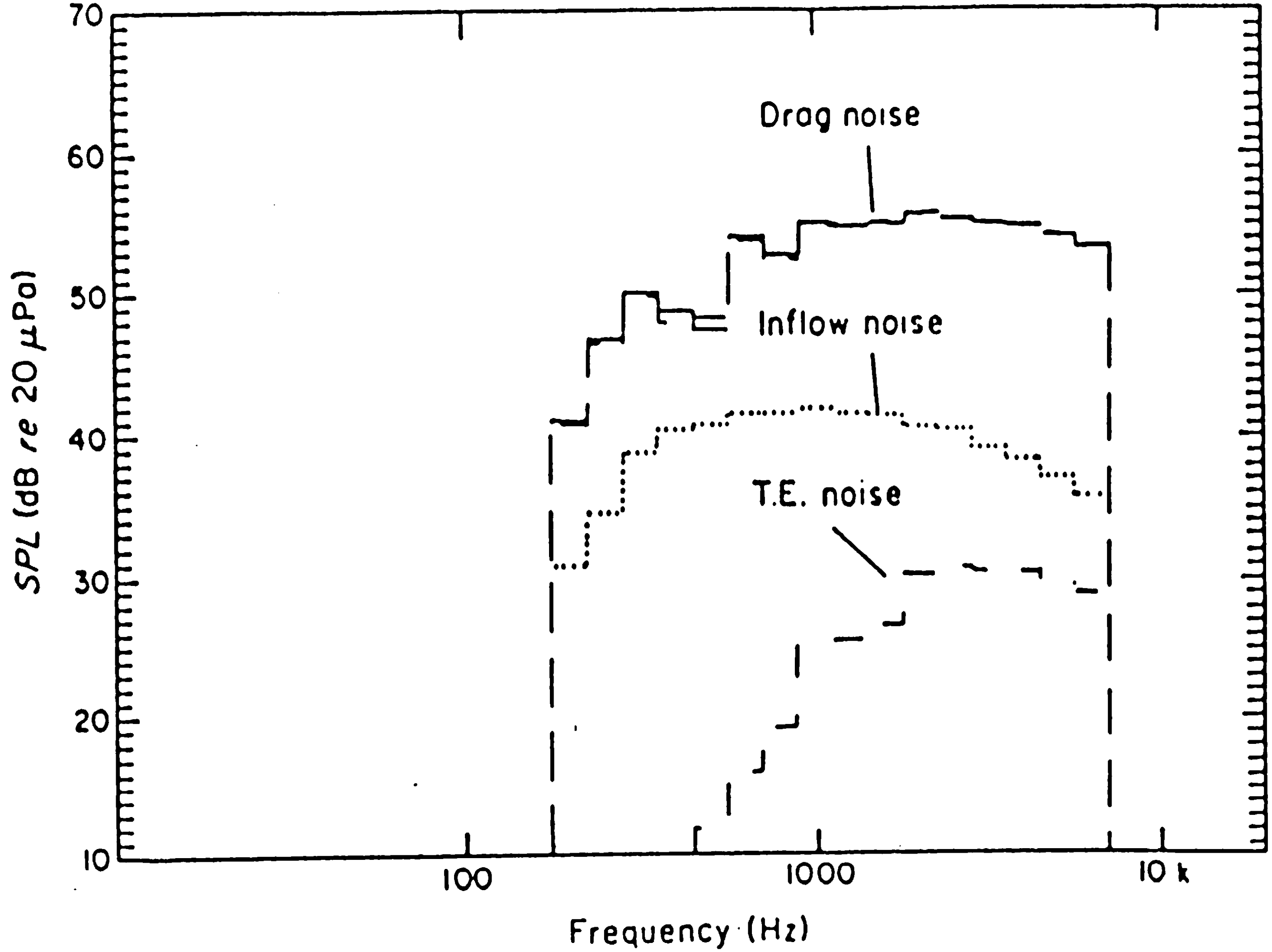
EFFECTS OF SOURCE - RECEIVER GEOMETRY

1. Blade Rotation.

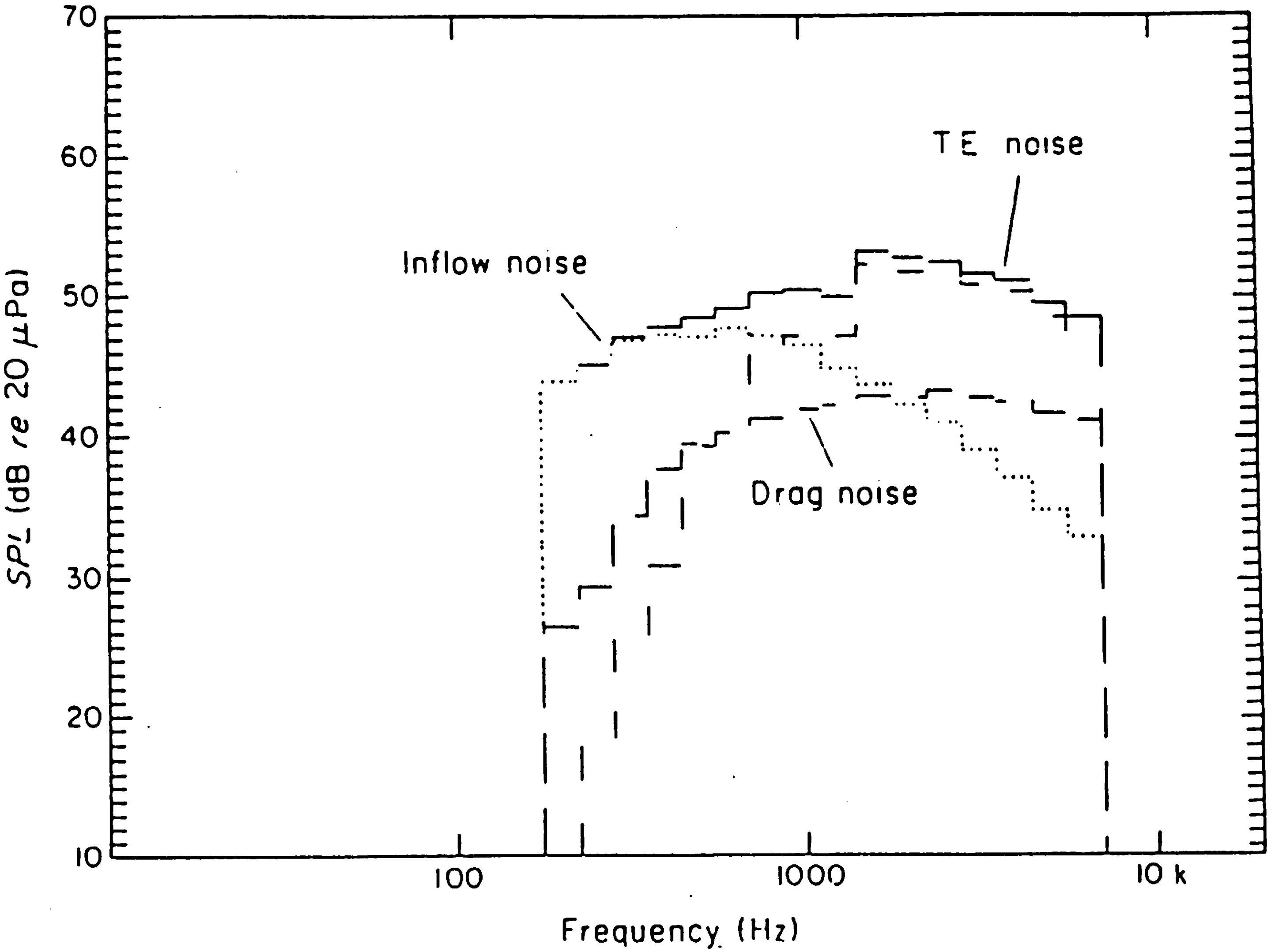
2. Support Tower Scattering.

3. Atmospheric Absorption.

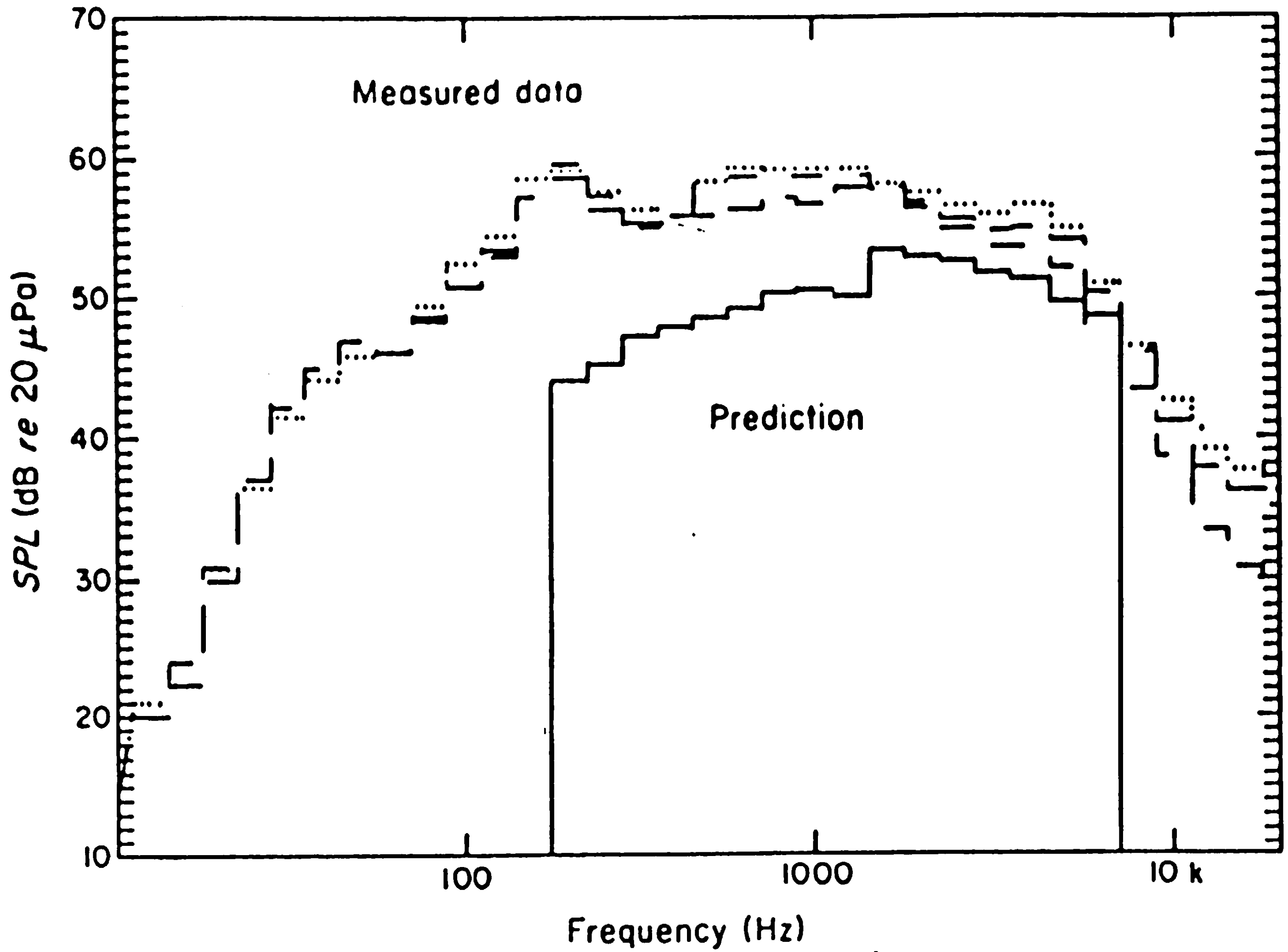
4. Ground Reflections.



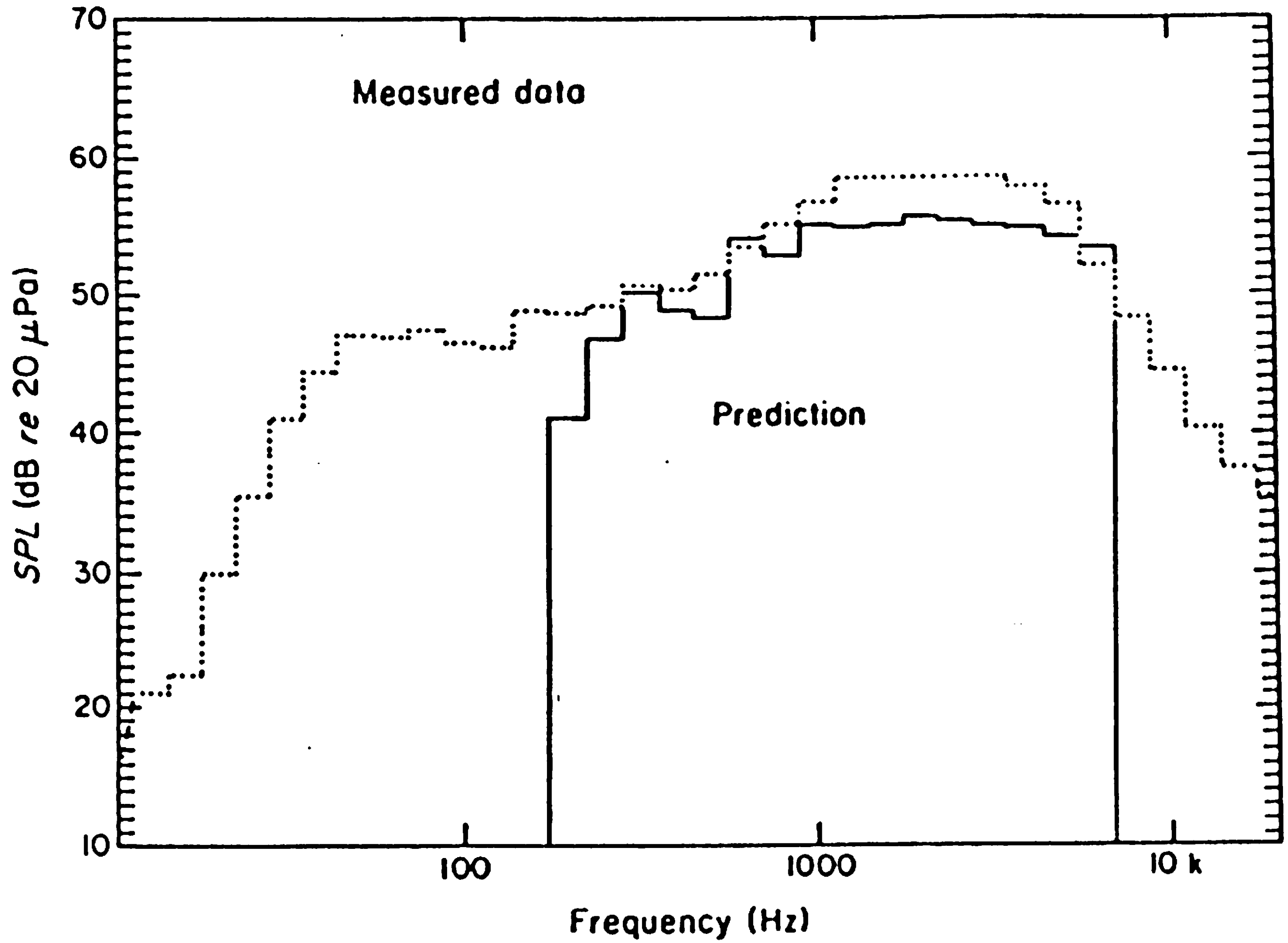
COMPARISON OF PREDICTED SOURCE LEVELS FOR W.E.G. 20 M WIND TURBINE AT 90° TO THE ROTOR, WITH WIND BASED INFLOW TURBULENCE, ONE-THIRD OCTAVE, A-WEIGHTED



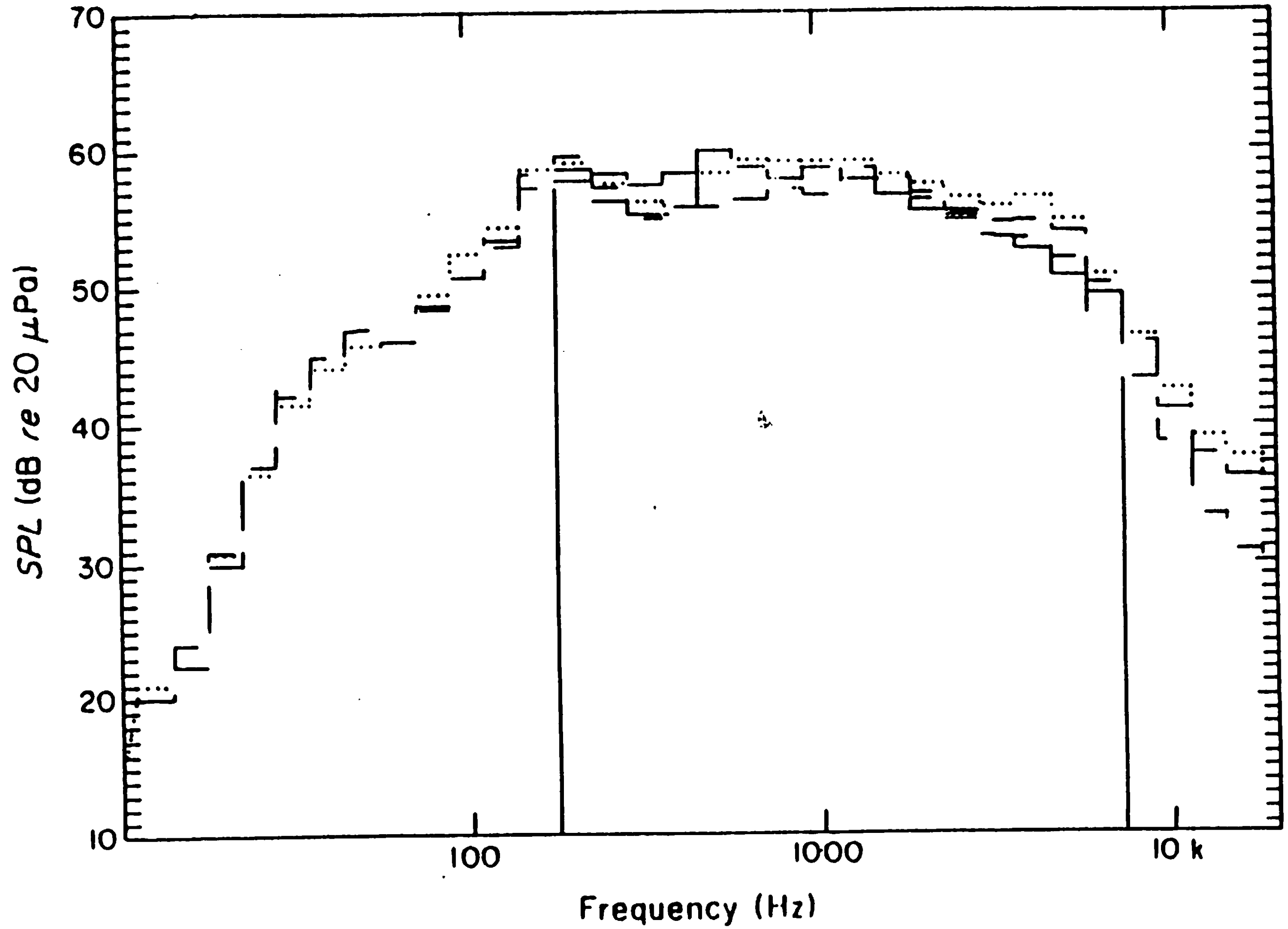
COMPARISON OF PREDICTED SOURCE LEVELS FOR W.E.G. 20 M WIND TURBINE UPWIND OF THE ROTOR, WITH WIND BASED INFLOW TURBULENCE, ONE-THIRD OCTAVE, A-WEIGHTED



. Comparison of prediction with measured data for W.E.G. 20 m wind turbine upwind of the rotor, with wind based inflow turbulence. One-third Octave, A-weighted.



Comparison of prediction with measured data for W.E.G. 20 m wind turbine at 90° to the rotor, with wind based inflow turbulence. One-third Octave, A-weighted.



COMPARISON OF PREDICTED SOURCE LEVELS FOR W.E.G. 20 M WIND TURBINE UPWIND OF THE ROTOR, WITH BLADE BASED INFLOW TURBULENCE, ONE-THIRD OCTAVE, A-WEIGHTED

AERODYNAMIC NOISE REDUCED DESIGN OF LARGE ADVANCED WIND TURBINES

F. Hagg

Introduction/summary

Noise production must be distinguished both in the conceptual stage and design stage of a wind turbine product. This is specifically true for aerodynamic noise. Once a wind turbine is produced, no insulation nor modification can be applied without expensive design modifications.

Stork Product Engineering is working on the design technology of cost effective wind turbines in co-operation with the Netherlands Energy Foundation, ECN in commission of the Netherlands Agency for Energy and the Environment, NOVEM and the EEC.

These technology studies are meant for large wind turbines with high tip speed ratios, flexible structures, passively controlled tip pitch and soft power variable speed machines. One of the studies concerns the noise production of such a machine, taking into account the noise problem of large machines with high tip speed ratios, the slit of the pitch control mechanism in the tip of the blades and the rotor speed excursions of the soft variable speed machine.

One of the features of variable speed machines is the control of the tip speed, which is also the main parameter in aerodynamic noise producing mechanisms. This feature is meant for production improvement and for rotor torque reduction, but can be used otherwise for the control of noise in delicate periods, such as at night. This so called night control appears to be an important tool for noise reduction at minor costs. In the study the noise influence of this parameter and others are empirically derived with the RHOAK model of NLR 1] and some measurements at the ECN test field.

With this empiric relation and the cost optimization model OPTIHAT of SPE, conceptual and parameter sensitivity studies are performed to achieve at a cost effective and noise reduced design. While studying the concept for a noise reduced design a new concept was found, based on a shift of the constant λ operation to lower wind speeds. A surprising result was not only a reduction of

- the determined machine is a large cost effective machine with flexible structures, passive tip pitch control and a soft power variable speed machine:
  - below v_{rated}: constant-λ operation, with speed excursions up to 30% for energy improvement
  - above v_{rated}: passive tip control with speed excursions up to 30% for soft power control

  The cost of noise reduction measures will be determined by the cost optimization model OPTIHAT of SPE.
- the wind turbine will be located at the NEWECS 45 site in Medemblik, a representative location for wind turbines in The Netherlands.

New noise reduced design concept

Relations (1) and (2) require the following physical and technical modifications to achieve at a noise reduced design:

- lower rotor speed, made possible by design rotor speed reduction which results in a lower axial rotor force (if only round V_{rated}), but with a higher rotor torque, and also by a temporary speed reduction at night.
- small blade area, made possible by a slender blade and an increase of design tip speed ratio λ
- small axial force coefficient C_{dax}, made possible by design speed reduction round V_{rel} and power flattening.
- large rotor diameter? with a question mark. Other empirical relations, although stemming from helicopter theory 3], do not support this diameter influence, which is also true for the other parameters and other relations 4] (see table 1)

These modifications tend to a new concept; a shift of constant λ operation to lower wind speeds than v_{rated}. The consequences of this new concept are explained by the diagrams 2a to 2e, in which the original concept in dotted lines is compared with the new concept in solid lines. A blade area reduction is obtained by lowering the blade chord, which leads to a higher design λ. This can be seen in the Ω-v diagram of figure 2a, which also shows the shift of the constant λ operation. This shift leads to a new constant speed control concept between the constant λ operation and V_{rated}, but with a decent reduction in rotor speed. For better understanding a more detailed Ω-v diagram is shown in figure 3.

In figure 2b the C_p-v diagram shows the maximum power coefficient in the constant-λ operations, but also the worsening around V_{rated}, which is moderate.

For this reason the power loss is also moderate, which is represented in the P-v diagram of figure 2c. In this figure the power flattening is also represented with a dot and dash line.

In the E-v diagram of figure 2d the energy production is shown with the losses round v_{rated} and the profit round v_{ct}. The sum of both leads to a small overall loss.

Figure 2e shows the reduction in axial force, produced by the speed reduction (dotted line) round v_{rated} and by the power flattening (dot and dash line). This leads to an extra noise reduction as well as a cost reduction in the turbine construction. Another important cost reduction is realized by the blade area decrease, leading to an axial force reduction during parking conditions, mostly the ultimate load for flexible systems.

The cost reduction in the structure exceeds the cost increase caused by the higher rotor torque (more expensive gearbox) and the small energy loss. Quantitatively the new concept means a decrease of the kWh-price with 8.6% and of the noise with 2 dBa at V_{rel} and 5 dBa at v_{rated}, with the parameters only optimised to the kWh-price.

This result was surprising, as a design concept improvement for noise reasons leads also to an improvement in costs.
This result was not found earlier, because the new concept is a sophistication on the global concept of rotor torque impact improvement by the soft power system, which is only globally evaluated yet. The latter reduces the influence of the gearbox on the kWh-price and the need for high rotor speeds (small rotor torque) around v_{rated}. The axial force is now dominating in the costs of this wind turbine. Thus there is a need for a smaller blade area (high tip speed ratio λ) and a speed reduction to improve the axial force around Vrated, which is made possible by the constant λ operation shift.

Sensitivity study
The effect of the different design parameters on the noise of the machine and the kWh-price is analyzed by a sensitivity study. In this study first the cost optimum values of the important parameters are determined as a base point of the study on the new turbine concept. The noise immission of the base point of the concept is at a distance of 2.5 rotor diameters 58 dBa, which is 18 dBa too much.

From this base point the value of the determined parameters is changed 10% in the direction of noise reduction, not taking in account the eventual relations between the parameters. The result of this study is compiled in table 2 below. The first column gives the parameter and the second the figure number of the graphical reproduction of the analysis. The next columns show the parameter value, the modification for noise improvement, the kWh-price increase, the noise improvement and the effect. The effect is defined as the ratio of the noise improvement and the kWh-price increase, normalized with the effect of the best parameter. For comparison reasons only, also the new design concept with respect to the original design (not included in this sensitivity study) is shown.

As can be noted, the temporary speed reduction at night only causes little loss of energy. On the other hand with the reduction of design speed the machine design must be changed, which will imply extra costs (gearbox, modification of eigenfrequencies and aerodynamic shape) and more energy loss.

The effects of the other parameters are small, as they are already used for the new design concept. In particular the tower height is not important anymore, since V_{rel} lies in the constant speed operation of the machine.

Cost effective noise reduced design

Using the results of the sensitivity study the cost effective noise reduced design is analyzed, now taking into account the relations between the different parameters. It is cost effective in the sense that the kWh-price will increase only by a few percents. This analysis results in a maximum possible (till Ω_{min}) night speed reduction of 30%, a design speed reduction from 29 to 25 rpm, a larger rotor diameter from 60 to 70 m and a higher tower from 60 to 70 m.
In the study for the determination of the noise immission a relative distance of 2.5 times the rotor diameter is used. At this distance the noise is 44 dBa at night, being 4 dBa too much.

To get more noise improvement another parameter is needed; the distance of the observer to the wind turbine. In figure 10a the influence of this distance is represented at different values of the rotor speed. In figure 10b the same is done for different values of speed reduction at night in case of a design speed of 25 rpm. In the latter the night limit of 40 dBa is reached at a maximum night speed reduction of 30% and a distance of 4 D.
The cost consequences are presented in figure 11 as function of the main noise reducing parameters; night speed reduction and design speed reduction. In the function area, dotted lines are plotted, which indicate the equidistance to the turbine with a noise immission level of 40 dBa at night. As noted earlier the night speed reduction is most cost effective in noise reduction, which can be

scope of the present study.

However an analysis of the environmental noise is an important topic for the judgement of noise reducing measures in a design analysis.
We highly recommend further study on environment noise with regard to wind turbine noise.

Literature

1] W.B. de Wolf, Parameter study on the aerodynamic noise of a 35m diameter Flexhat rotor (Dutch), NLR, AV-88-011 L, 25-10-88, (see also the NLR presentation of H.M.M. van der Wal on the IEA expert meeting :Noise generating mechanisms of wind turbines, 27 November 1989, Petten)

2] W.J. Stam, Wind turbines in test field: noise measurement (Dutch), ECN 83-136. (see also presentation of ECN on the IEA expert meeting)

3] W. Johnson,Helicopter Theory, Princeton University Press, New Jersey, 1980, ISBN 0-691-07917-4

4] A.E. Pfeiffer, "Brochure Windturbinegeluid" (Dutch). HW 88.1704.1, 1988 (see also presentation of Holland Windturbine on the IEA expert meeting)

$$L_{aeq} = C_1 \log(\lambda\ V_{rel}) + C_2 \log(N\ A_b/A_r) + C_3 \log\ (C_{da}) + C_4 \log\ (D/a) -$$
$$- C_5 \log\ (D) - C_6$$

Constant	RHOAK 1]	Brochure 4]	Johnson 3]
C_1	63.6	50	60
C_2	11.5	-	-10
C_3	2.5	-	20
C_4	20	20	20
C_5	10	10	0
C_6	27.5	15	44-48

table 1: Comparison of the used empirical relation with others

parameter	figure	optimum value	adjusted value for noise	kWh-price increase %	noise decrease dBa	effect % of best
night correction	4	1	0,9	0,7	3	100
desin speed	5	33 opm	30	3,0	3	23,3
blade chord	6	0,1R	0,09	1,6	0,5	7,5
diameter	7	60 m	66	9,2	2,1	5,3
design λ	8	11,8	10,6	2,8	0,5	3,8
hub height	-	51 m	56	0,4	0,06	3,3
power	-	871 kW	785	1,7	0	0
new concept		yes	no	-8,6	2	

table 2: Cost and noise sensitivity study on design parameters

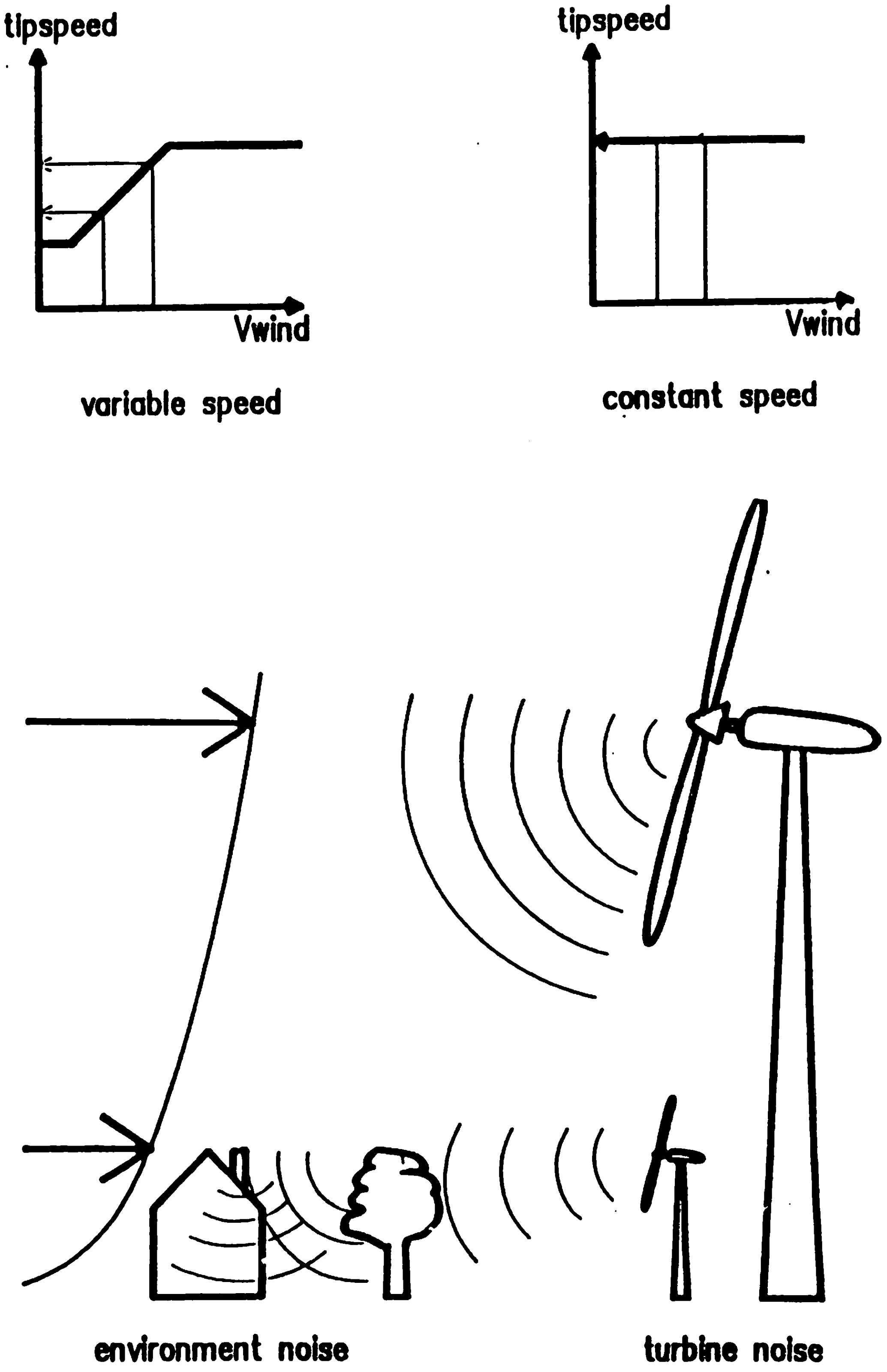


figure 1: Height correction of noise immission relevant windspeed

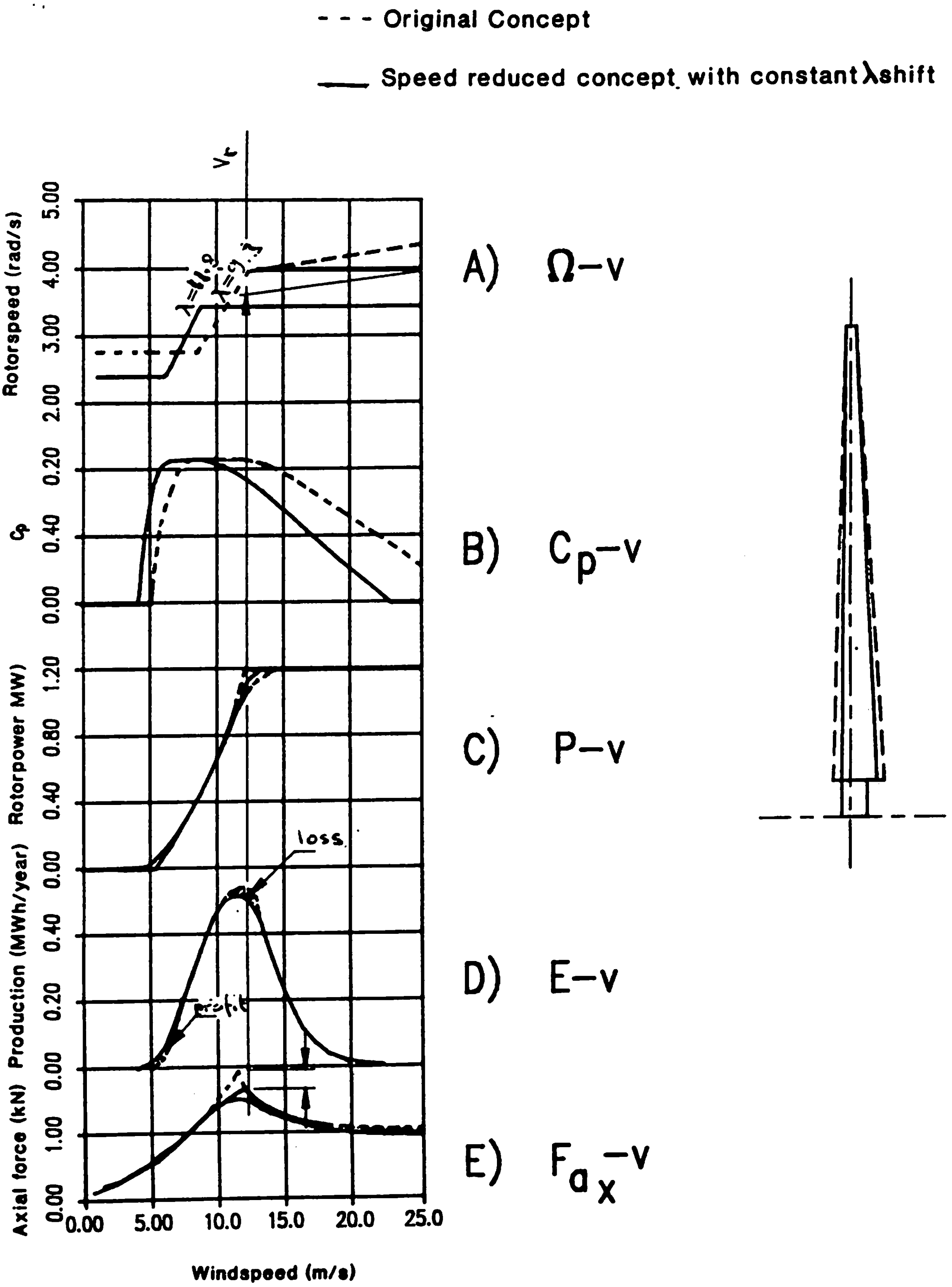


figure 2: Comparison of original and new speed reduced concept

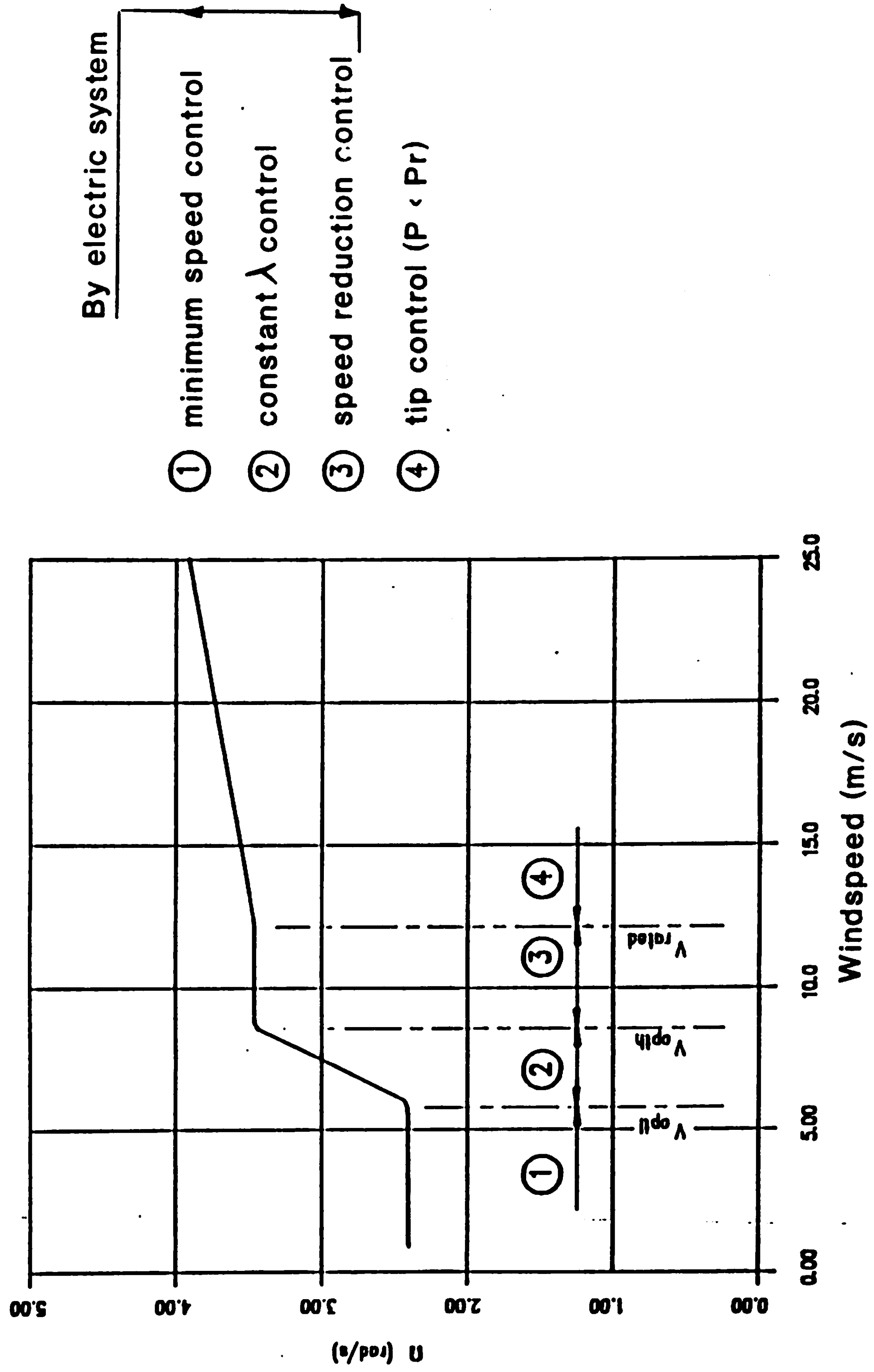
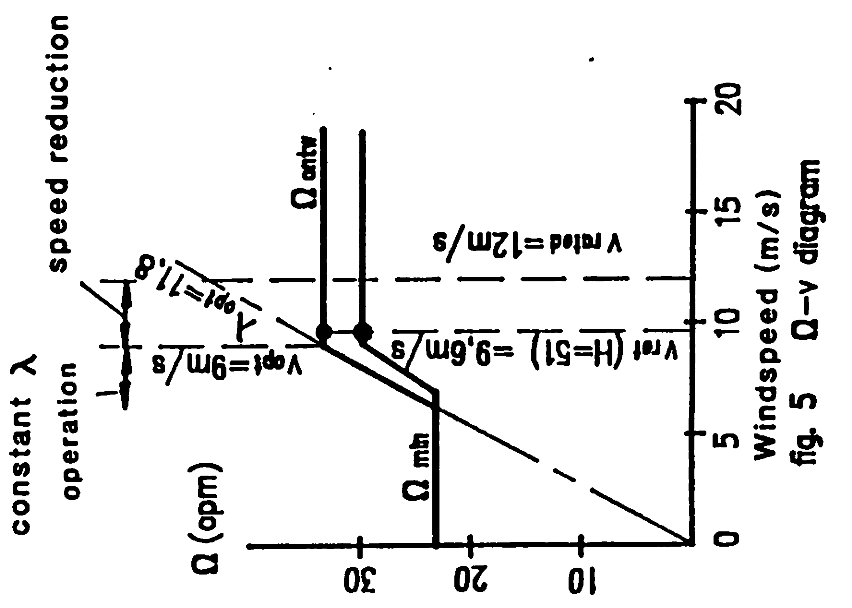


figure 3: Rotor speed-wind speed diagram of speed reduced concept



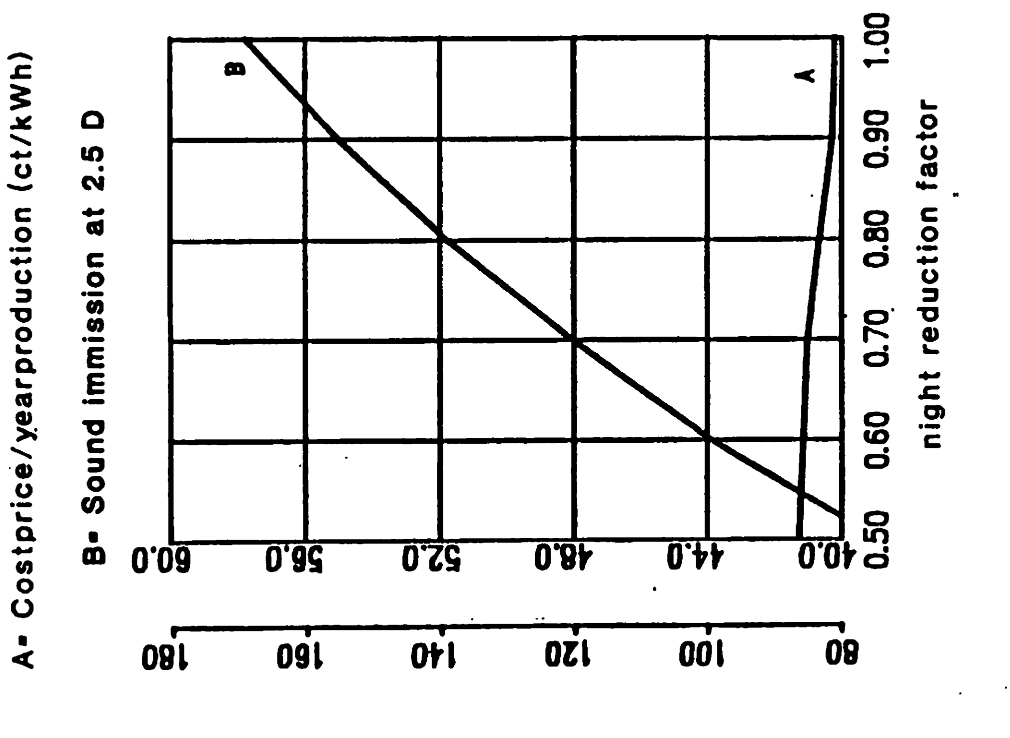
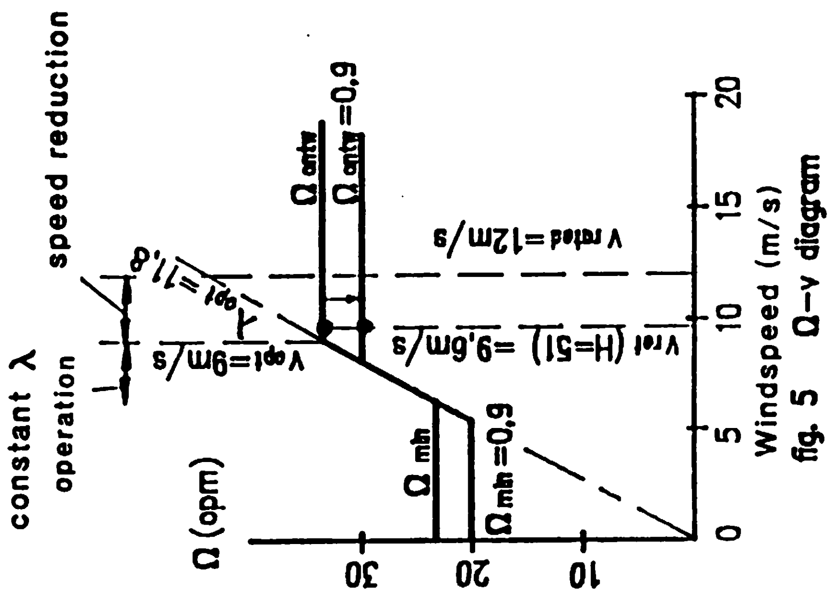


figure 4: Sensitivity study on night speed reduction



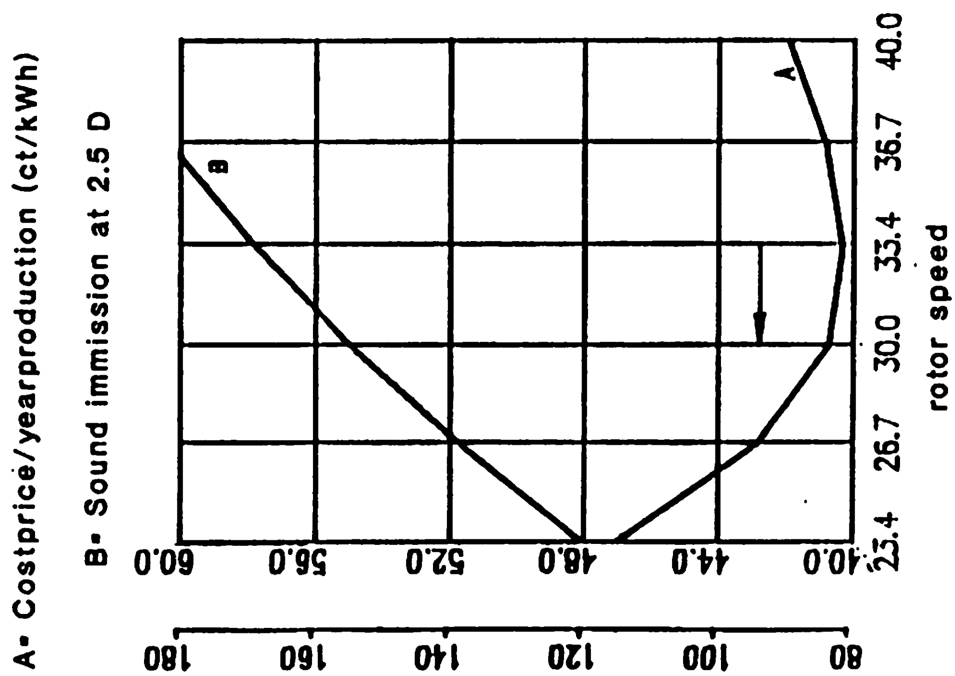


figure 5: Sensitivity study on design rotor speed

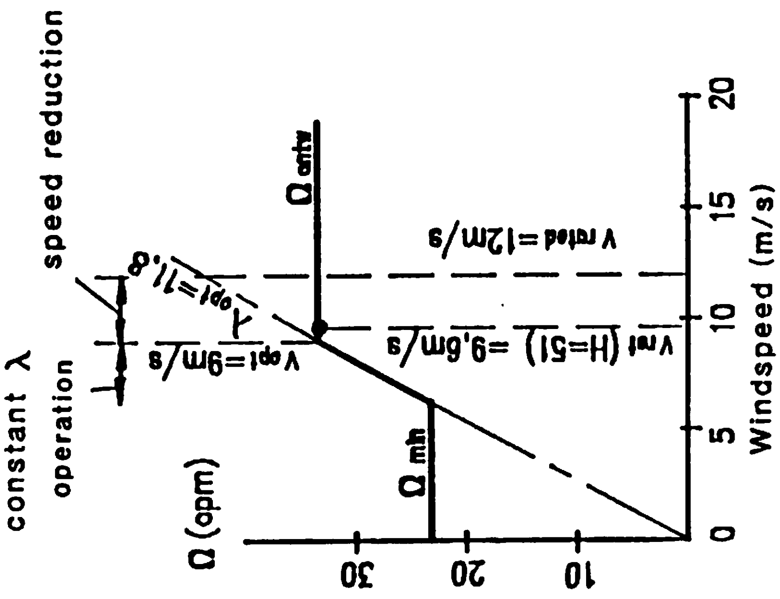


fig. 5 Ω–v diagram

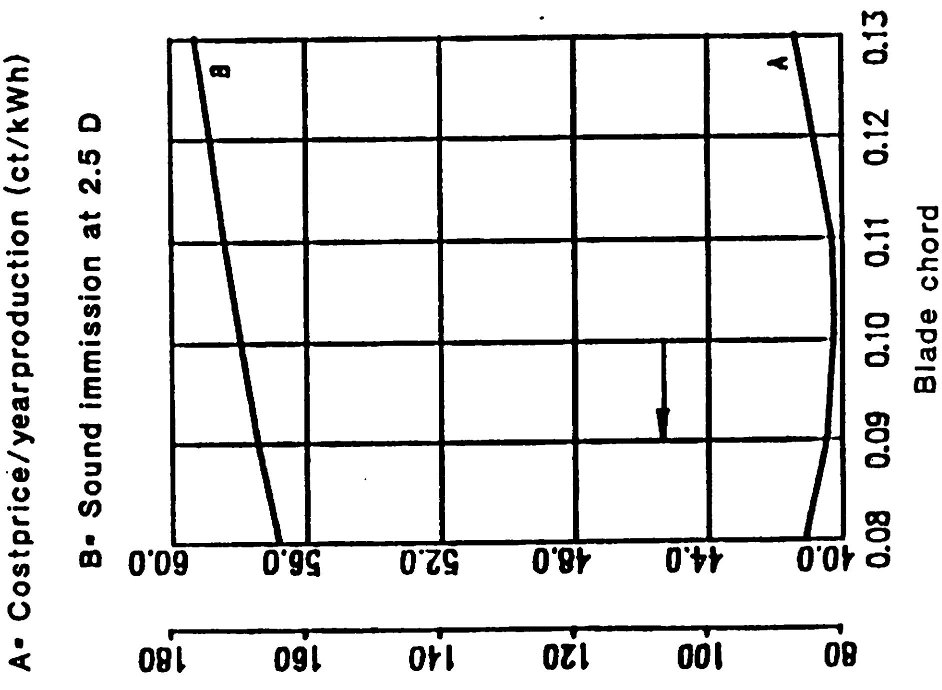
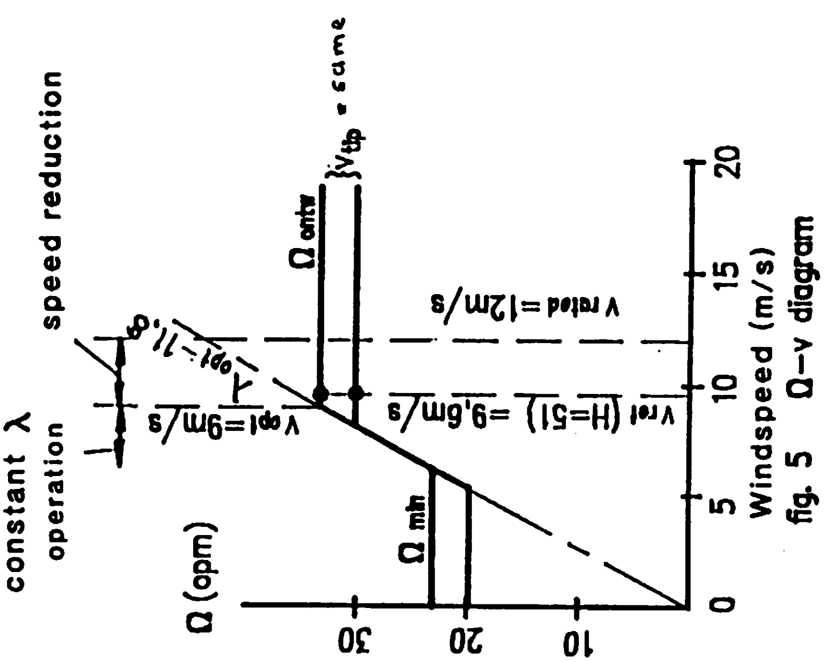


figure 6: Sensitivity study on blade chord



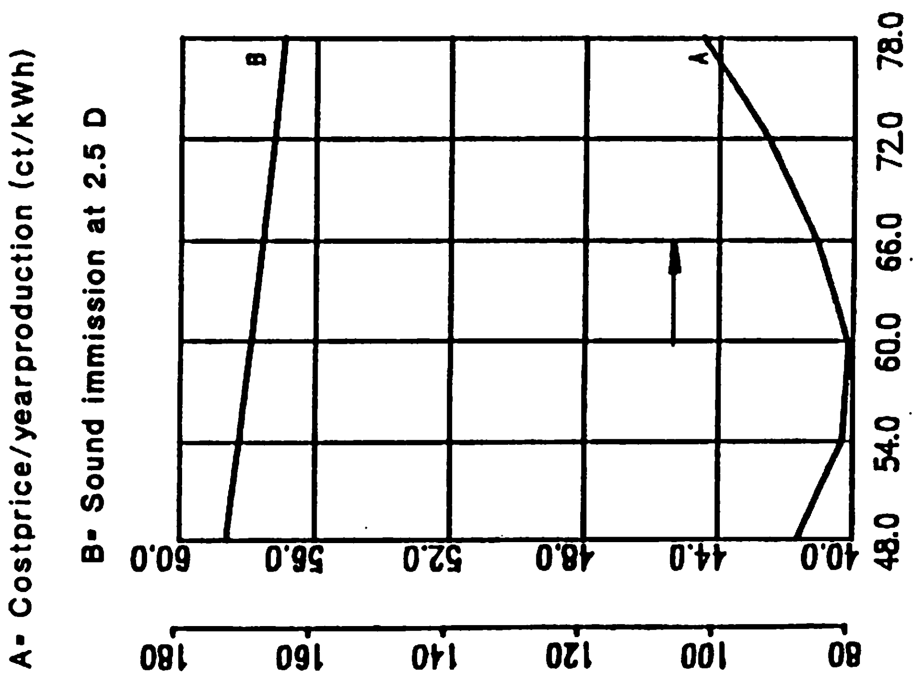
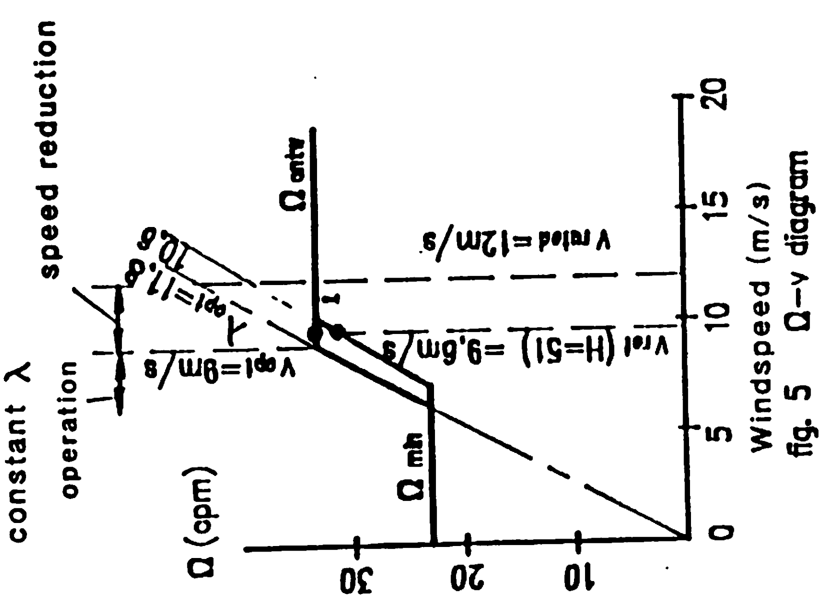


figure 7: sensitivity study on rotor diameter



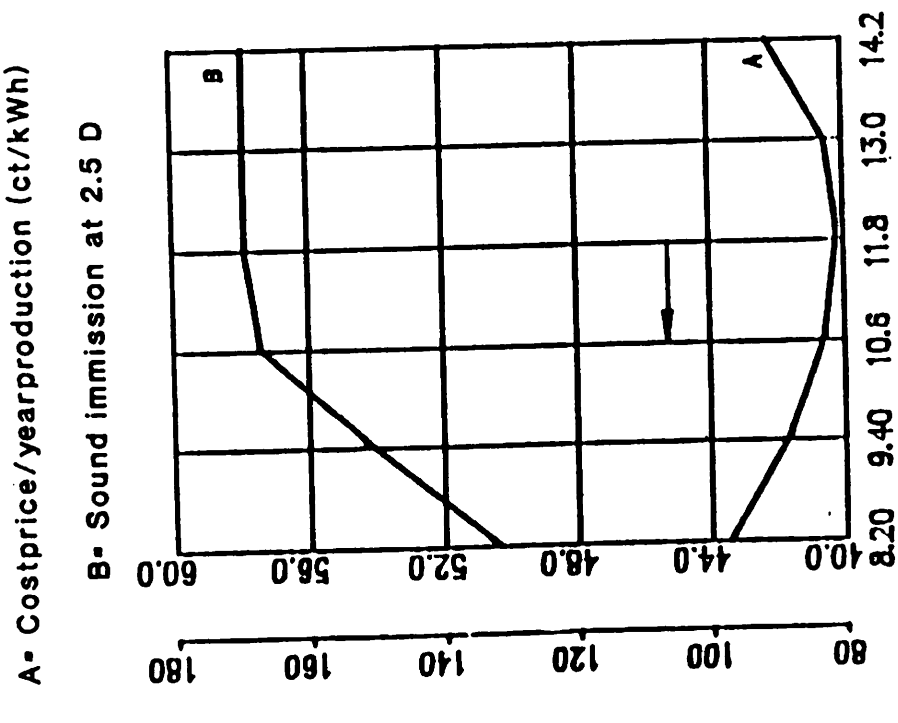


figure 8: Sensitivity study on design tipspeed ratio λ

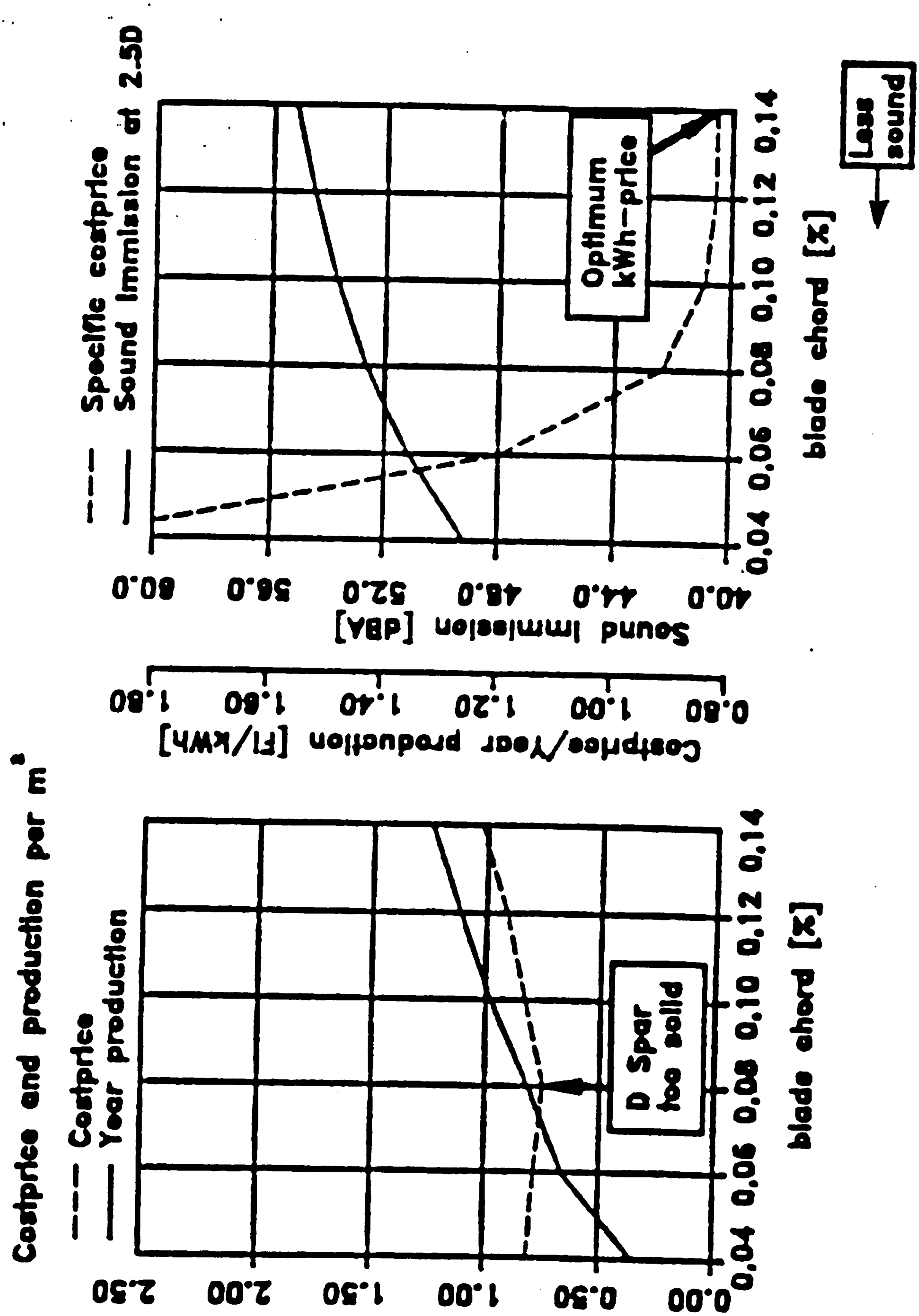


figure 9: Sensitivity study of the blade chord in combination with a increase of the design rotor speed of 14% (25 rpm).

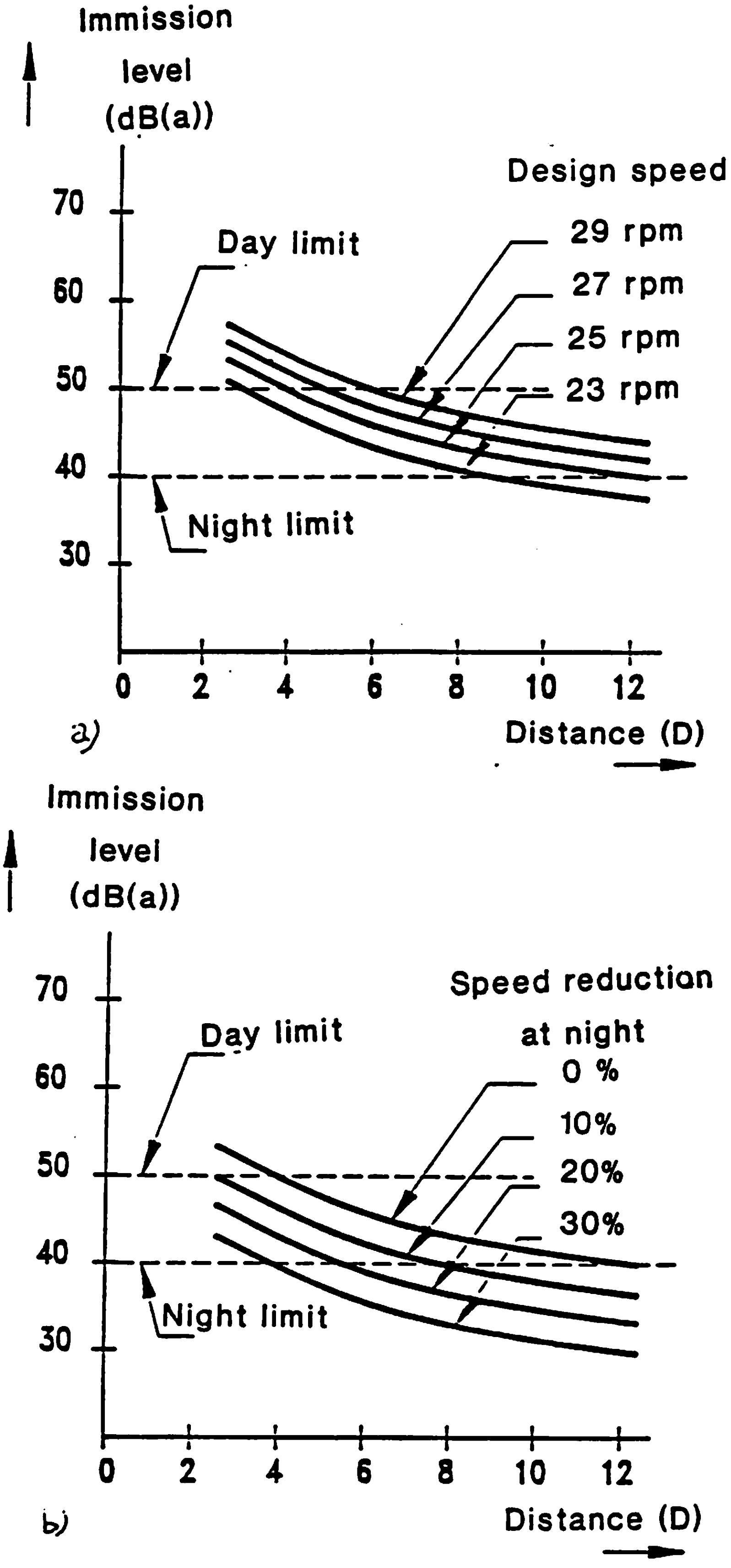


figure 10: Noise immision as function of the turbine distance and a) the design speed and b) the temporary reduction at night

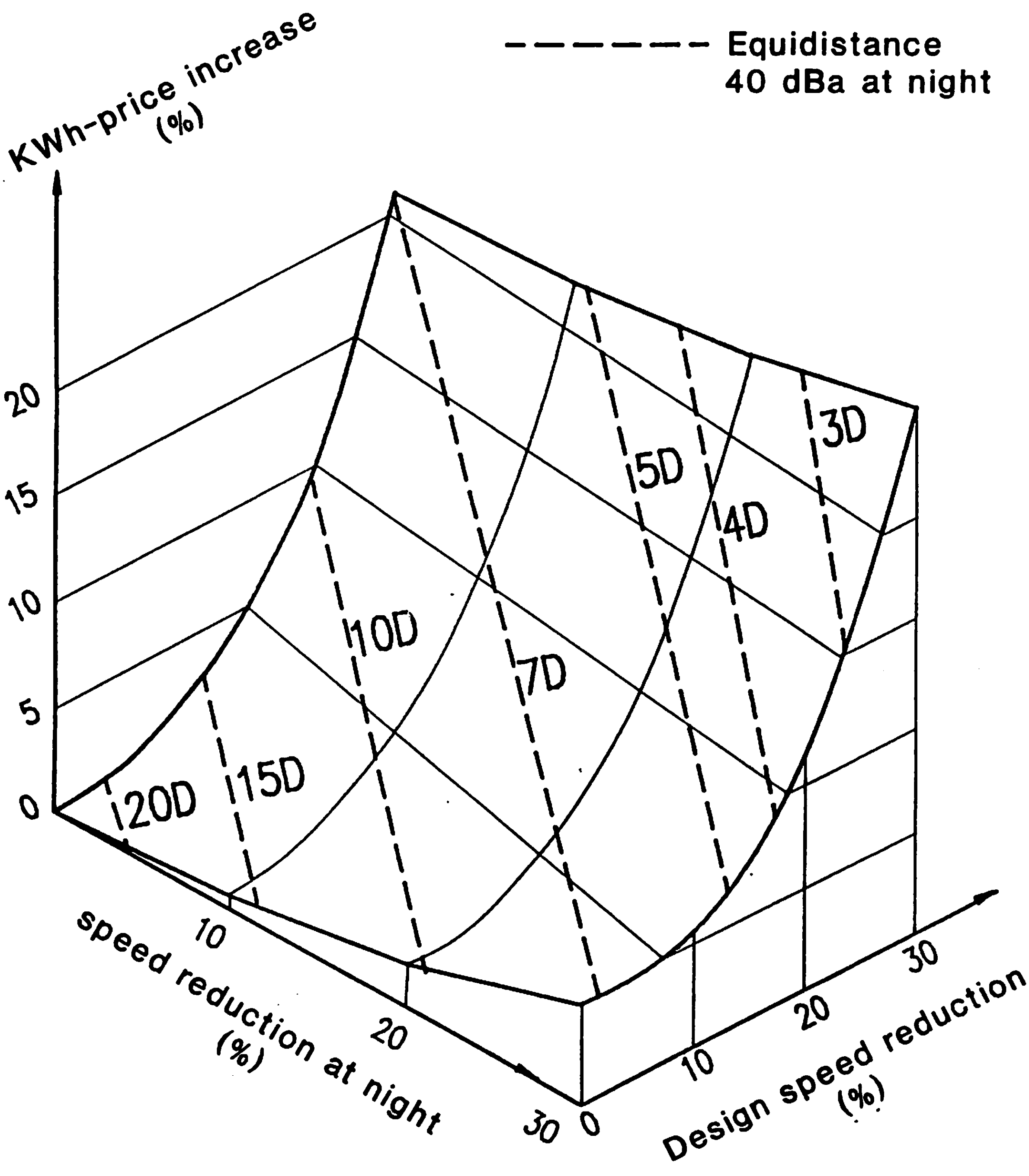


figure 11: Kwh-price increase as function of design speed reduction and temporary reduction at night

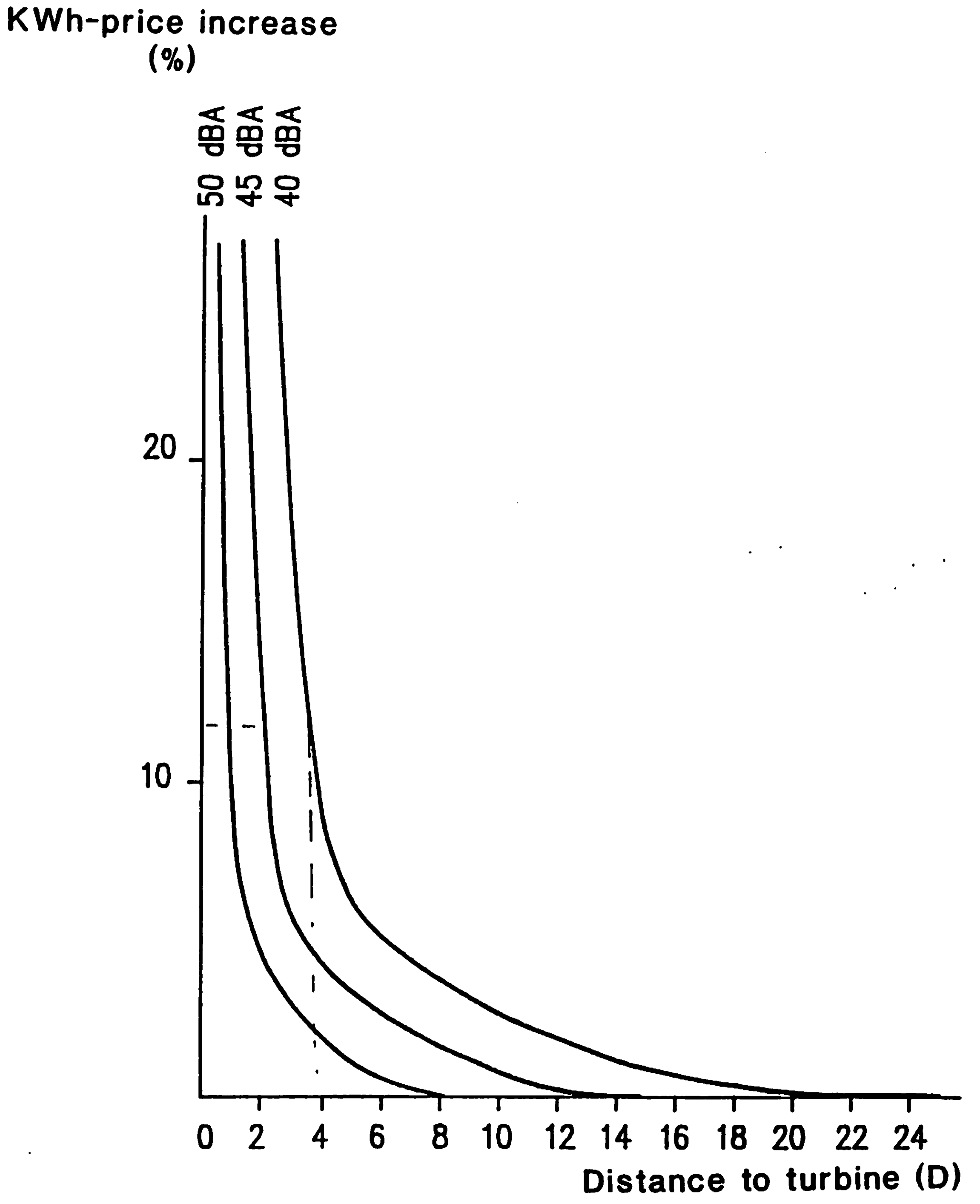
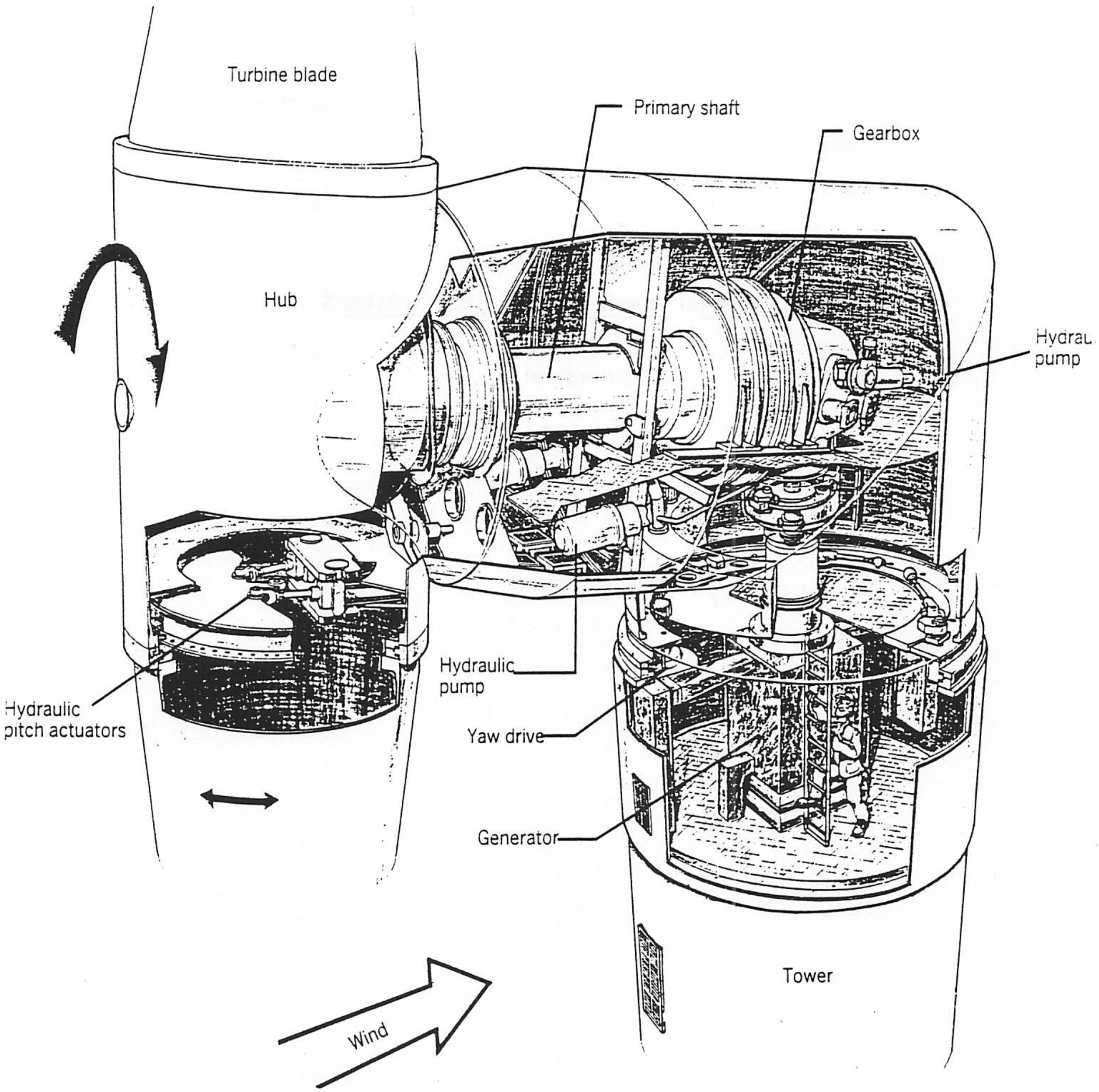


figure 12: kWh-price increase as function of the turbine distance and the required immission level.

MECHANICAL NOISE FROM LARGE WIND TURBINES

S. Ljunggren

THE NÄSUDDEN PROTOTYPE



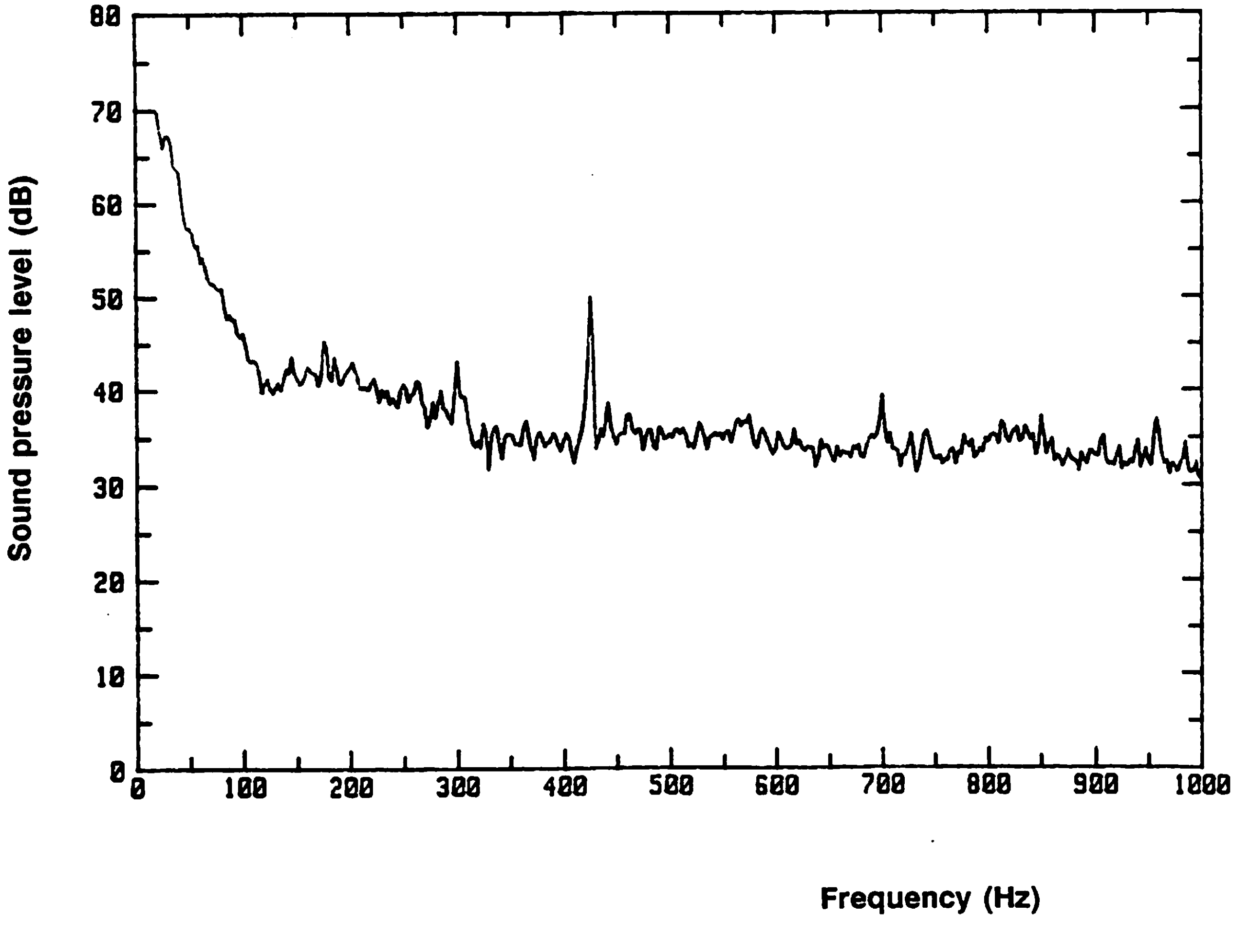
NÄSUDDEN

INTRODUCTION

SOUND PRESSURE LEVEL MEASURED ON THE GROUND AT THE REFERENCE POINT (114 M DOWNSTREAM THE TOWER).

BANDWIDTH: 3.75 HZ.

WIND SPEED: 10-14 m/s. POWER OUTPUT: 2 MW.

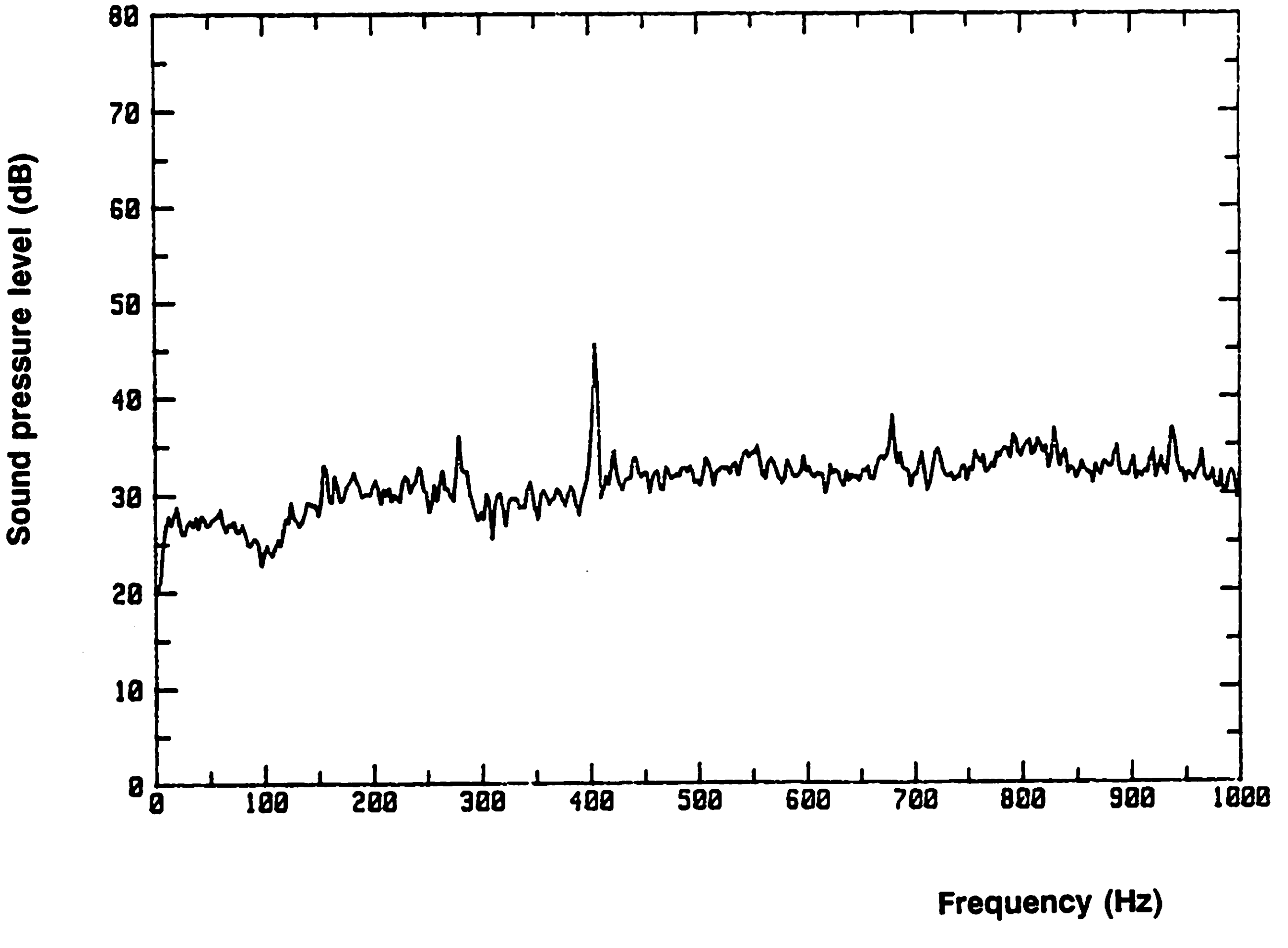


NÄSUDDEN

INTRODUCTION

A-WEIGHTED SOUND PRESSURE LEVEL MEASURED ON THE GROUND AT THE REFERENCE POINT (114 M DOWNSTREAM THE TOWER). BANDWIDTH: 3.75 HZ.

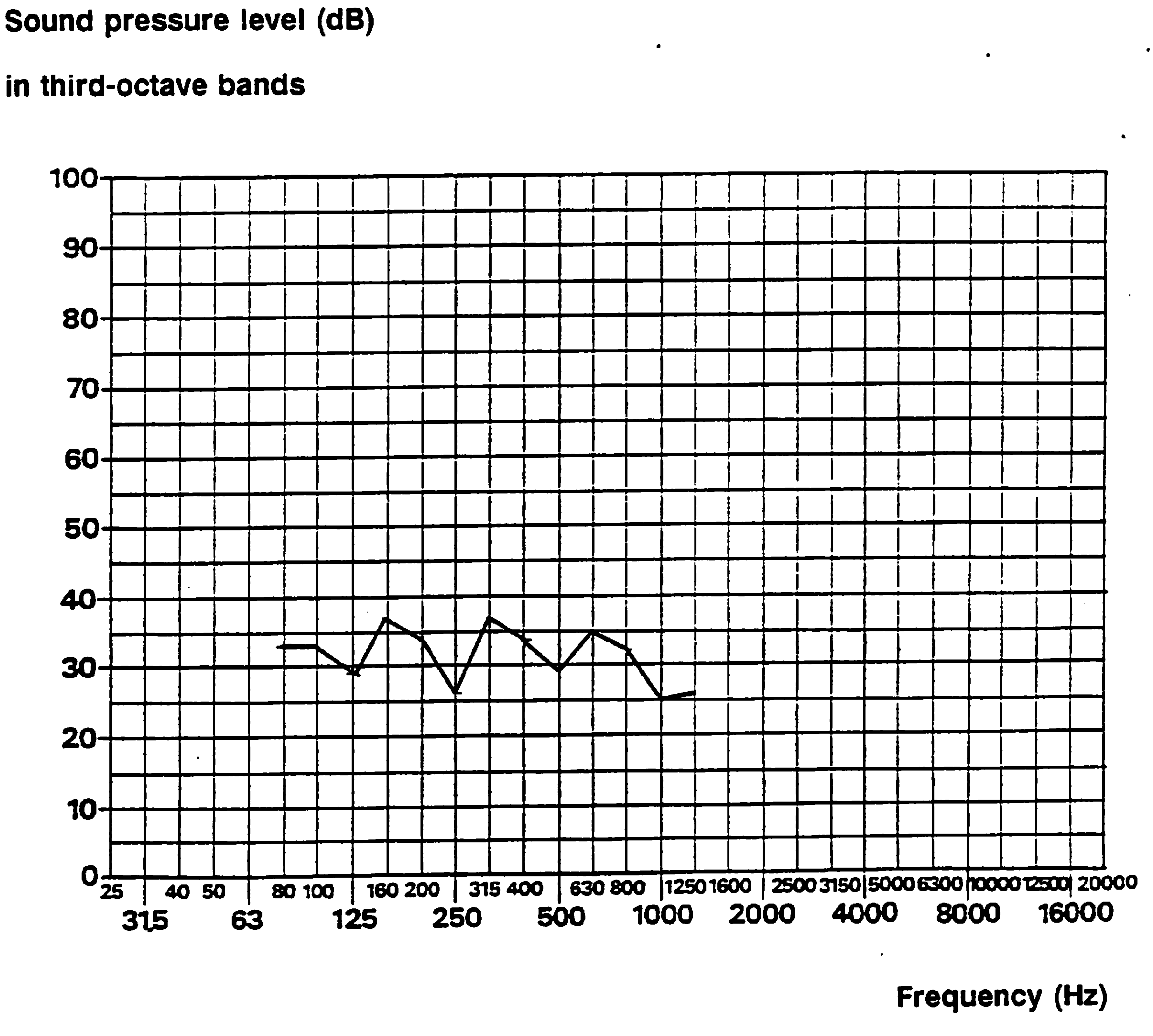
WIND SPEED: 10-14 m/s. POWER OUTPUT: 2 MW.



NÄSUDDEN

IDENTIFICATION OF SOURCES - AN EXAMPLE.

SOUND PRESSURE LEVEL AT THE REFERENCE POINT DUE TO THE PUMP FOR BLADE PITCH CONTROL (THE PUMP ALONE IS WORKING).

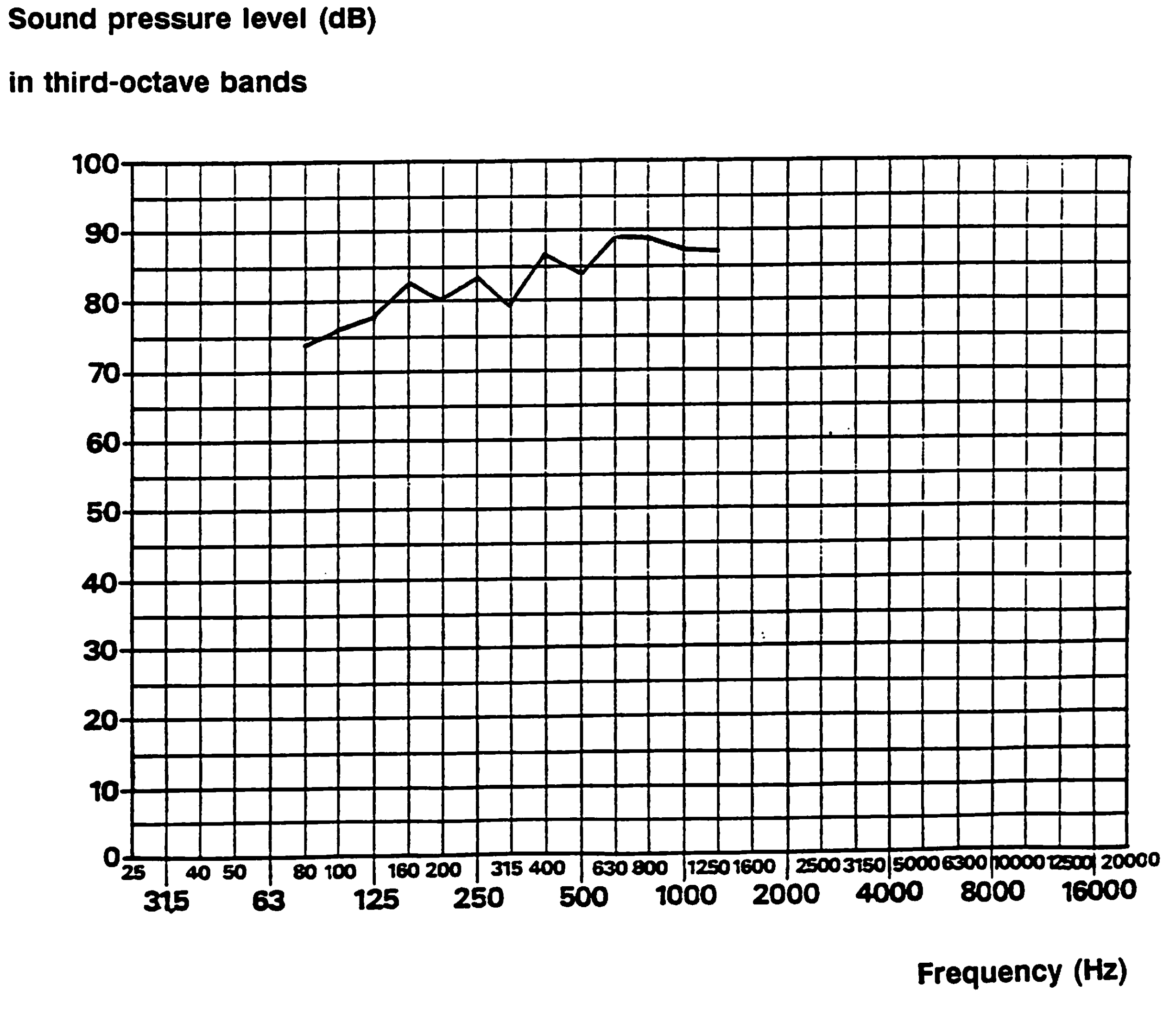


NÄSUDDEN

AIR-BORNE SOUND.

SOUND PRESSURE LEVEL IN THIRD-OCTAVE BANDS IN GENERATOR ROOM. THE CORRESPONDING A-WEIGHTED LEVEL IS 93 dB(A).

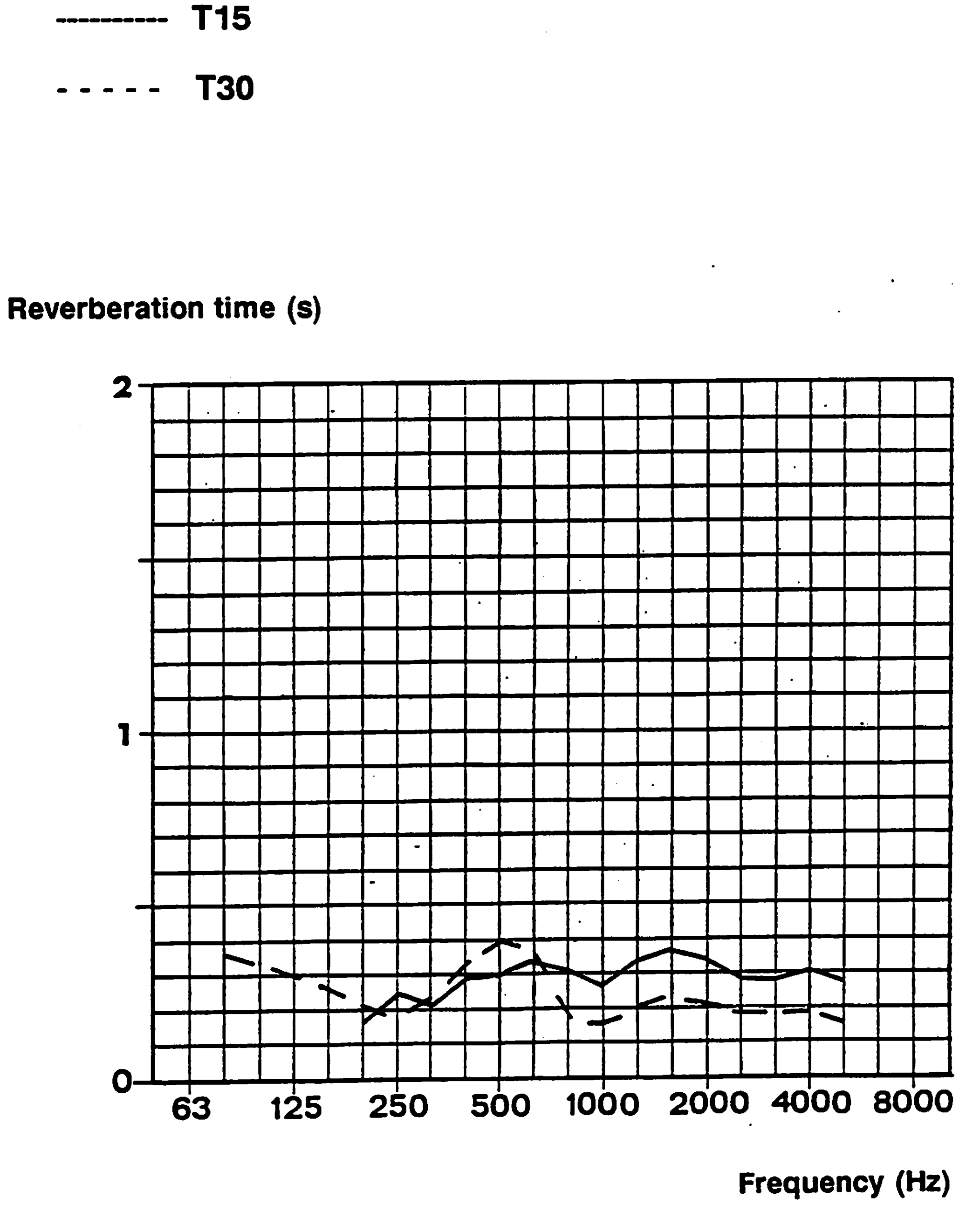
WIND SPEED: 8-10 m/s. POWER OUTPUT: 2 MW.



NÄSUDDEN

AIR-BORNE SOUND

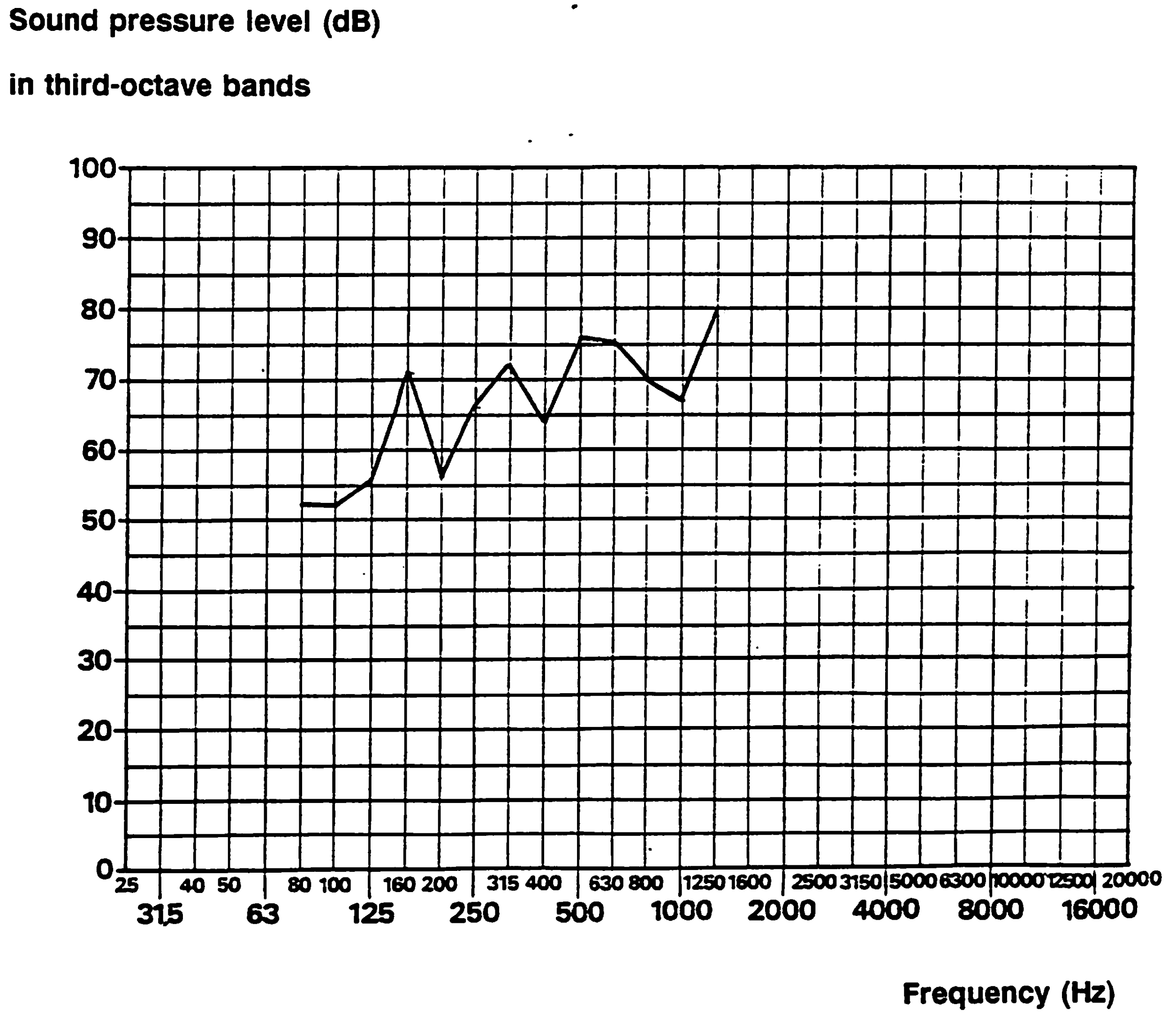
REVERBERATION TIME IN MACHINE HOUSE



NÄSUDDEN

AIR-BORNE SOUND

SOUND PRESSURE LEVEL IN MACHINE HOUSE DUE TO PUMP FOR BLADE PITCH CONTROL (PUMP ONLY IS WORKING)

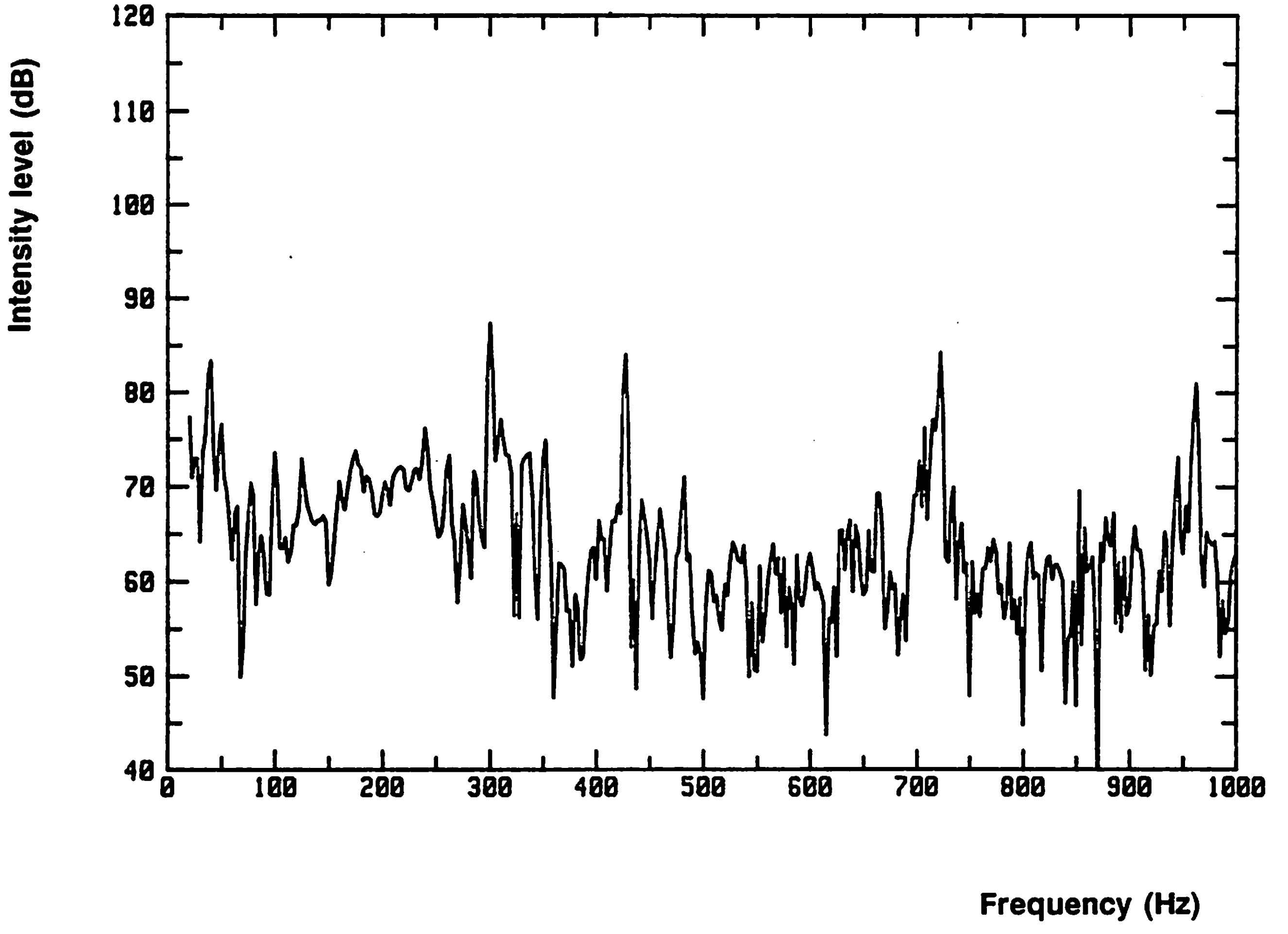


NÄSUDDEN

AIR-BORNE SOUND

SOUND INTENSITY MEASURED OVER A SURFACE CLOSE TO THE GENERATOR. BANDWIDTH: 3.75 Hz.

WIND SPEED: 10-14 m/s. POWER OUTPUT: 2 MW.



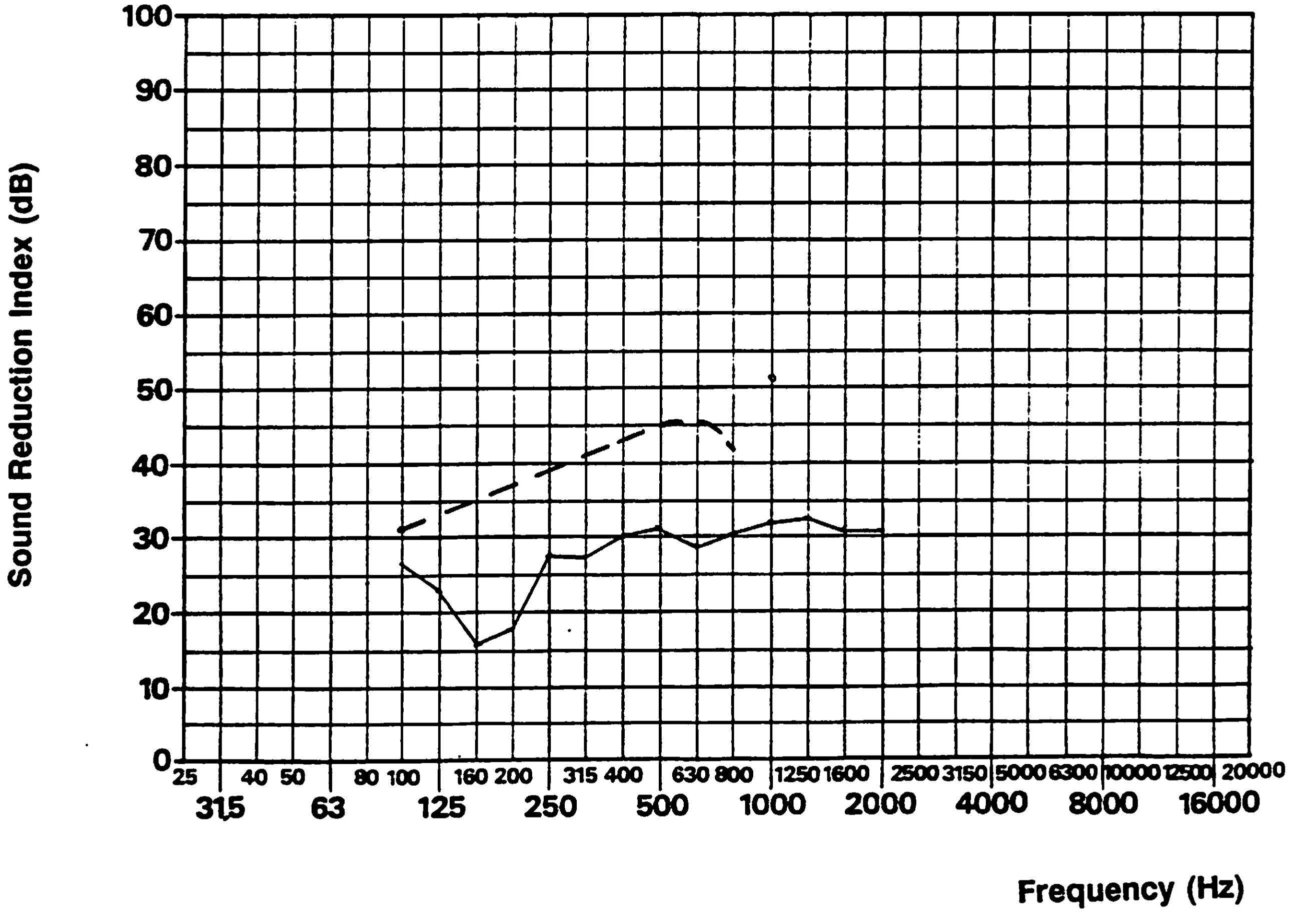
NÄSUDDEN

AIR-BORNE SOUND

SOUND REDUCTION INDEX OF MACHINE HOUSE WALLS

SOLID LINE: MEASURED USING LOUDSPEAKER INSIDE AND MICROPHONES INSIDE AND 3 M OUTSIDE THE WALLS, FLOOR AND ROOF.

DOTTED LINE: CALCULATED ASSUMING MASS LAW.



NÄSUDDEN

AIR-BORNE SOUND

CALCULATED LEVELS AT THE REFERENCE POINT ON THE GROUND.

THE CALCULATIONS ARE BASED ON MEASURED SOUND POWER LEVELS IN THE MACHINE HOUSE AND ON THE MEASURED SOUND REDUCTION INDEX OF THE MACHINE HOUSE WALLS.

NOISE SOURCE	SOUND LEVEL IN dB(A)
GENERATOR	35
GEAR BOX	32
HYDRAULIC EQUIPMENT	24

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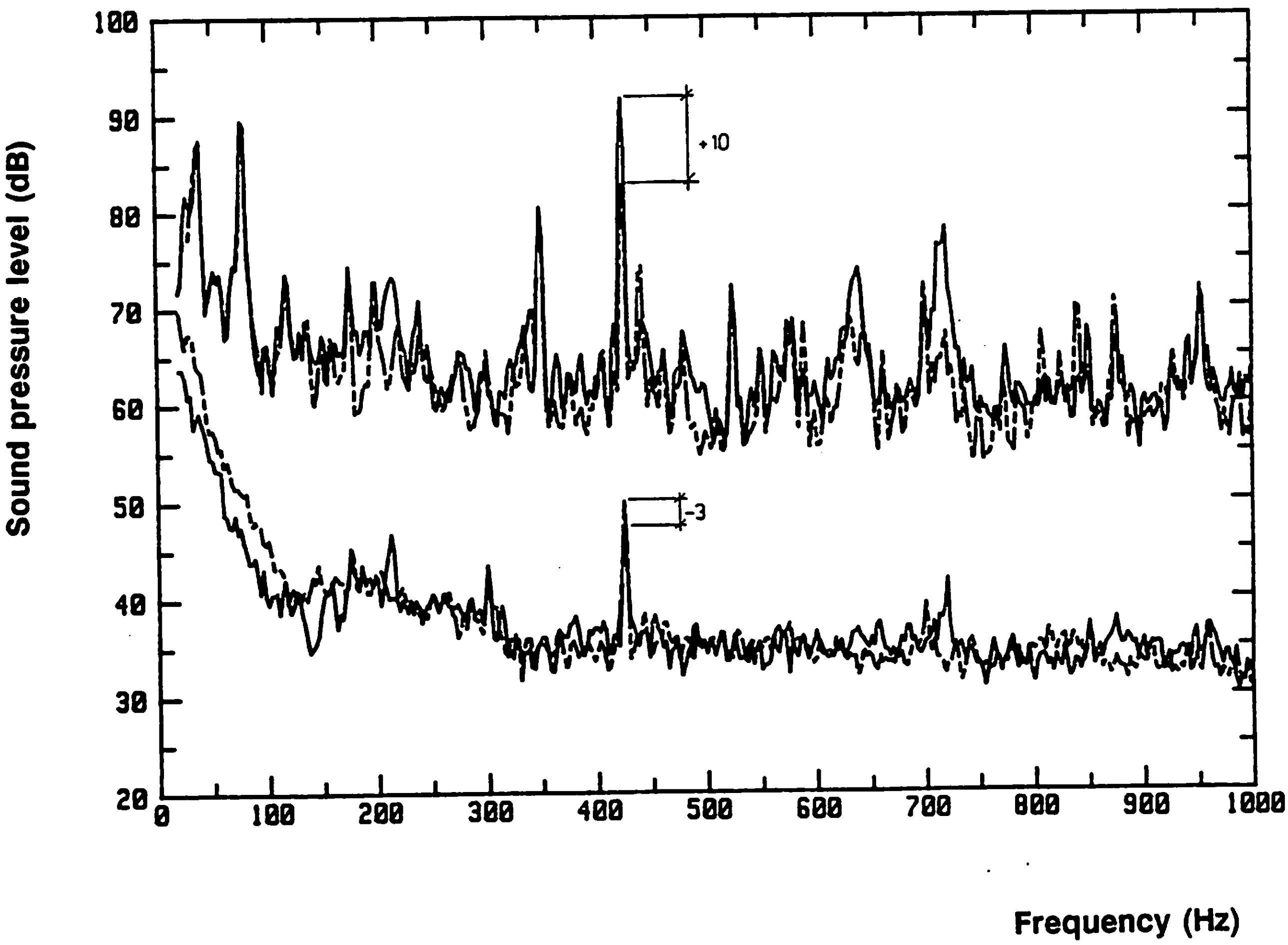
AIR-BORNE SOUND VS STRUCTURE-BORNE SOUND

THE TWO UPPER CURVES SHOW THE MACHINE ROOM LEVELS; THE TWO LOWER CURVES SHOW THE LEVELS ON GROUND.

SOLID LINE: SOUND PRESSURE LEVEL RAISED 10 dB AT 425 Hz WITH THE HELP OF A LOUDSPEAKER.

DOTTED LINE: MACHINERY NOISE ONLY.



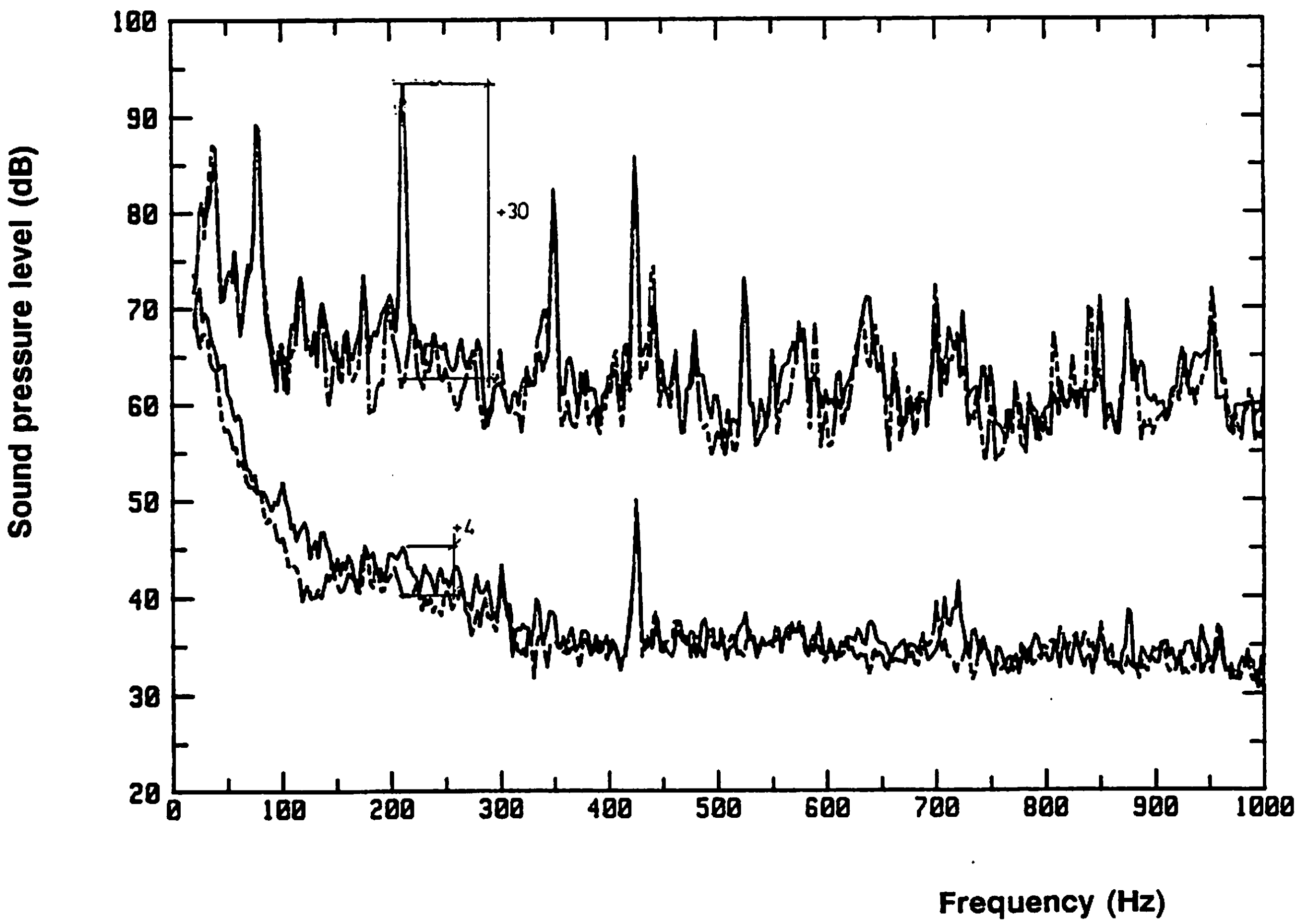
NÄSUDDEN

AIR-BORNE SOUND VS STRUCTURE-BORNE SOUND

THE TWO UPPER CURVES SHOW THE MACHINE ROOM LEVELS; THE TWO LOWER CURVES SHOW THE LEVELS ON GROUND.

SOLID LINE: SOUND PRESSURE LEVEL RAISED 30 dB AT 212 Hz WITH THE HELP OF A LOUDSPEAKER.

DOTTED LINE: MACHINERY NOISE ONLY.



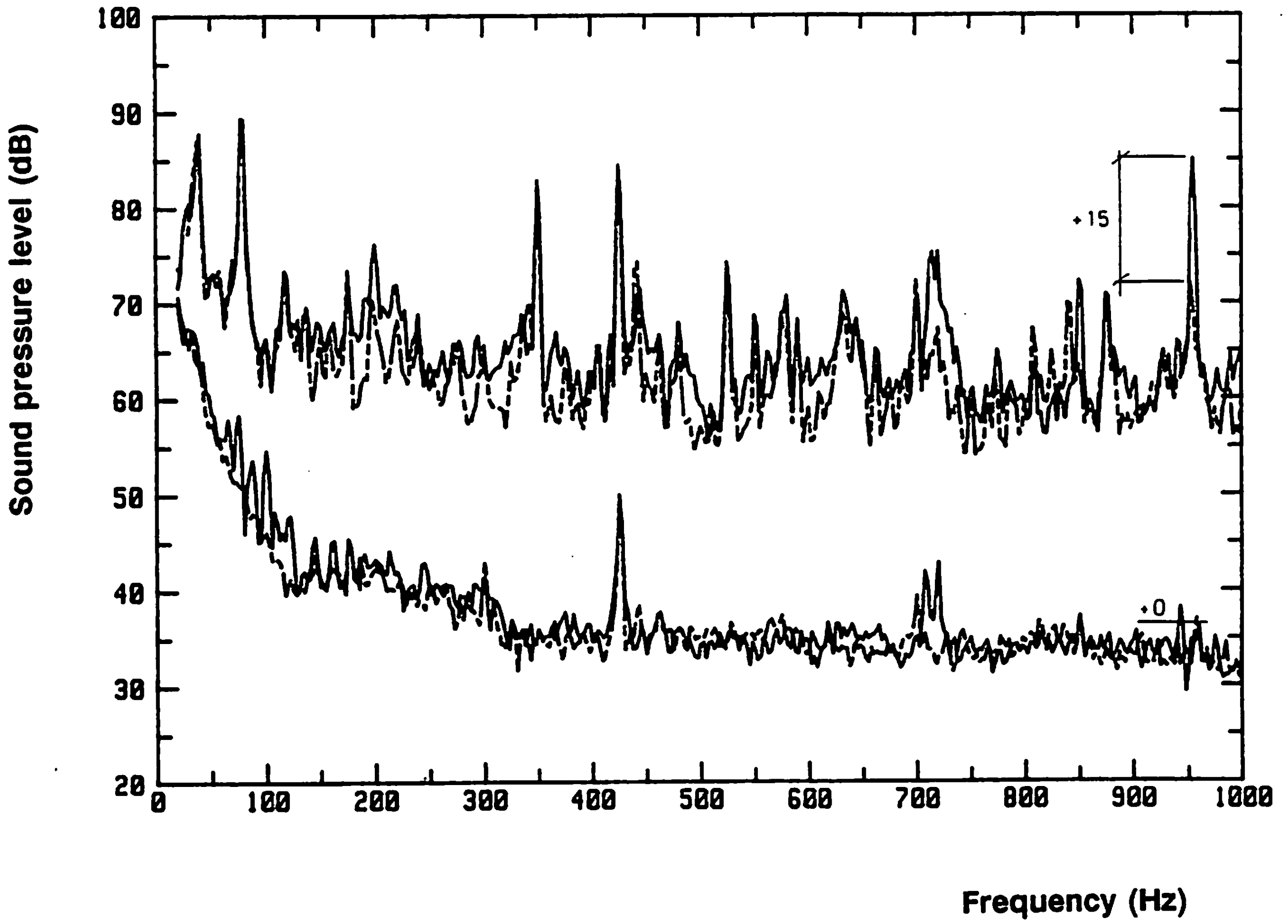
NÄSUDDEN

AIR-BORNE SOUND VS STRUCTURE-BORNE SOUND

THE TWO UPPER CURVES SHOW THE MACHINE ROOM LEVELS; THE TWO LOWER CURVES SHOW THE LEVELS ON GROUND.

SOLID LINE: SOUND PRESSURE LEVEL RAISED 15 dB AT 955 Hz WITH THE HELP OF A LOUDSPEAKER.

DOTTED LINE: MACHINERY NOISE ONLY.

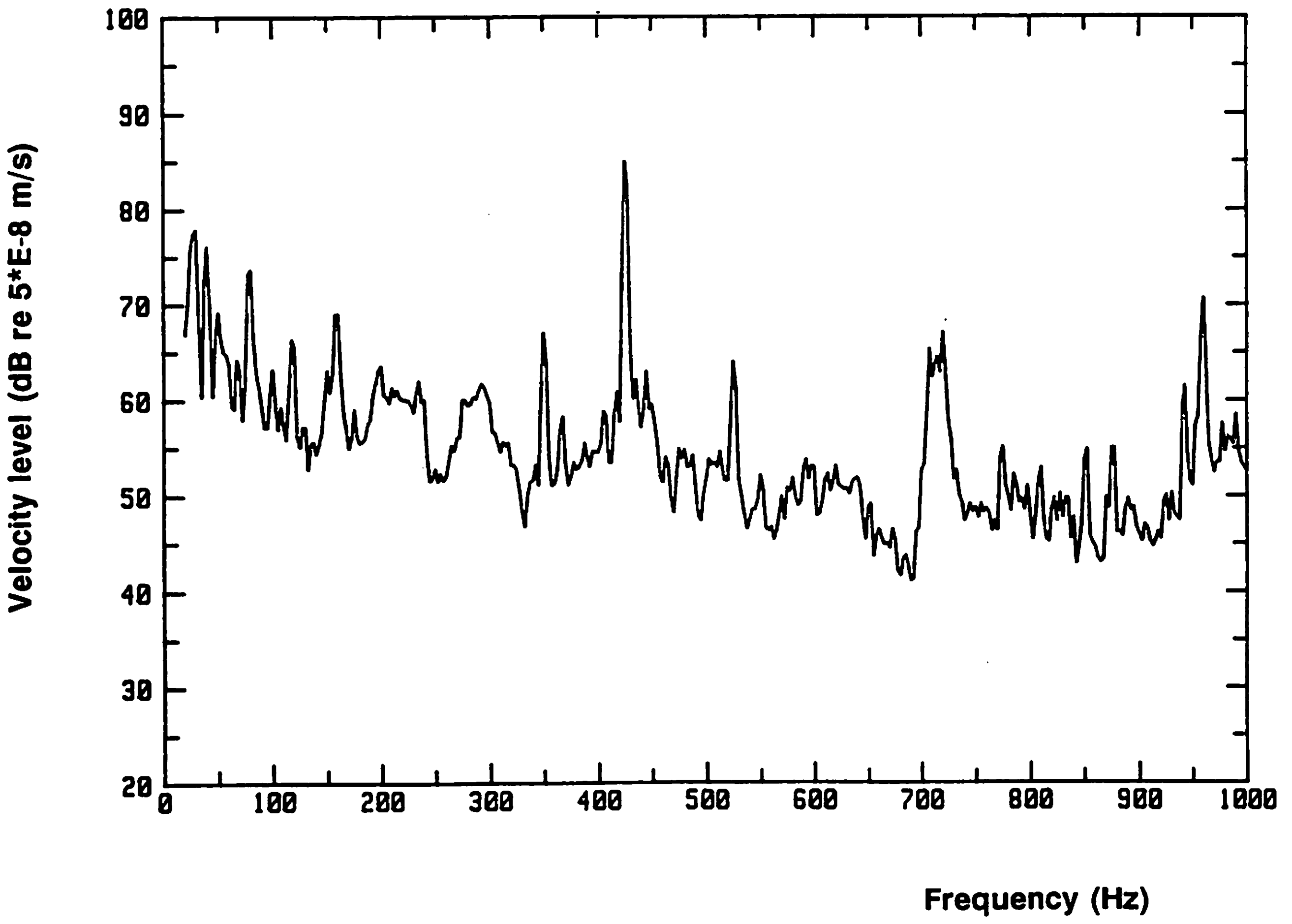


NÄSUDDEN

STRUCTURE-BORNE SOUND

VELOCITY LEVEL (IN dB RE 5*E-8 M/S, BANDWIDTH 3.75 Hz) OF GEAR BOX FOUNDATION.

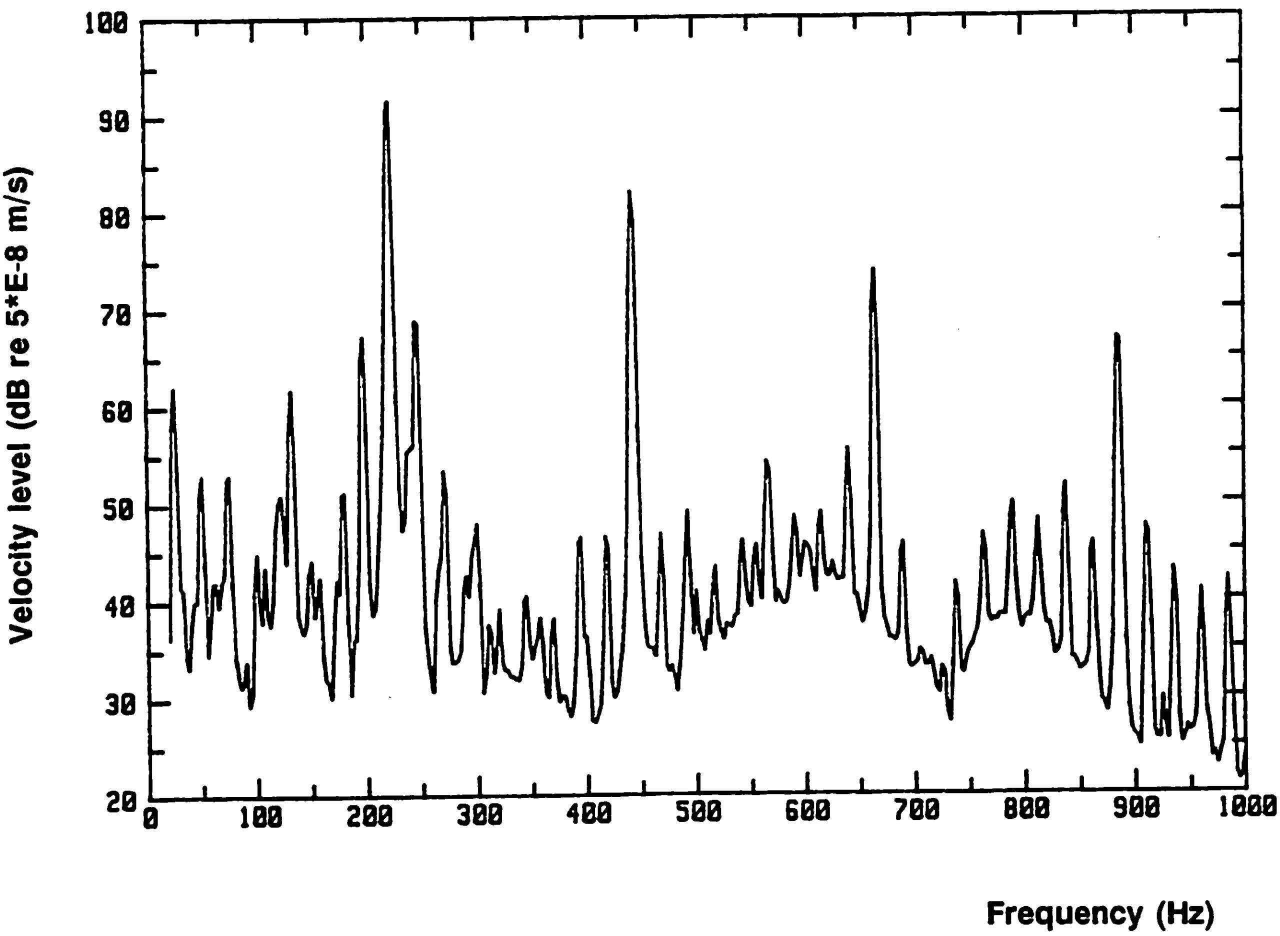
WIND SPEED: 10-14 m/s. POWER OUTPUT: 2 MW.



NÄSUDDEN

STRUCTURE-BORNE SOUND

VELOCITY LEVEL OF LUBRICATING OIL PUMP. PUMP ONLY IS WORKING (IDLING).

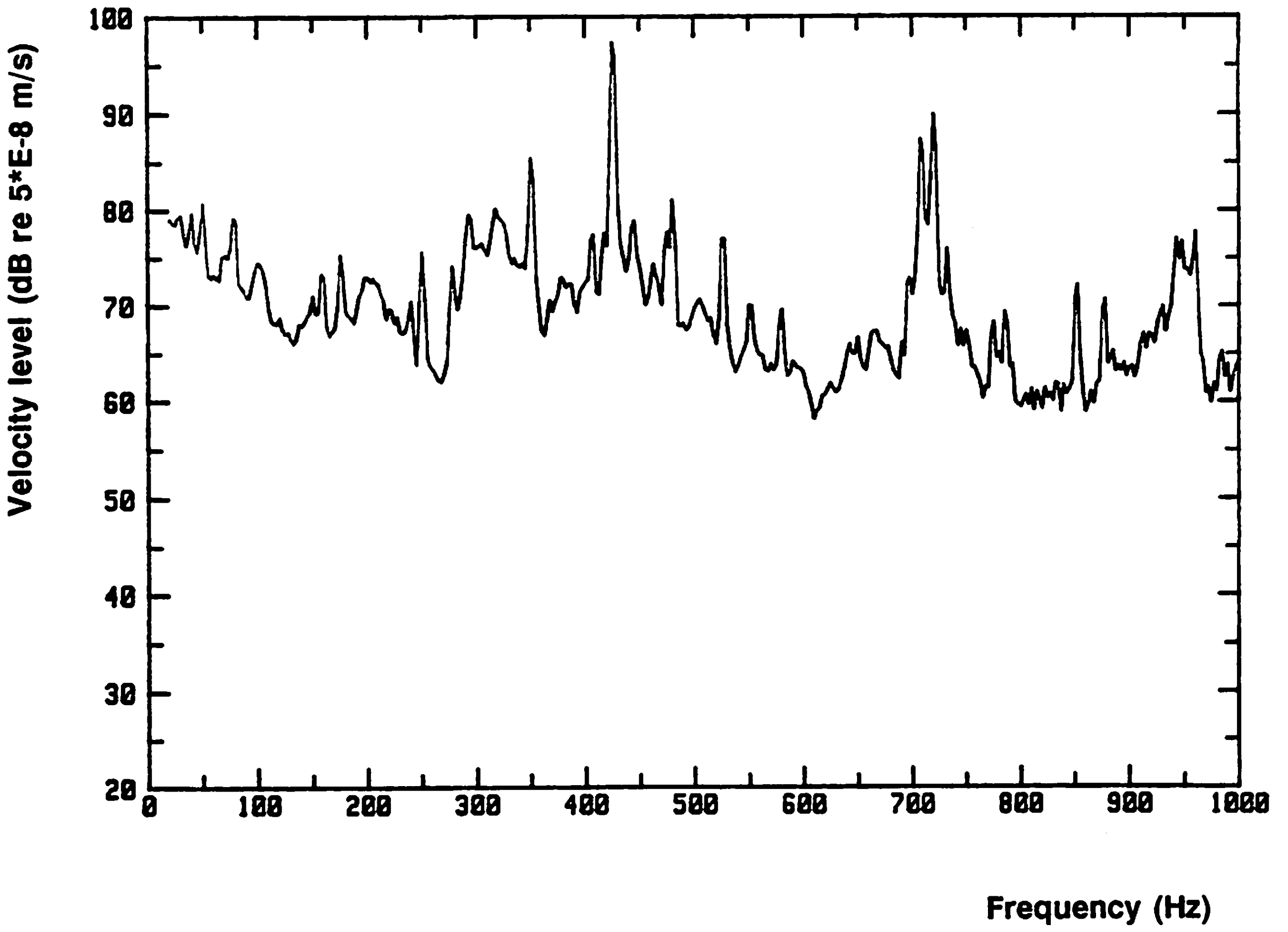


NÄSUDDEN

STRUCTURE-BORNE SOUND

VELOCITY LEVEL ON MACHINE HOUSE WALL (MEAN VALUE OVER ONE WALL).

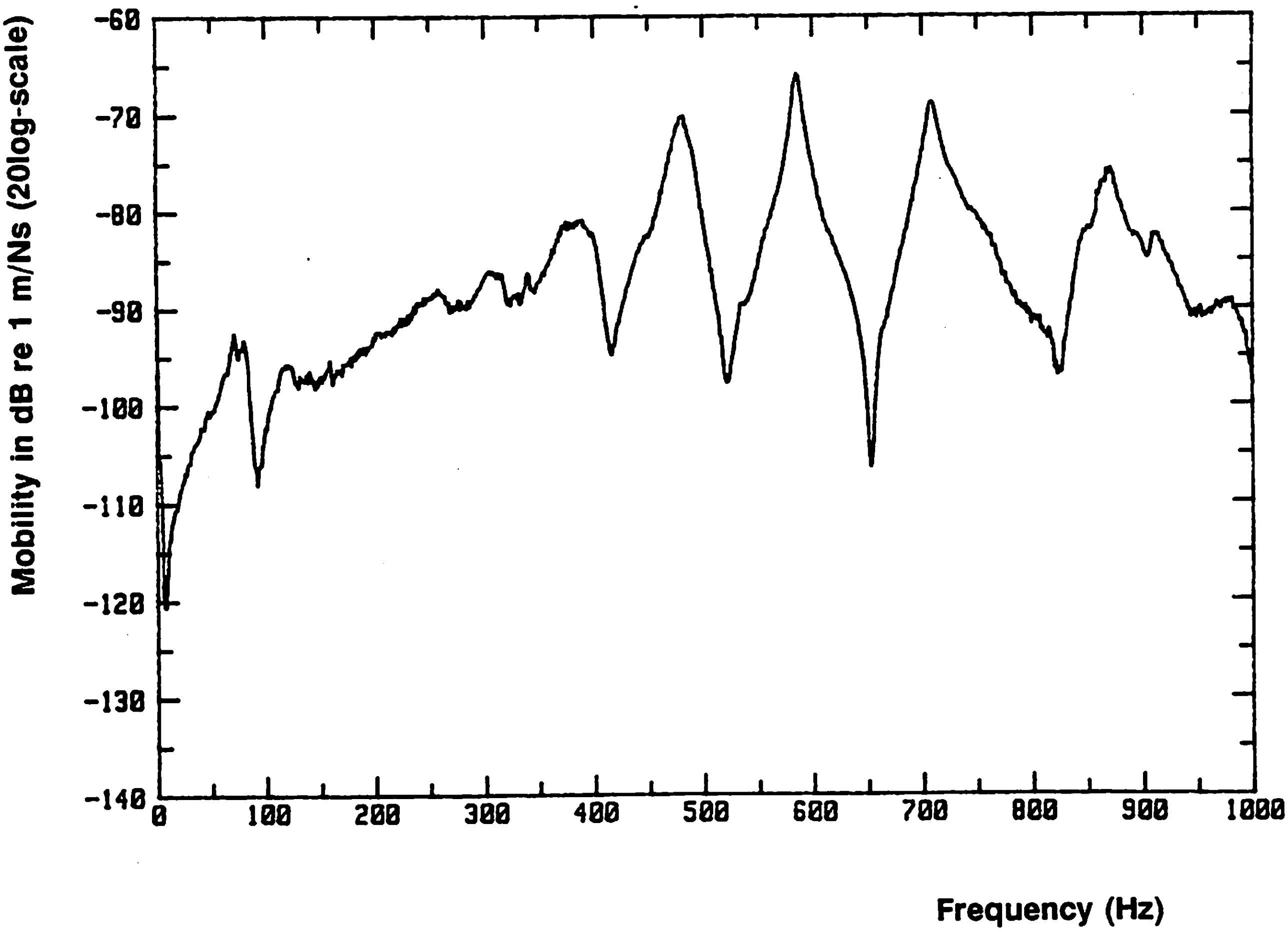
WIND SPEED: 8-10 m/s. POWER OUTPUT: 2 MW.



NÄSUDDEN

STRUCTURE-BORNE SOUND

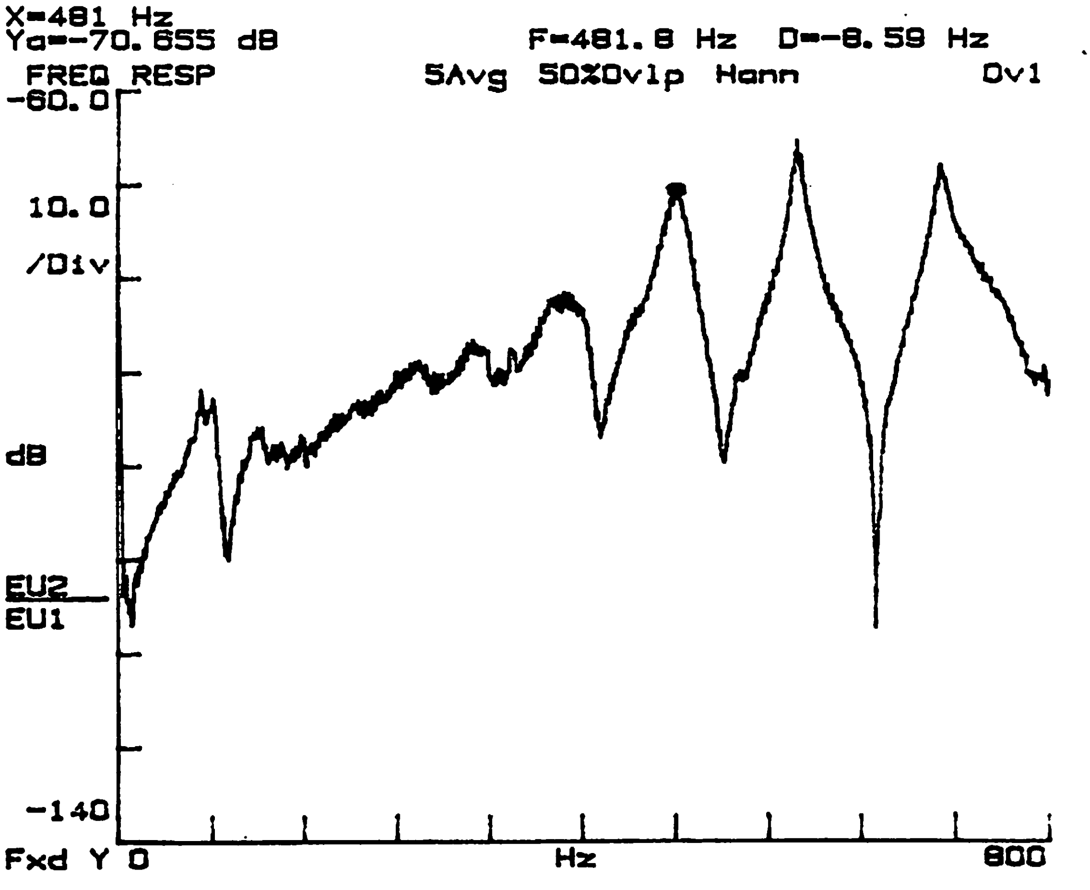
POINT MOBILITY (IN dB RE 1 m/Ns, 20log-SCALE) OF MACHINE HOUSE ROOF.



NÄSUDDEN

STRUCTURE-BORNE SOUND

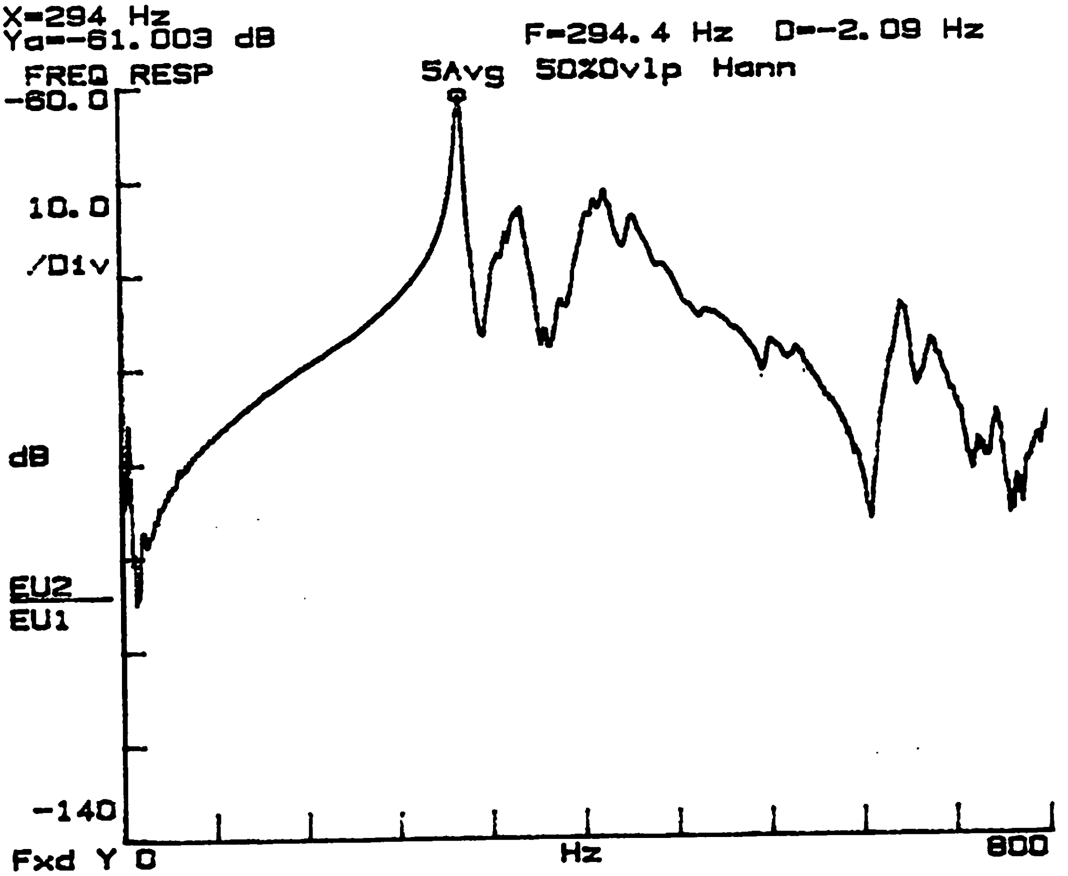
POINT MOBILITY (IN dB RE 1 m/Ns, 20log-SCALE) OF MACHINE HOUSE WALL. THE HALF-POWER BANDWIDTH OF THE PEAK AT 481.8 Hz IS 8.59 Hz, WHICH GIVES A LOSS FACTOR OF 1.8%



NÄSUDDEN

STRUCTURE-BORNE SOUND

POINT MOBILITY (IN dB RE 1 m/Ns, 20log-SCALE) OF MACHINE HOUSE WALL. THE HALF-POWER BANDWIDTH OF THE PEAK AT 294.4 Hz IS 2.09 Hz, WHICH GIVES A LOSS FACTOR OF 0.7%



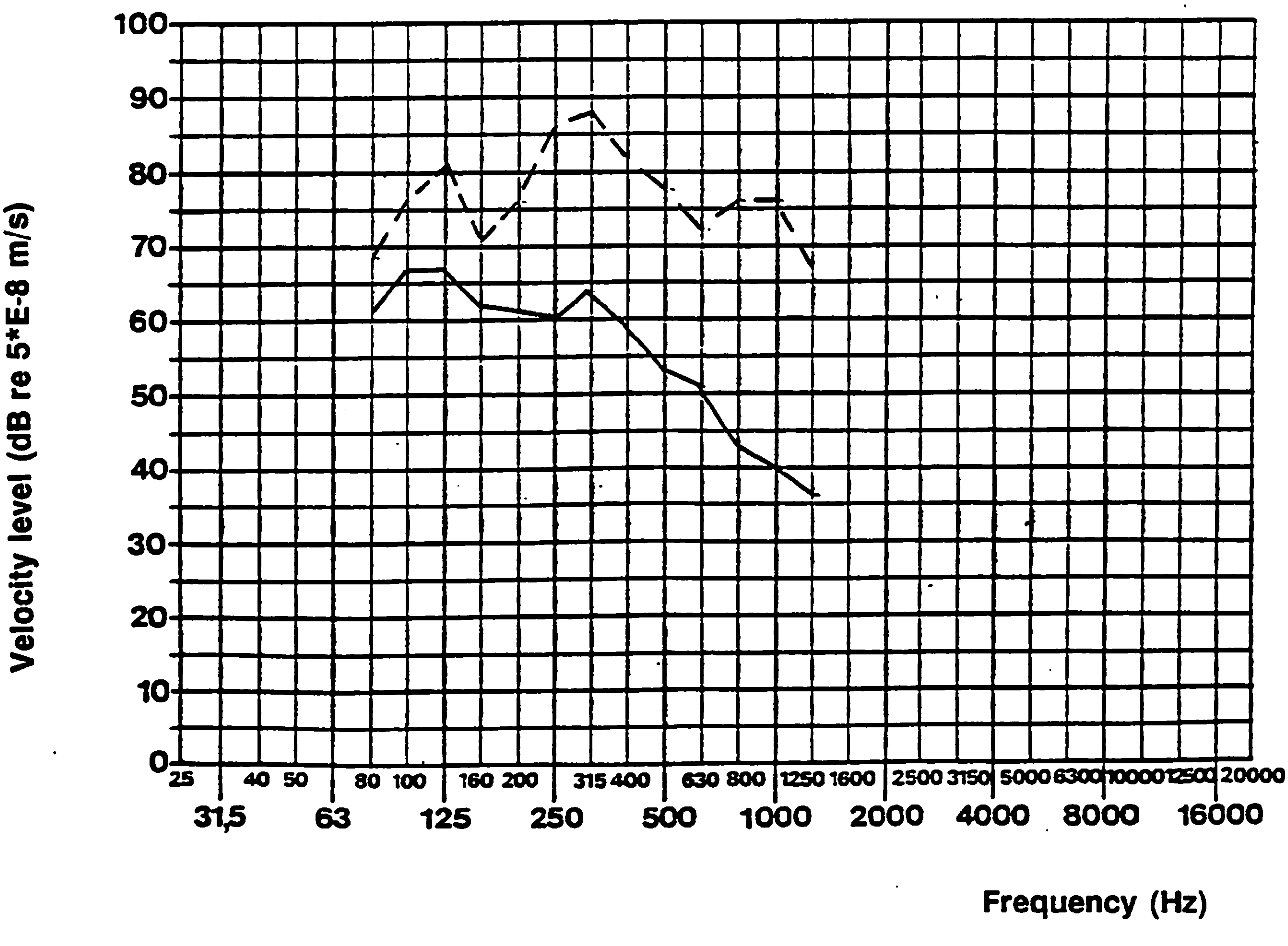
NÄSUDDEN

STRUCTURE-BORNE SOUND

MEASURED VELOCITY LEVELS. TURBINE WORKING.

SOLID LINE: CONCRETE TOWER

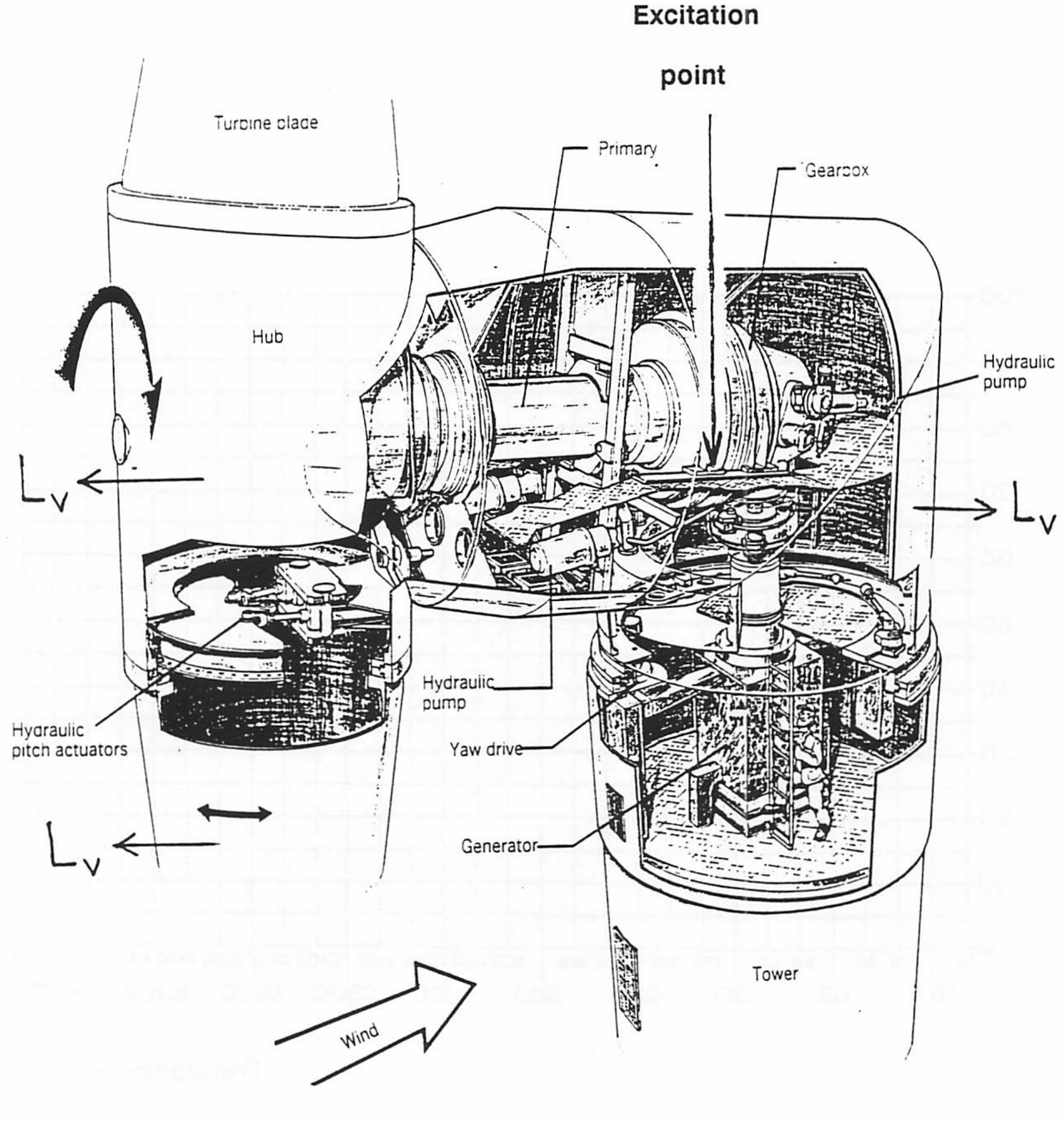
DOTTED LINE: MACHINE HOUSE WALL



NÄSUDDEN

STRUCTURE-BORNE SOUND

SHAKER EXCITATION OF GEAR BOX FOUNDATION TO EVALUATE RADIATION FROM DIFFERENT PARTS.



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STRUCTURE-BORNE SOUND

RADIATION OF GEAR NOISE (CALCULATED FROM SHAKER MEASUREMENTS)

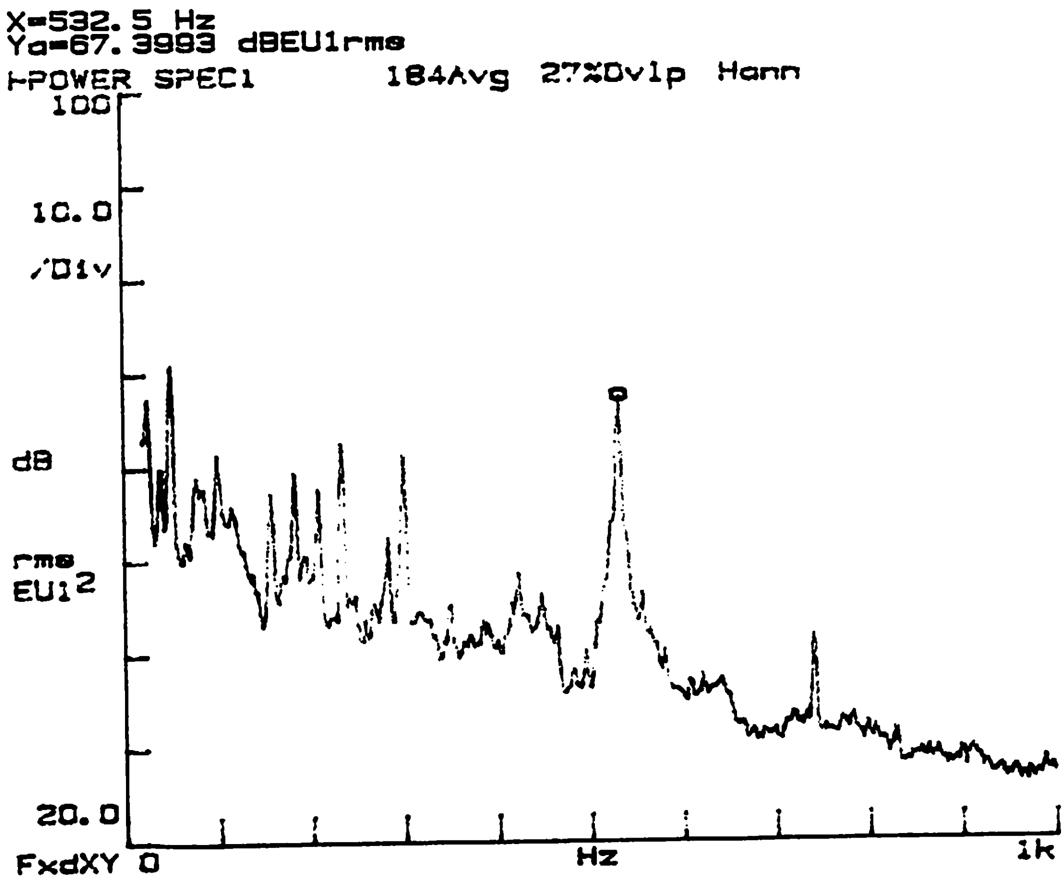
RADIATING SURFACE	SOUND LEVEL, dB(A)
MACHINE HOUSE	55
HUB	47
BLADES	49
TOWER	29

MAGLARP

STRUCTURE-BORNE SOUND

MEASURED VELOCITY LEVEL OF MACHINE HOUSE WALL.

TURBINE WORKING.

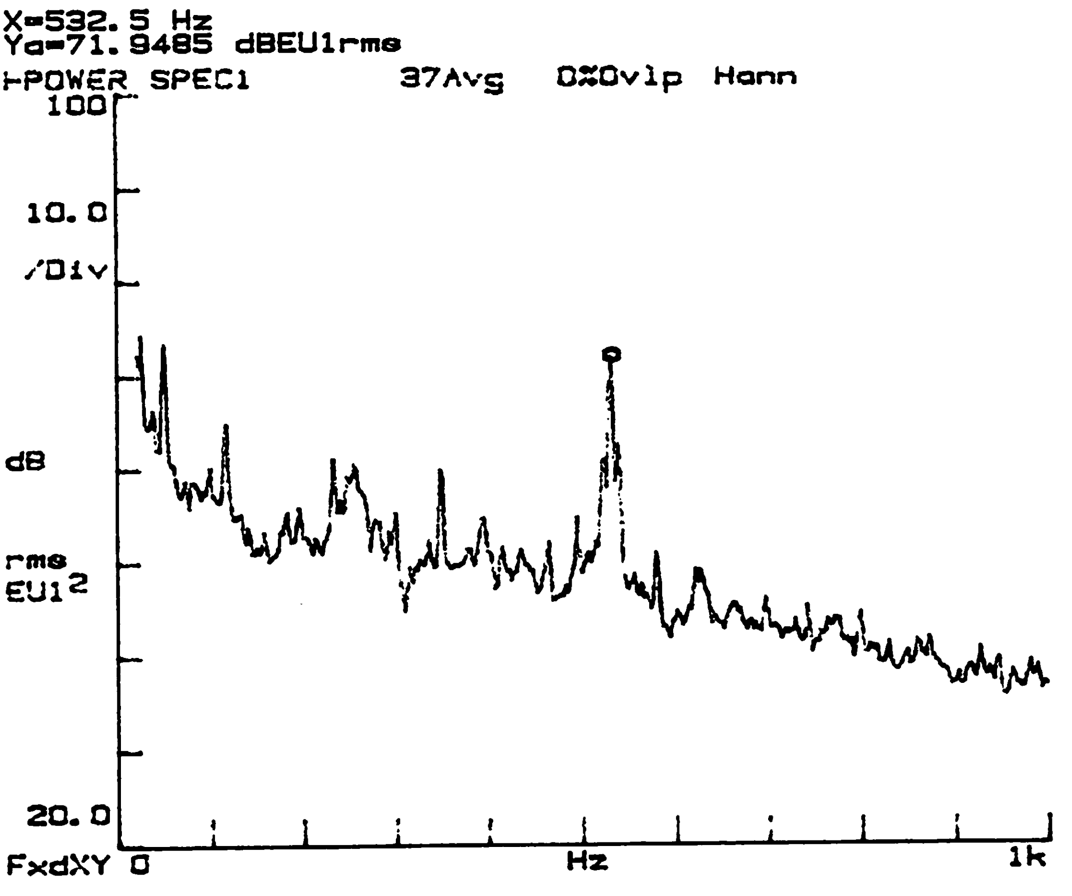


MAGLARP

STRUCTURE-BORNE SOUND

MEASURED VELOCITY LEVEL OF STEEL TOWER.

TURBINE WORKING.

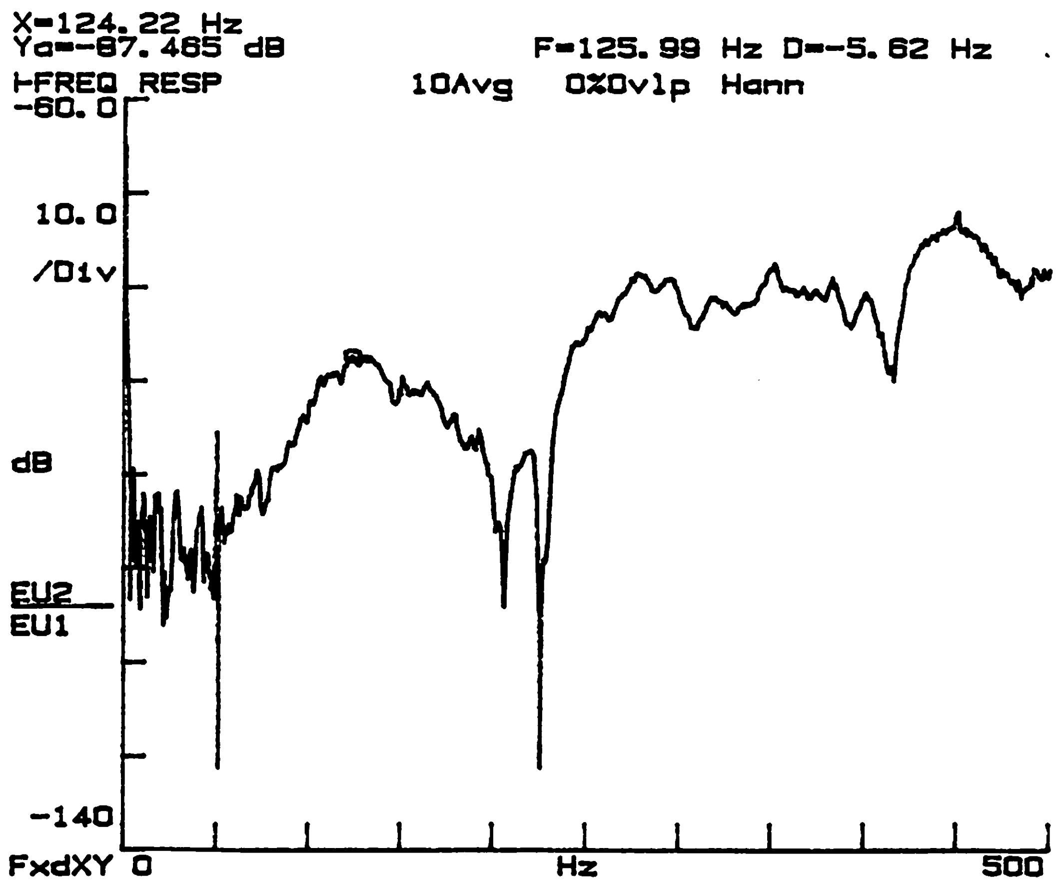


MAGLARP

STRUCTURE-BORNE SOUND

MEASURED POINT MOBILITY OF MACHINE HOUSE WALL.

THE LOSS FACTOR AT THE PEAK AT 125.99 Hz IS 4.5%

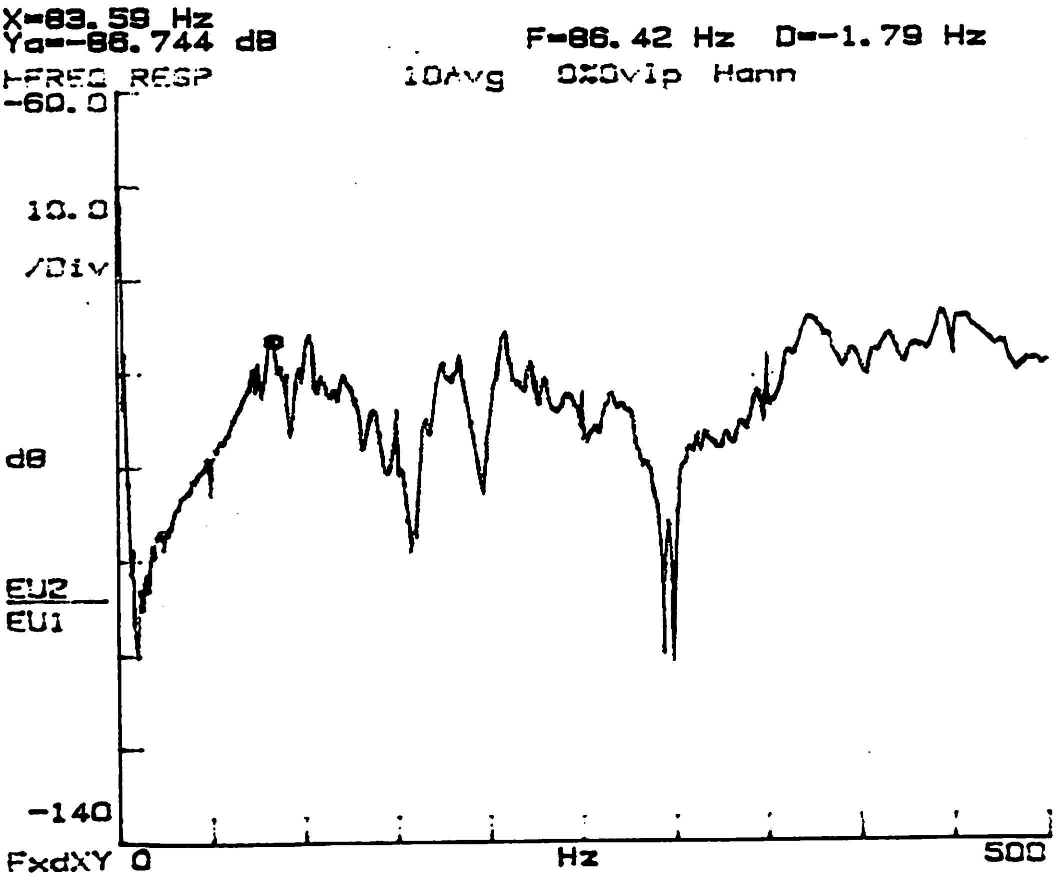


MAGLARP

STRUCTURE-BORNE SOUND

MEASURED POINT MOBILITY OF STEEL TOWER.

THE LOSS FACTOR AT THE PEAK AT 86.42 Hz IS 2.1%



CONCLUSIONS

THE MEASUREMENTS SHOW THAT FOR THE TWO SWEDISH PROTOTYPES

- STRUCTURE-BORNE SOUND IS MORE IMPORTANT THAN AIR-BORNE SOUND

- THE GEAR BOX IS THE PREDOMINANT NOISE SOURCE (SPUR GEAR!)

- THE STRUCTURE-BORNE SOUND FROM THE GEAR BOX IS FED INTO THE STRUCTURE BY MOTION IN THE HORIZONTAL PLANE

- RADIATION FROM A STEEL TOWER MAY BE IMPORTANT.

- THE RADIATION FROM A CONCRETE TOWER IS NOT IMPORTANT

- RADIATION FROM HUB AND TURBINE BLADES IS SOMEWHAT SMALLER THAN RADIATION FROM MACHINE HOUSE

- RADIATION FROM A MACHINE HOUSE MAY BE HEAVILY INFLUENCED BY WIDELY SPACED RESONANCES

- THE LOSS FACTOR OF THE MACHINE HOUSE IS IMPORTANT

- THE SOUND REDUCTION INDEX OF A MACHINE HOUSE MADE FROM STEEL MAY BE FAIRLY LOW DUE TO RESONANT TRANSMISSION CAUSED BY STIFFENERS

- SEPARATE FRAME FOR POWER LINE SEEMS TO BE ADVANTAGEOUS COMPARED TO A MONOCOQUE DESIGN

THE INFLUENCE OF A WIND TURBINE'S GEOMETRICAL AND OPERATIONAL PARAMETERS ON AERODYNAMIC NOISE GENERATION AND ON ENERGY PRODUCTION

S. Meijer

SUMMARY

A short description of a simple semiempirical method to estimate broadband wind turbine noise due to aerodynamic noise sources is given in this paper.

Two alternative ways of scaling the noise due to inflow turbulence are discussed, using measurements from the large wind turbine at Näsudden for comparisons with calculations. During the measurements the rotational speed was varied between 25 rpm and 12.5 rpm. In one method the fluctuating velocity is considered to be proportional to the wind speed and in the other method the fluctuating velocity is considered to be proportional to the relative velocity between the blade and the air. It is concluded that the first method seems to make it easier to explain the measurement results.

The effects on the noise generation and the energy production of choosing to optimize a wind turbine at different tip speed ratios and for different number of blades are finally discussed. It is concluded that choosing the design in such a way as to decrease the noise has no negative effects on the production of energy.

NOISE CALCULATION METHOD

The semiempirical method to estimate aerodynamic broadband noise described here is implemented in a computer program at FFA. FFA also has computer programs for the calculation of the low frequency noise due to the passage of a wind turbine blade close to the wind turbine tower, but these will not be described here.

The variation of the sound level with parameters such as blade velocity, blade length etc is given by theoretical considerations. Measurements of the noise from the big prototypes at Näsudden and Maglarp have been used together with experimental results from [1] and [2] to get realistic noise levels.

In the noise measurements at the prototype sites the total noise was measured and this means that it was not possible to discern the contributions from the trailing edge bluntness. The level of the trailing edge bluntness noise is based wholly on the results in [1] .

The noise sources that are considered are :

Turbulent boundary layer trailing edge noise (TBL-TE) with a sharp trailing edge.

Noise due to a blunt trailing edge (BE).

Noise due to inflow turbulence (TURB).

For the TBL-TE noise the overall sound pressure level, L_{TBL-TE}, in a 1/3 octave band from a blade element of length dl at the radial position l is is given by :

$$L_{TBL-TE} = 10\log_{10}(dlBr^{-2}\delta U^5) - F_0(R_e) + F_1(f)$$

where

B is the number of blades

r is the distance to the observer

$\delta = 2.569CC_d$ is the boundary layer thickness

C is the chord width of the blade element

C_d is the drag coefficient

U is the rotational speed of the blade element

R_e is the Reynolds number based on the blade speed and the chord

F_0 is a function based on the results in [2]

F_1 is a function based on a nondimensional spectrum given by Fink [3]

f is a nondimensional frequency.

For the BE noise the sound level in a 1/3 octave band is given by :

$$L_{BE} = 10\log_{10}(dlBr^{-2}tU^{5.3}) + F_2(f)$$

where

t is the blunt edge thickness

F_2 is a function derived from the experiments in [1]

For the TURB noise the sound level in a 1/3 octave band is given by :

$$L_{TURB} = 10\log_{10}(dlBr^{-2}CU^4w^2) + F_3(f)$$

where

w is the turbulent velocity variation normal to the blade element

w/V is the turbulence intensity and

V is the wind speed.

F_3 is a function derived from an analysis of a thin airfoil in a turbulent flow with an idealized isotropic turbulence model [4]

The noise contributions from all blade elements are summed as if the noise sources are uncorrelated.

It must be pointed out that although the method described above gives a reasonable variation of the sound power level with parameters such as rotational speed and turbine size, it is too blunt an instrument to be used to answer questions about how the detailes of the blade geometry influence the sound level.

ALTERNATIVE MODELS FOR THE TURBULENT INFLOW NOISE

As was pointed out in [5] models based on estimates of the turbulence level in the atmospheric boundary layer, and which use these estimates to calculate the blade loading and then the resulting noise, always give results of the wrong order of magnitude. A model of this type at FFA [6] is for example of no practical use.

In the semiempirical formula given above, for the noise due to inflow turbulence, the sound level was adjusted to give a reasonable result for a specified condition for the Näsudden prototype. The question is now whether the formula gives reasonable results for different conditions. It was suggested in [5] that it might be more reasonable that the turbulent velocity variation is proportional to the blade speed instead of being proportional to the wind speed. In order to in a simple manner test this suggestion, the velocity w in the formula above is expressed in two different ways :

$w \sim V$ and $w \sim U$

When the conditions differ from the reference condition, the value of w will depend on which of the expressions above that is used. The idea is now to compare noise measurements, taken when the Näsudden turbine was run with a series of different rotational speeds, with results calculated using the two expressions for w.

The results of the measurements are reported in [7] and an excerpt from this report is shown in Fig 1. The wind was gusty during the measurements with a variation between 6.5 m/s and 12.5 m/s. The wind turbine was operated so as not to produce any net power output during the experiments.

The calculations are performed for two different values of V, 7 m/s and 11 m/s, when using the model where $w \sim V$. When using the model where $w \sim U$ the calculations are performed for just one wind speed as the influence of the wind speed in this case is very small.

The results for the 1/3 octave band at 31.5 Hz are shown in Figures 2 and 3. For a few of the measured results the sound levels actually increases when the rotational speed decreases. This can be explained by the fact that the wind was gusty. In Fig 2 the influence of a variation of the wind speed on the calculated results are shown for the model $w \sim V$. The calculated results for $w \sim U$ are shown in Fig 3. A variation in wind speed does not result in different sound levels in this case.

It seems to be easier to explain the variation in the measurement results if one uses the hypothesis that the turbulent velocity variation is proportional to the wind speed and not to the blade speed.

The A-weighted overall sound pressure level as a function of rotational speed is shown in Fig 4. One can hardly expect a better agreement.

In Figures 5 and 6 are examples of the whole sound spectrum shown for the case when $w \sim V$ is used. It is obvious from the overpredictions for frequencies around 100 Hz that the simple model used for the turbulent inflow noise spectrum gives an incorrect slope of the spectrum.

EFFECTS ON ENERGY PRODUCTION OF DESIGNING A WIND TURBINE TO MINIMIZE NOISE GENERATION

The consequences of selecting different tip speed ratios, (TSR), in horizontal axis wind turbine blade design, were discussed in [8] . Some of the results from that study will be discussed here.

For a given rotor diameter turbine blades were optimized, with respect to power production, for a number of different tip speed ratios. The resulting chord distributions for a two-bladed turbine are shown in Fig 7. The power production was then calculated for off-design tip speed ratios, using both fixed pitch and variable pitch regulation. These results are shown in Figures 8 and 9. In Fig 9 it can be noted that the variation of TSR only slowly affects the power coefficient for the blades designed for low TSR. The blades designed for low TSR will thus produce energy at least as efficient as the blades designed for higher TSR.

For a given vind velocity rotors running at a lower TSR will always generate less noise. The variation in noise generation due to varying wind speeds is shown in Fig 10 for the cases when the rotors are operated at their optimal TSR. The noise generation is expressed in terms of a dimensionless distance. This distance shows how much further away you must move from the turbine in order to experience the same overall noise level when the rotor or the operational conditions are changed.

REFERENCES

[1]	Brooks, T F Hodgson, T H	Trailing Edge Noise Prediction from Measured Surface Pressures. Journal of Sound and Vibration 78, 1981
[2]	Brooks, T F Marcolini, M A	Airfoil Self Noise - Effect of Scale AIAA 8th Aeroacoustic Conference AIAA-83-0785
[3]	Fink, M R	Minimum On-Axis Noise for a Propeller or Helicopter Rotor J Aircraft Vol 15, no 10, 1978
[4]	Goldstein, M E	Aeroacoustics McGraw-Hill 1976
[5]	Glegg, S A L Baxter, S M Glendinning, A G	The Prediction of Broadband Noise from Wind Turbines Journal of Sound and Vibration 118, 1987
[6]	Juvet, P J-DF	Analysis of Sound Produced by Wind Turbines in Turbulent Flow FFA TN 1983-44
[7]	Walter, P-O	Studie av varvtalsvariationens inverkan på bulleremissionen från Näsuddens vindkraftverk, Gotland IFM Akustikbyrån, TM 7.765.01, 1985
[8]	Meijer, S Montgomerie, B Ronsten, G	Consequences of Selecting Different Tip Speed Ratios in Horizontal Axis Wind Turbine Blade Design FFAP-176, 1987

MASKIN- OCH DRIFTSTILLSTÅND (D)	MÄTPUNKTSBESKRIVNING (P)
Aggregatet är ej infasad på nät. D. 25,0 rpm E. 22,5 rpm F. 20,0 rpm G. 17,5 rpm H. 15,0 rpm I. 12,5 rpm	1. $1/2R_o$

SYMBOL	D	P	dB (A)	ANMÄRKNING
———	D	1	68,4	
– – –	E	1	67,2	
• • • • •	F	1	64,5	
—·—	G	1	60,2	
— — —	H	1	57,6	
– – – —	I	1	53,0	

A 42
A 43

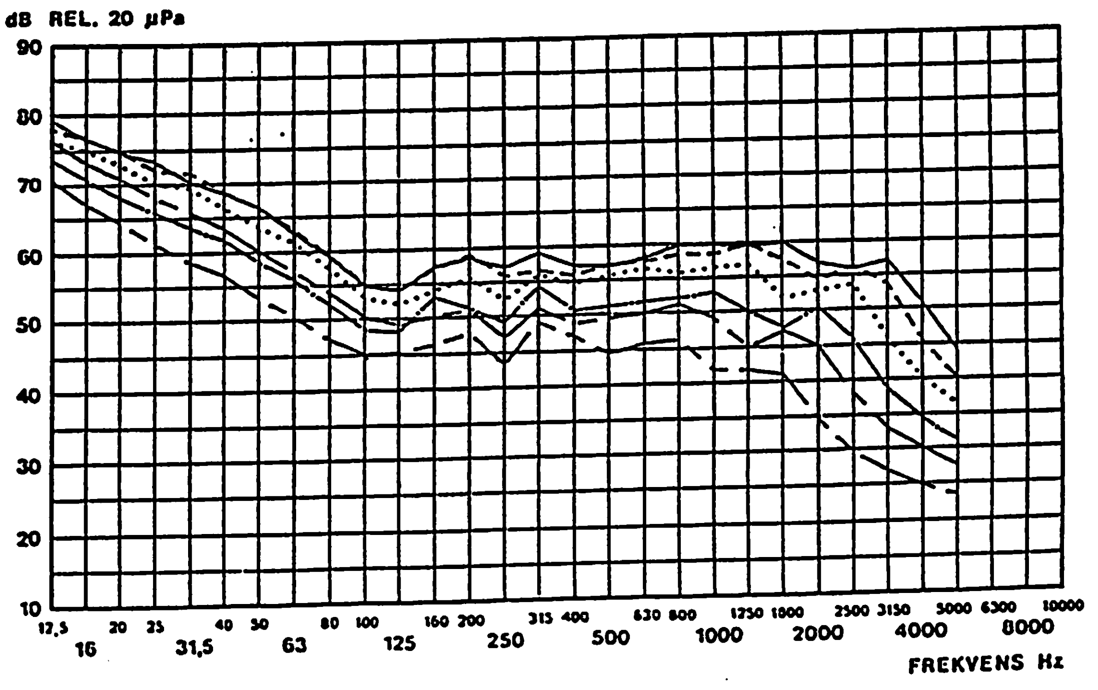


Fig. 1 Measurement results from Näsudden.

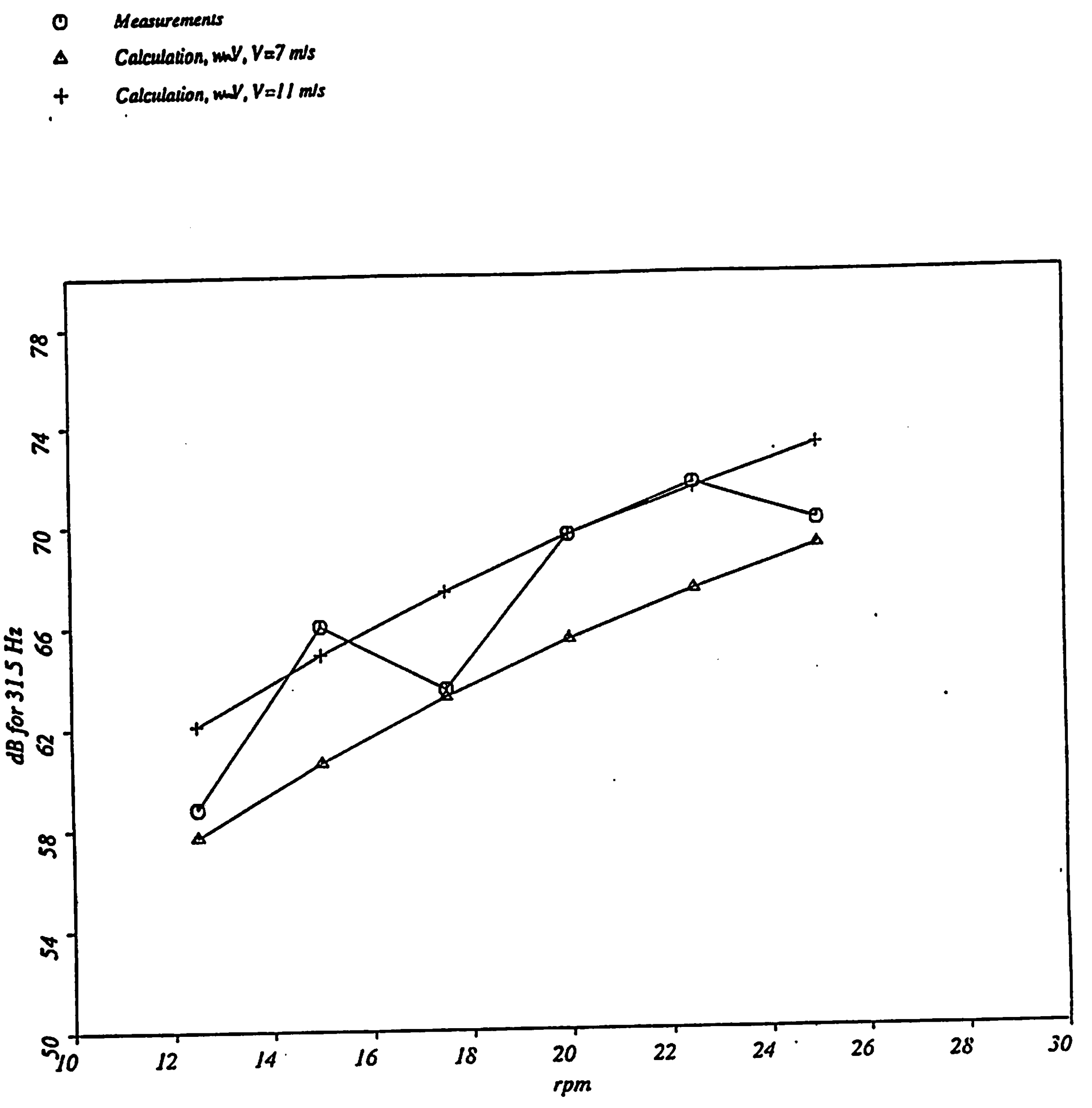


Fig. 2 Sound pressure level in the 1/3 octave band centered at 31.5 Hz

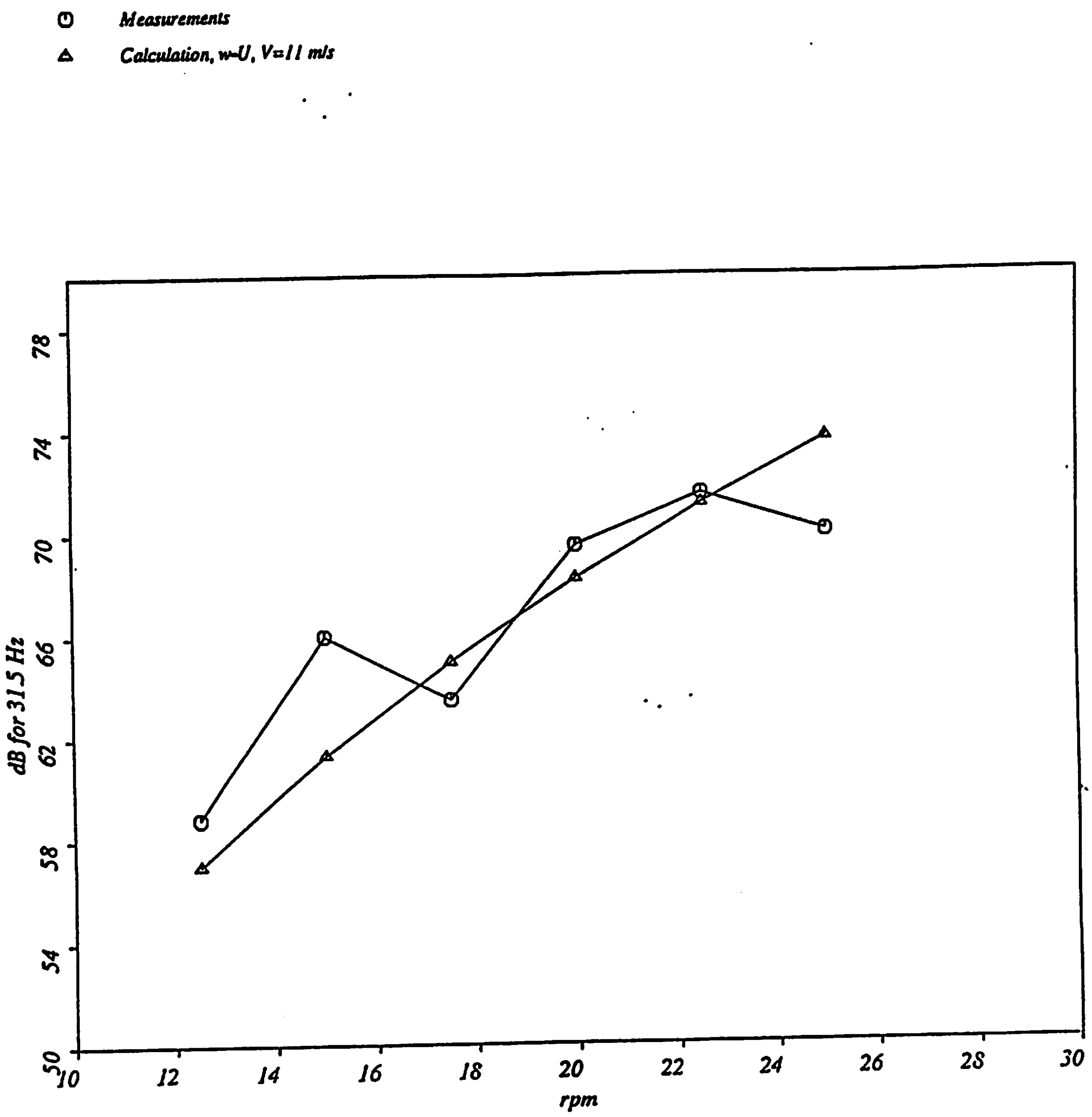


Fig. 3 Sound pressure level in the 1/3 octave band centered at 31.5 Hz

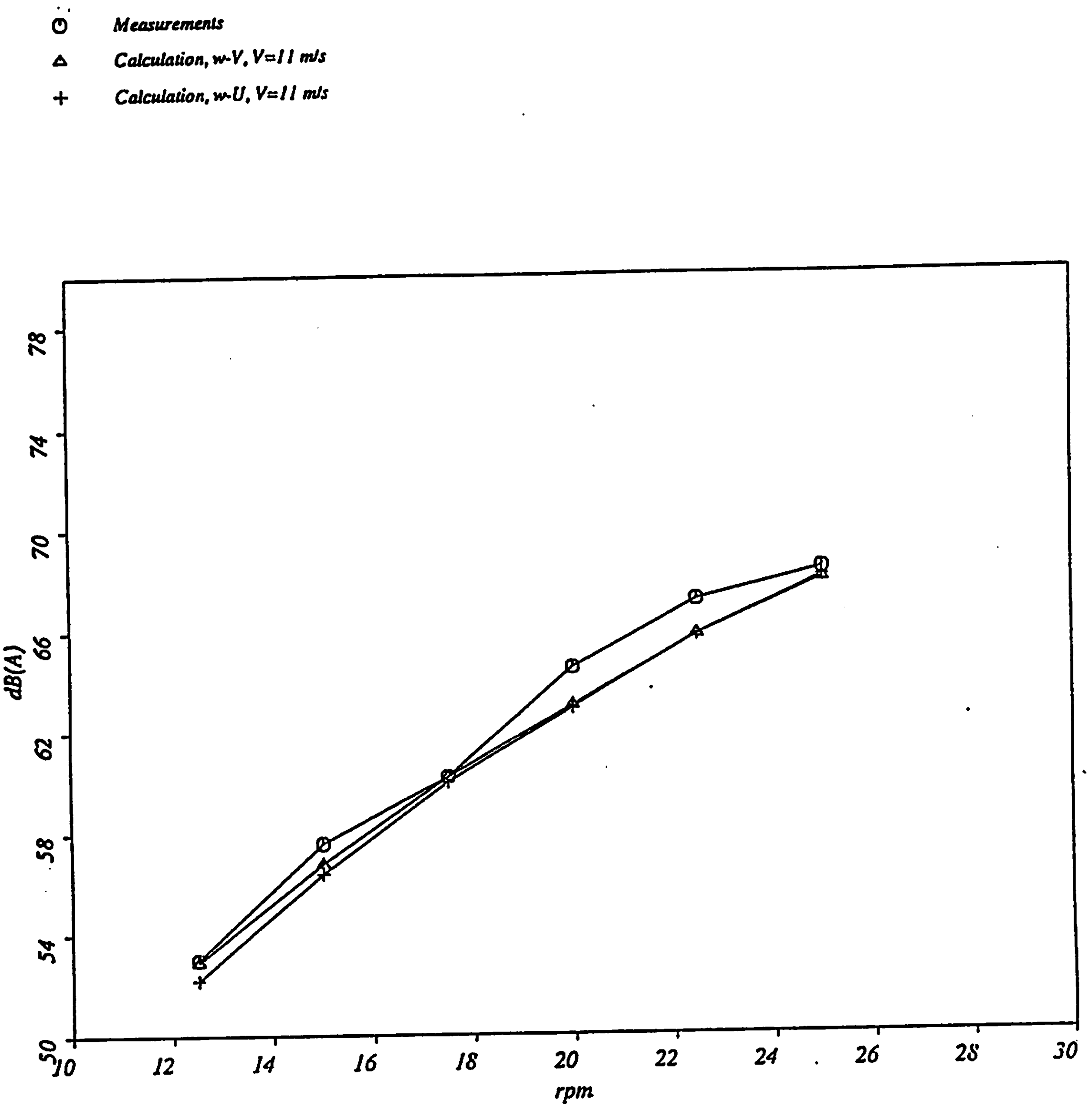


Fig. 4 A-weighted overall sound pressure level.

Sound pressure level in 1/3 octave band

RPM = 25.0
Windspeed = 11.0 m/s
Distance to observer = 0.06 km

Sound Power Level = 115.5 dB(A)

Freq. > 10 Hz	Infrasound
dB(lin)= 86.3 dB(A) = 68.0 dB(B) = 71.4 dB(C) = 79.5	dB(G1) = 91.2 dB(G2) = 89.3

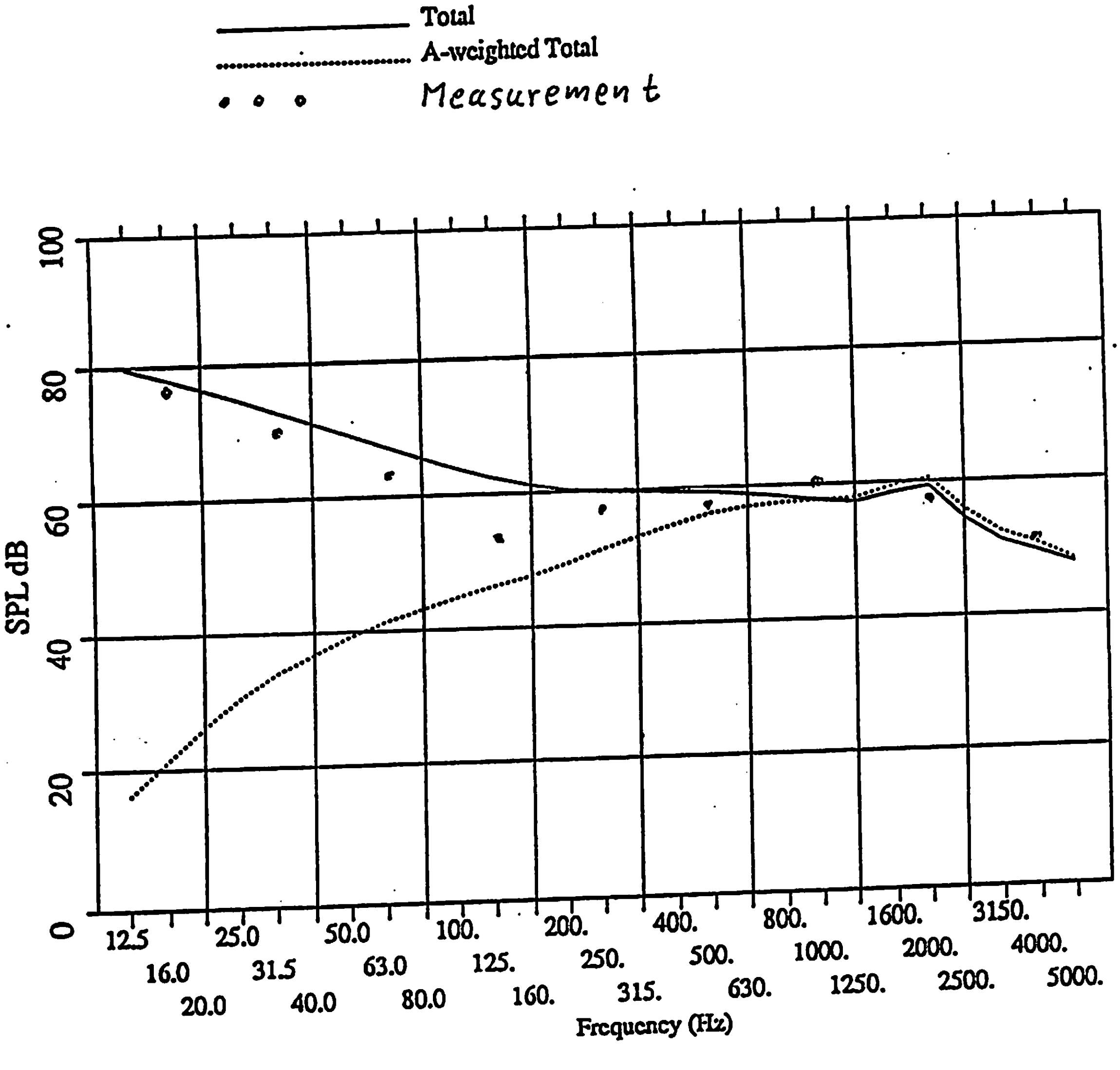


Fig. 5

Sound pressure level in 1/3 octave band

RPM = 12.5
Windspeed = 11.0 m/s
Distance to observer = 0.06 km

Sound Power Level =100.4 dB(A)

Freq. > 10 Hz	Infrasound
dB(lin)= 75.8 dB(A) = 52.9 dB(B) = 59.3 dB(C) = 68.6	dB(G1) = 80.9 dB(G2) = 79.7

Total
A-weighted Total
Measurement

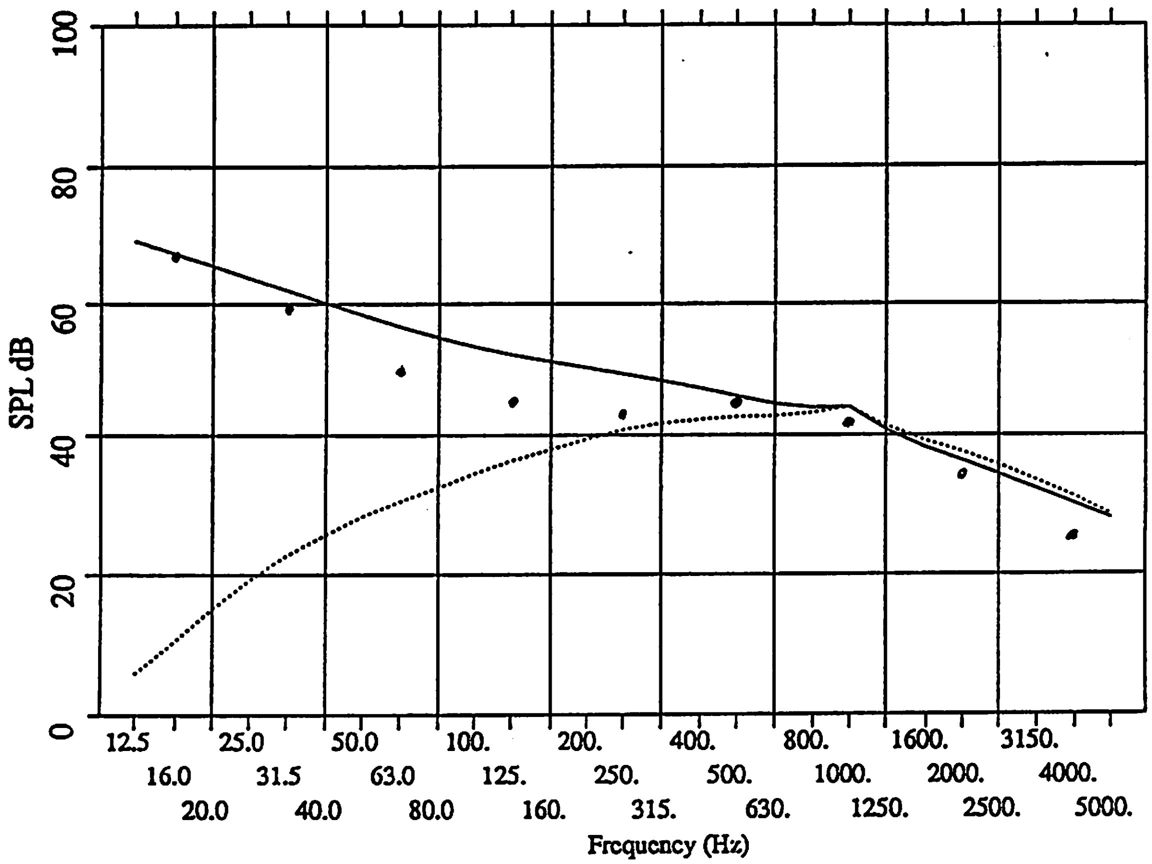


Fig. 6

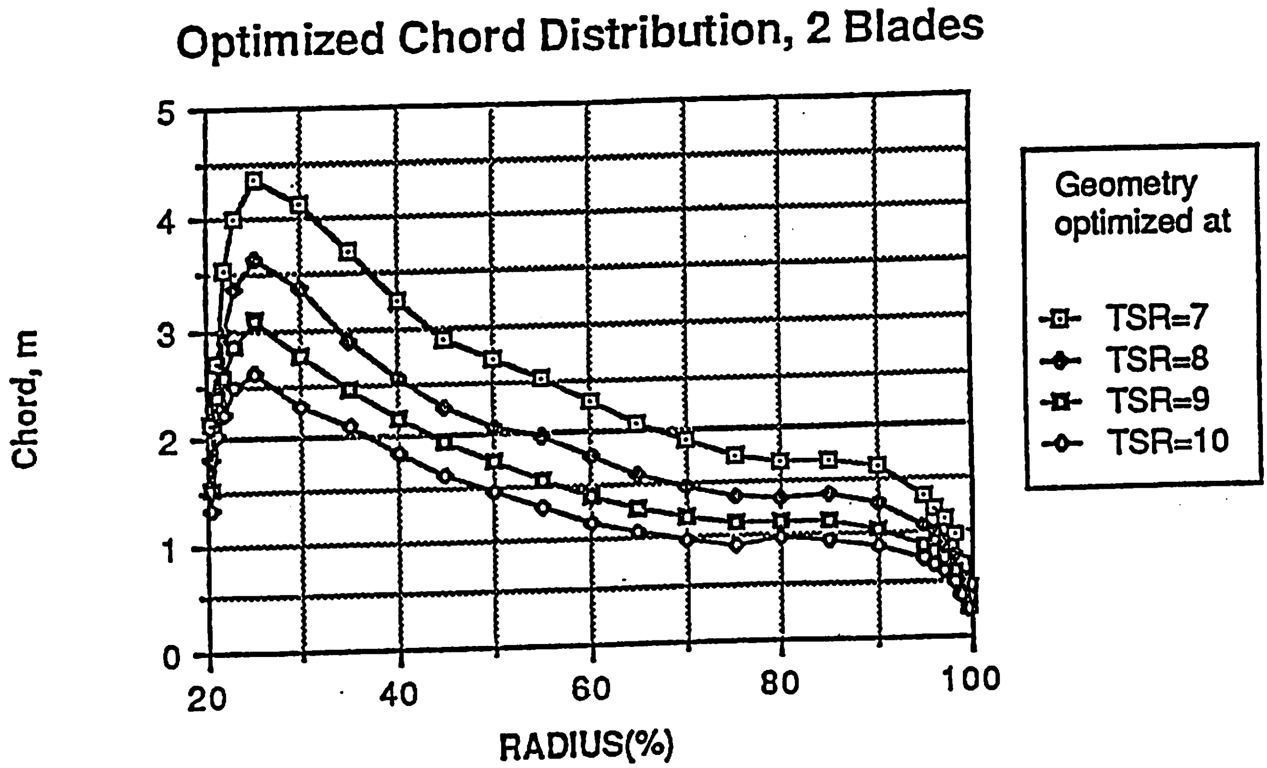

Optimized Chord Distribution, 2 Blades
Chord, m
5
4
3
2
1
0
20
40
60
80
100
RADIUS(%)
Geometry optimized at
TSR=7
TSR=8
TSR=9
TSR=10

Fig. 7

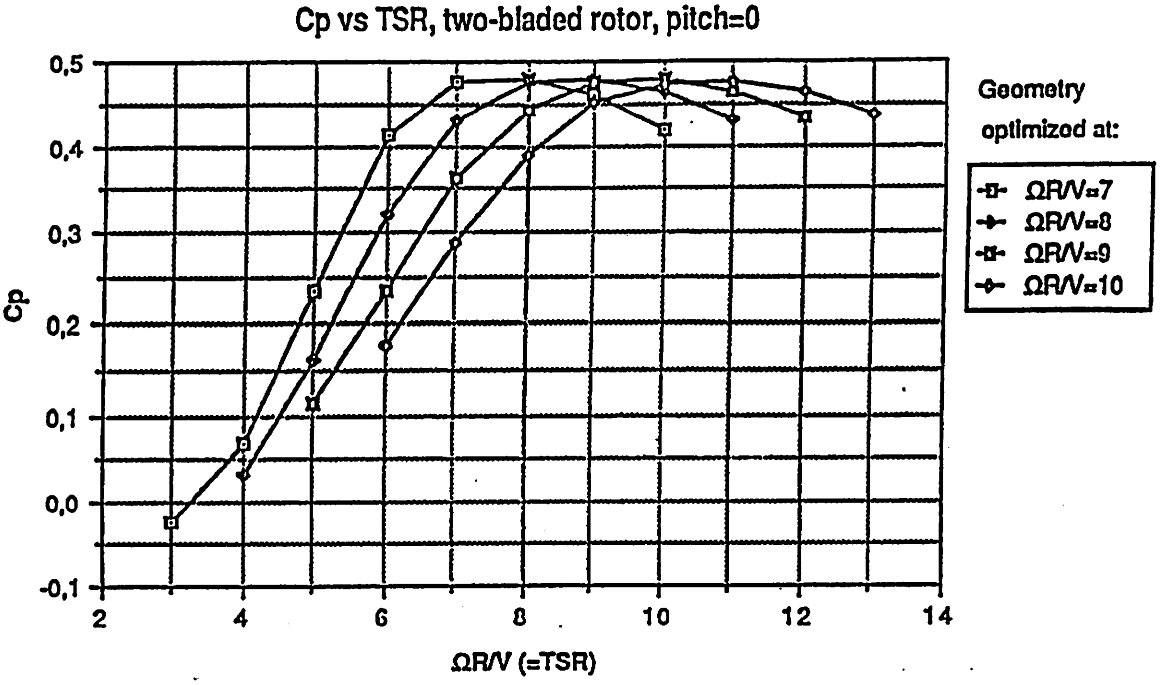

Cp vs TSR, two-bladed rotor, pitch=0
0,5
0,4
0,3
0,2
0,1
0,0
-0,1
Cp
2
4
6
8
10
12
14
ΩR/V (=TSR)
Geometry
optimized at:
ΩR/V=7
ΩR/V=8
ΩR/V=9
ΩR/V=10

Fig. 8

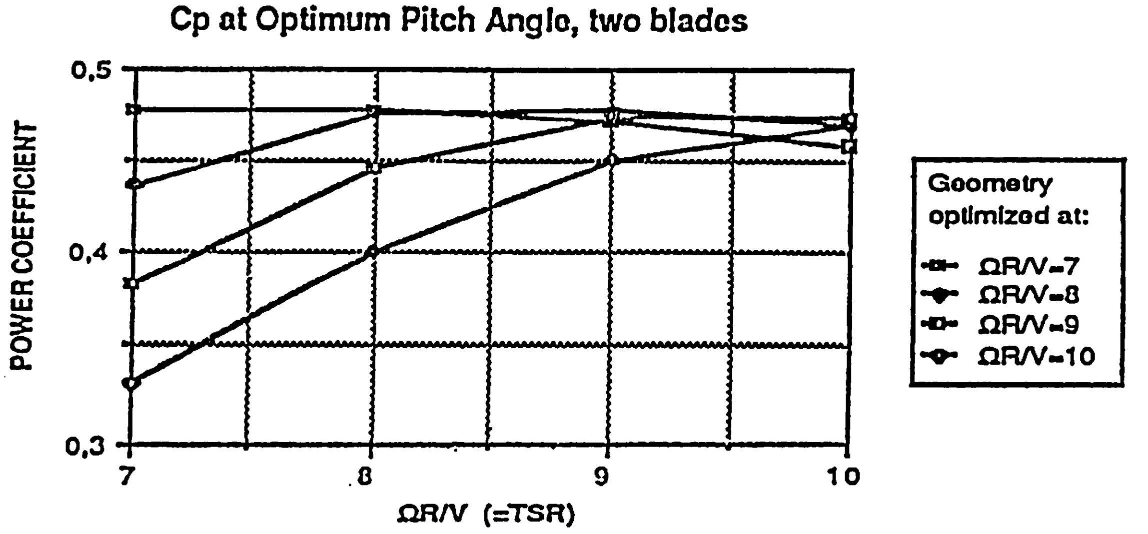
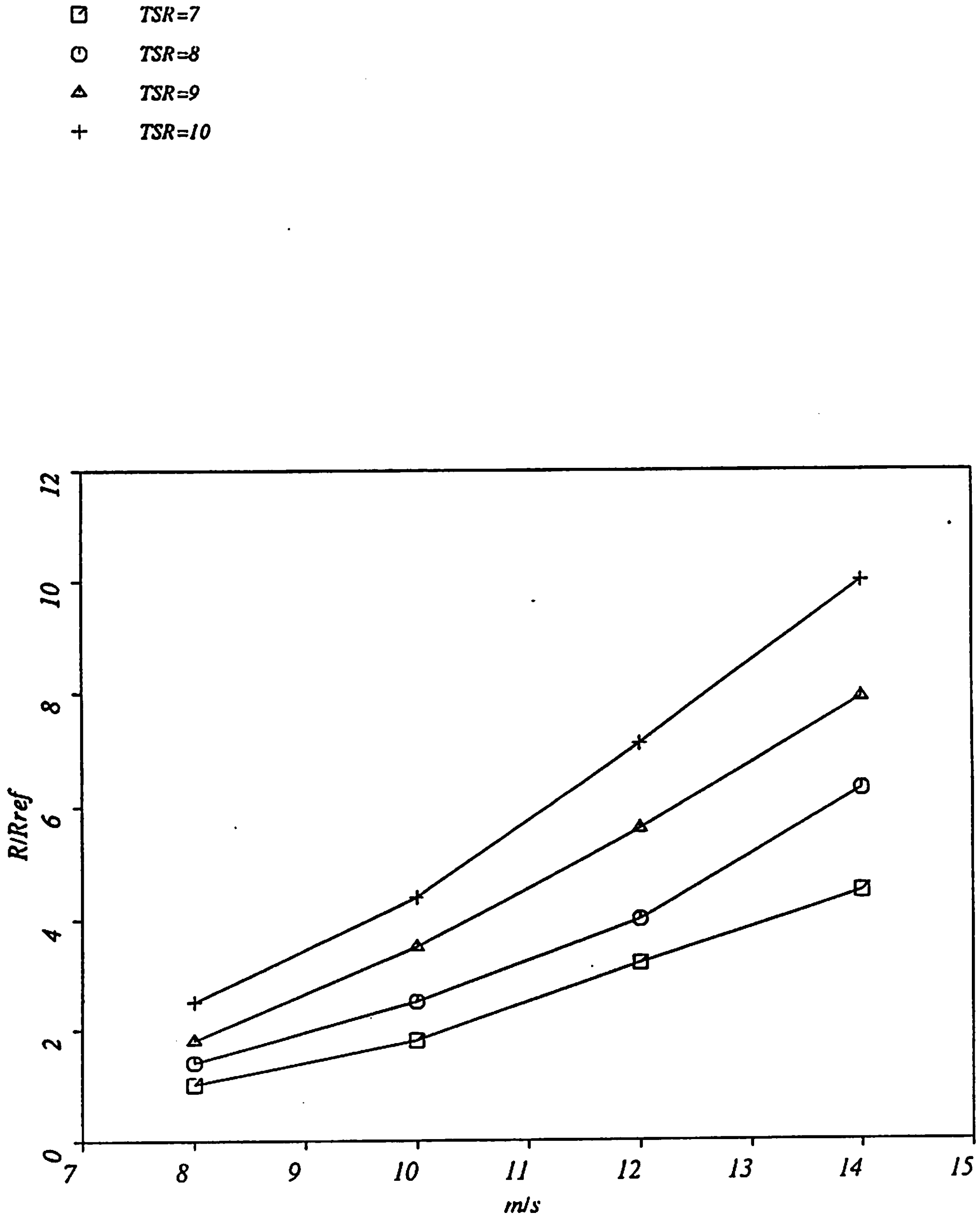

Cp at Optimum Pitch Angle, two blades
0,5
0,4
0,3
POWER COEFFICIENT
7
8
9
10
ΩR/V (=TSR)
Geometry
optimized at:
ΩR/V=7
ΩR/V=8
ΩR/V=9
ΩR/V=10

Fig. 9

Fig. 10 Noise generation as a function of wind speed for a 2-bladed turbine running with variable rpm.



ON THE PREDICTION OF AERODYNAMIC NOISE FROM WECS

H. Norstrud

Introduction

Wind energy conversion systems (or WECS) are increasingly being considered as alternative energy sources in connection with the worlds major use of fosile fuel power plants. The extraction of power from the wind represents an everlasting and seemingly pollution free method to produce the increasing demand for electrical power in modern society.

However, the unwanted production of noise from operating wind turbines (in sole or in multiple operation such as in farms) is a point of concern for the turbine operator since it affects the surrounding community. In order to asses the impact of the noise problem from WECS, the present note attempts to give a simplified analysis for the prediction of aerodynamic generated noise from such devices.

1. Analysis

Let us consider a horizontal axis propeller turbine (see figure 1) and apply the momentum theorem (Betz analysis) to relate the wind velocity u_∞ to the velocity u through the turbine plane, i.e.

$$u = \frac{2}{3} u_\infty \tag{1}$$

The induced velocity behind the turbine plane can to some extend be compared to the flowfield behind a subsonic jet (see figure 2) for which experimental data exists for the radiated acoustic power, see figure 3. The empirical relation between the acoustic power $P\left[W\right]$ and the flow data reads as follows

$$P = 10^{-4}\{0.5 \, \rho_\infty \, U^8 \, A/c_0^5\} \tag{2}$$

where $\rho_\infty\left[kg/m^3\right]$ – density of ambient air

[1]Paper presented at the IEA expert meeting on noise generating mechanisms of wind turbines at Petten, the Netherlands on November 27—28, 1989.

$U[m/s]$ – jet velocity at outlet

$A[m^2]$ – jet crossection at outlet ($= \pi D^2/4$)

$c_0[m/s]$ – speed of sound in ambient air.

Equation (2) is taken from reference [1] and can be transformed to the sound power level $L_P[dB]$ with reference to the power $P_0 = 10^{-12}$ W (or 1 picowatt) as

$$L_P = 10 \log \frac{P}{P_0}$$

$$= 10 \log P + 120 \quad (3)$$

Furthermore, if we assume a free acoustic radiated field equation (3) can be expressed as a sound pressure level $L_p[dB]$

$$L_p = L_P - 20 \log r - 10.9 \quad (4)$$

where $r[m]$ depicts the radial distance from the acoustic source to the field point for which level L_p is valid. A combination of equations (3) and (4) will finally yield

$$L_p = 10 \log P - 20 \log r + 109.1 \quad (5)$$

and this relation together with equation (2) will be utilized to predict the sound pressure level at wind speed $u_\infty = \frac{3}{2} u = \frac{3}{2} U$, see equation (1).

2. Results

Norway is well suited for wind turbine applications (see figure 4) and has historically been connected to the operation of such devices. Figure 5 shows an example of this in the polar region where community noise definitely is no problem.

In order to assess the validation of equation (5) some preliminary acoustics measurements were taken from the three–bladed, horizontal axis turbine at Titran, Norway (see figure 4 and 6). This 400 kW turbine will at a registered wind speed of $u = U = 14$ m/s (corresponding to $u_\infty \approx 21$ m/s) yield the acoustic power of

$$P_{Titran} = 1.9128 \cdot 10^{-5}\,W$$

or $L_P = 72.817$ dB. Here equation (2) has been utilized together with the sound speed relation

$$c_0 = (\kappa\, RT_0)^{1/2}$$

where the ratio of specific heat for air $\kappa = 1.4$, the air gas constant $R = 287\ J\ kg^{-1}\ K^{-1}$ and the measured ambient air temperature were $T_0 = 286$ K (i.e. $t_0 \approx 13^0$ C). The density of air at sea level were set to $\rho_\infty = 1.22\ kg/m^3$.

Reference [3] has given the empirical relation

$$L_P\ [dB(A)] = 10 \log D + 50 \log V_t - 4$$

for the aerodynamic acoustic source power where V_t [m/s] is the turbine tip speed. Since the wind turbine at Titran operates at a constant rotational speed of n = 38.2 r/min the above relation will in comparison yield the value $L_P = 103.5$ dB(A) where $V_t = n\pi D/60 = 69.6$ m/s.

Six measurements were taken with a Brüel & Kjær sound pressure level (Type 2203) with a windscreen (Type UA 0082) and the following results were obtained:

Measurement point	Radial distance r[m]	Measured level L_p, dB(A)	Predicted level L_p, dB
1	32.0	77	31.8
2	40.4	66	29.8
3	58.8	79	26.5
4	58.8	40	26.5
5	40.4	59	29.8
6	58.8	46	26.5

No adjustment with respect to the directivity (angle θ in figure 2) of the noise has at present been inplemented in the prediction.

The background noise at a different, but a similar place 2 km away from the wind turbine site were measured to 40 dB(A).

3. Conclusions

Preliminary results from acoustic noise measurement from a WECS has been reported together with a simple method of prediction of the sound pressure level. As the results shows, the predicted aerodynamic noise level L_p [dB(A)]. A more detailed analysis and improved measurements will be reported at a later time.

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[1] Howes, W.L., "Similarity of far noise fields of jets", NASA TR R–52, 1959.

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[3] de Wolf, W.B., "Aerodynamisch geluid van windturbines", NLR–MP87004U, January 1987.

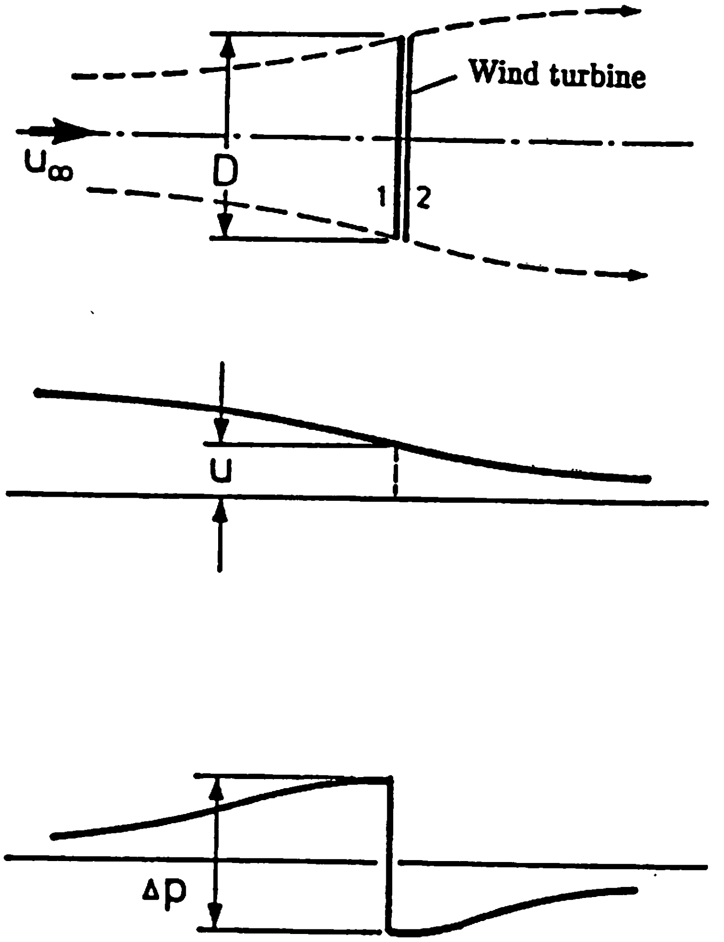

FIGURE 1 – Velocity and pressure distribution through a horizontal axis wind turbine

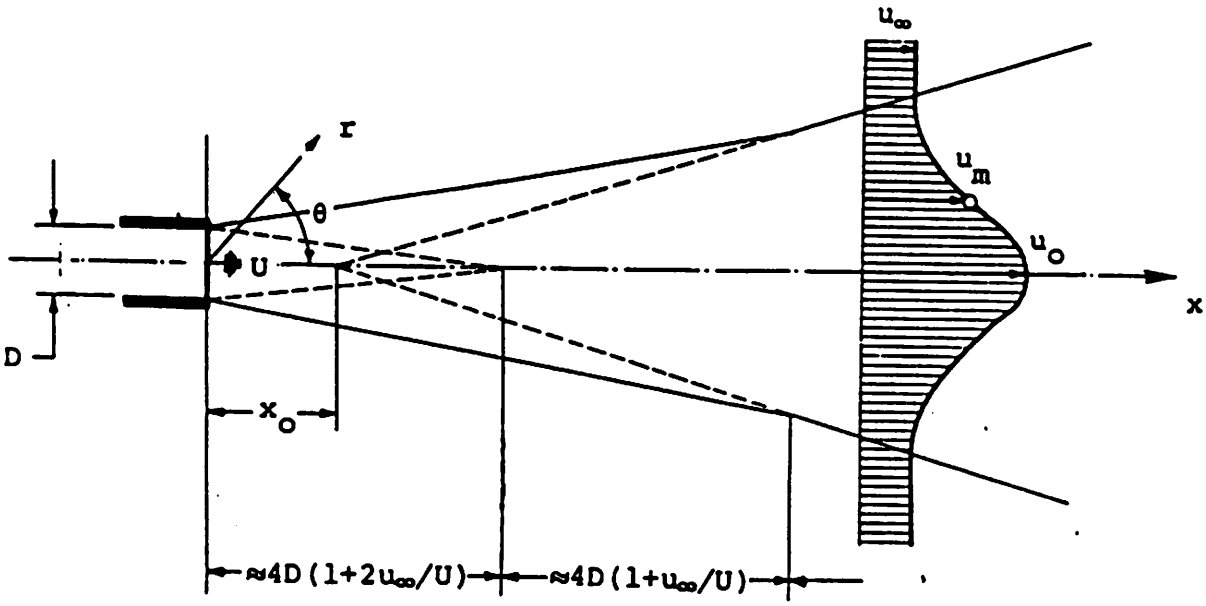

FIGURE 2 – Subsonic jet in a uniform flowfield

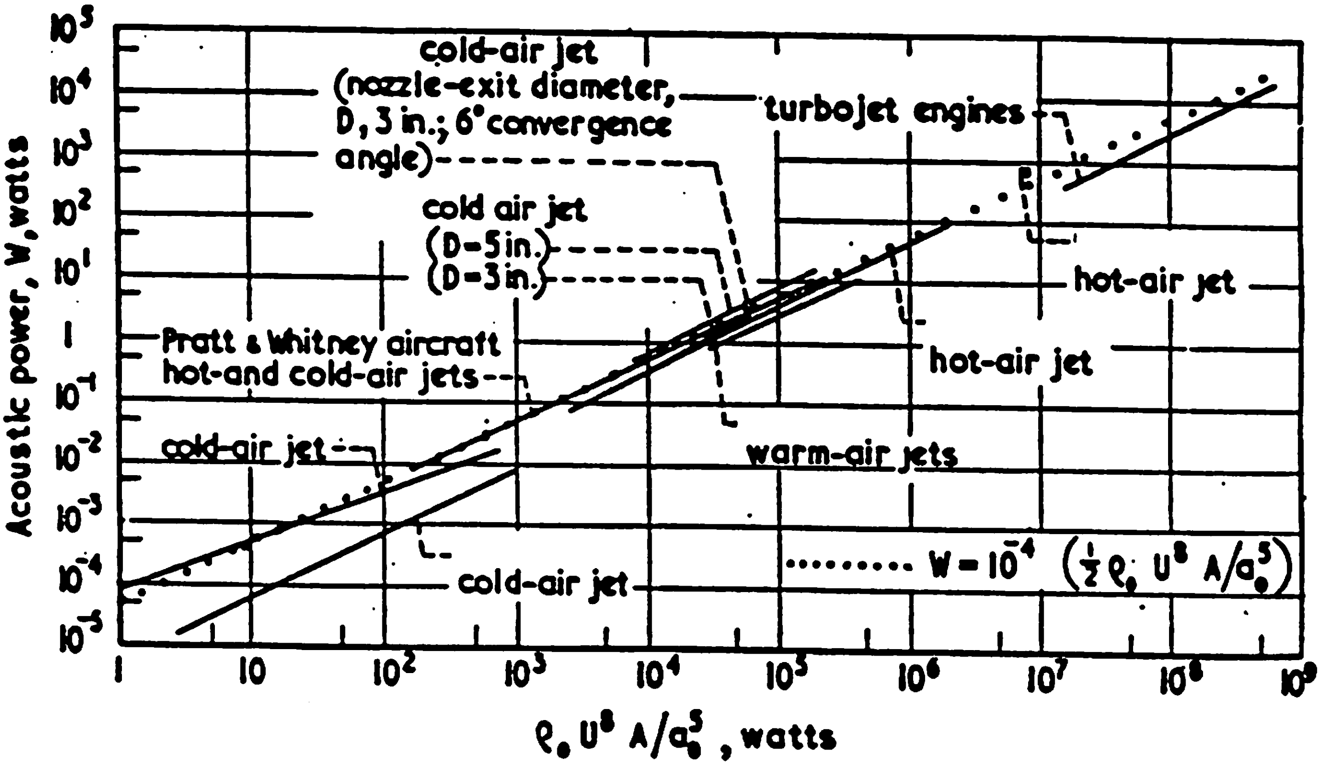

FIGURE 3 – Acoustic power radiated by jets

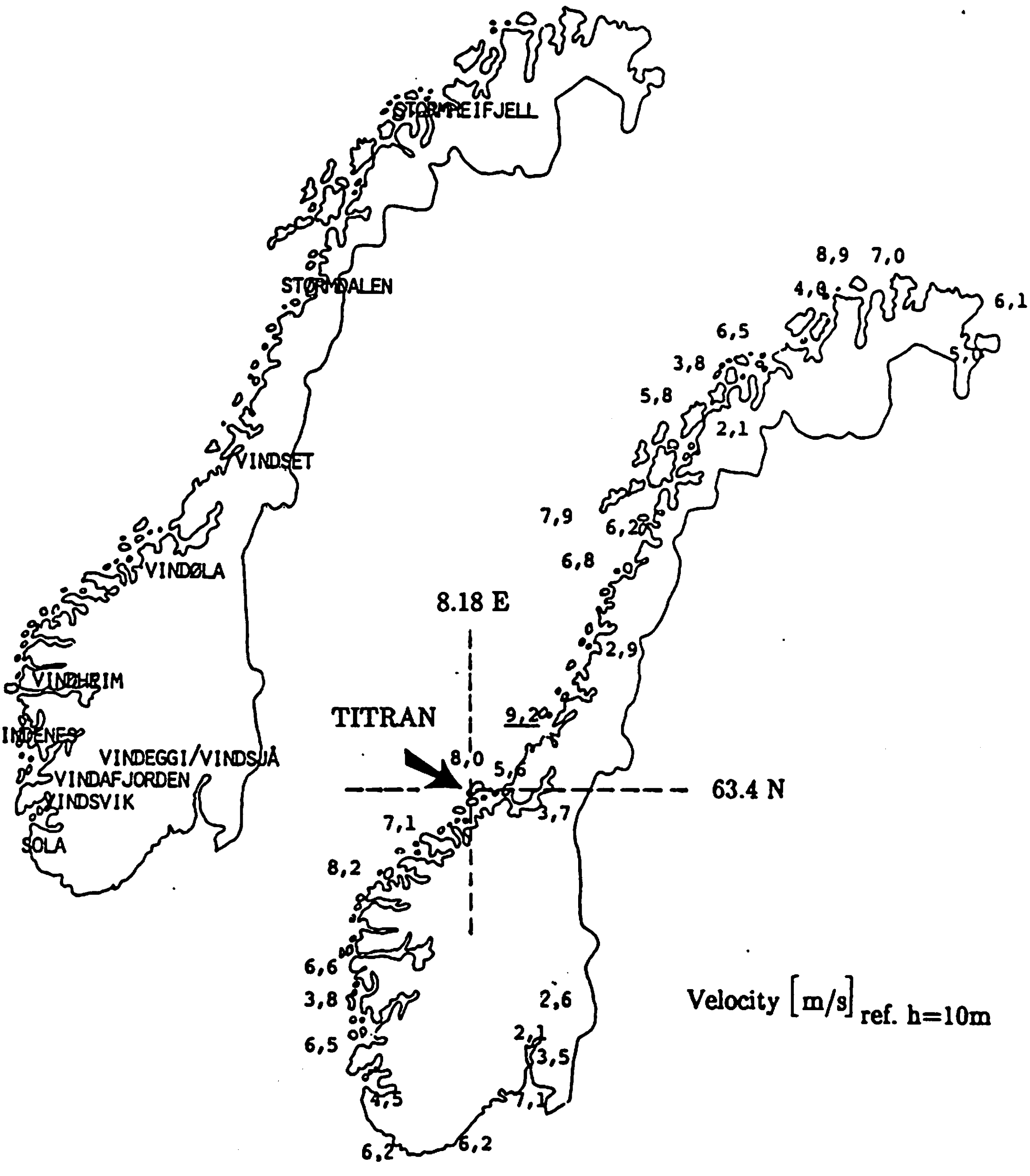

FIGURE 4 — Wind conditions in Norway in names and numbers

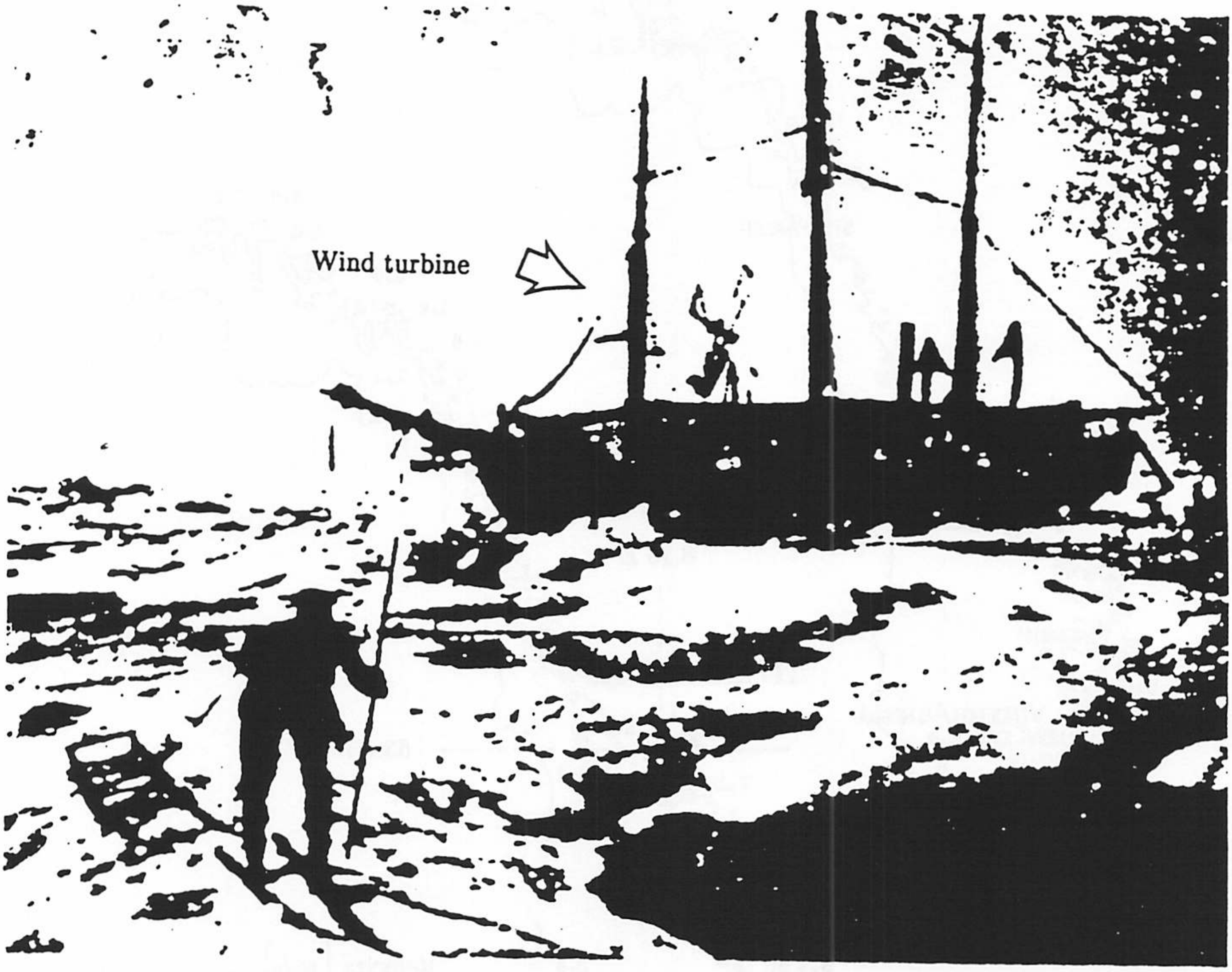

FIGURE 5 – The polar vessel "Fram" as used by:
i) Fridtjof Nansen (Arctic Ocean), 1893–1896
ii) Otto Sverdrup (North America), 1897–1902
iii) Roald Amundsen (South Pole), 1910–1912

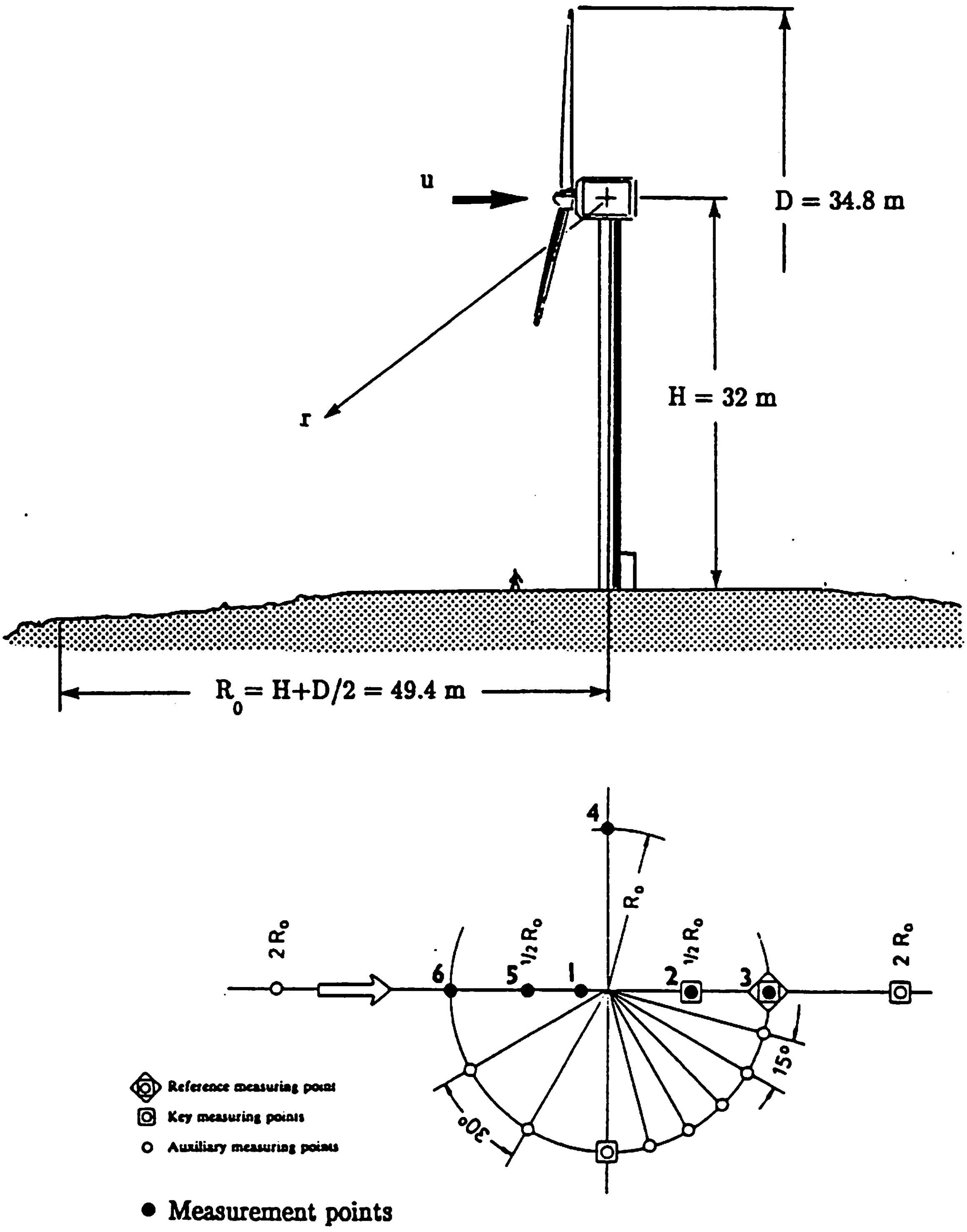

FIGURE 6 — The Titran wind turbine "ISAK"

MODELLING WIND TURBINE NOISE

A. Pfeiffer

INTRODUCTION

Noise generated by wind turbines is becoming more and more a topic of importance. Especially in countries like the Netherlands, where the available area for solitary placed wind turbines and wind power plants is limited. In order to make optimal use of the available area the sound power level of wind turbines should be as low as possible.

At the end of 1986 the Dutch government announced a program which has the intention to realize 1000 MW of wind turbine power in the Netherlands before the year 2000. In order to make this possible the program is supported by a research program. Within this research program the research in the field of noise reduction of wind turbines is an important item.

Research on wind turbine noise can be subdivided into several parts, namely:

1. Fundamental research, noise models
2. Research in the field of permits
3. Measuring methods
4. Cooperation IEA expertgroup acoustics
5. Measuring sound power levels for certification
6. Development of designers tools.

The paper is mainly dealing with the work that has been going on since 1987 in the field of developing tools for the designers and manufacturers of windturbines in order to accomplish a new generation of low noise wind turbines.

DESIGN TOOLS

From the beginning of the research in the field of wind turbine noise it was clear that besides the research it was very important to translate the gained knowledge into information that can be simply used by non acoustic experts. Only in this way the noise problems can be tackled for a lot of wind turbines. The translation concerns mainly the simplification of models developed through fundamental research in combination with the development of common useable measures and evaluation methods of both technical and economic nature.

One of the first reports in the Netherlands specially written for the designers of wind turbines was the "brochure wind turbine noise" (1). This brochure gives a review of the knowledge in the field of wind turbine noise. The brochure contains information about aerodynamic- and mechanical noise (figure 1), a model to predict the sound power level of a wind turbine and recommendations in order to reduce the generated noise.

The brochure was followed by other publications. Figure 2 gives a review of them. At the moment the "construction manual wind turbine noise" (2) together with a computer code, called TURBNOISE, are the main instruments meant for the designers of wind turbines in order to develop, with rather simple and common understandable tools, a silent wind turbine.

NOISE PREDICTION MODEL

With the noise prediction model the sound power level of a wind turbine can be calculated by means of the determination of the sub sources mentioned in figure 1. Before going into details a few definitions are given:

Definitions

Aerodynamic noise Noise generated by the rotor of the wind turbine as a result of the interaction between the air and the blade surface. Aerodynamic noise is caused by several phenomena; atmospheric turbulence, irregularities in the blade geometry and turbulence caused by blade rotation. The latest can be subdivided in turbulence in the boundary layer, turbulence around the blade tip (tip-vortex noise) and turbulence in the separated flow (trailing edge noise). De Wolf (3) gives a detailed description of aerodynamic noise.

Mechanical noise Noise generated by devices that are situated in the nacelle of a wind turbine, mainly the gear box and generator.

Contact noise Noise resulting from the transport of energy by means of mechanical vibration. For example a vibrating gear box induces vibrations in the nacelle walls because they are both attached to the main frame in the nacelle. Due to the vibrations present in the nacelle walls second order air noise is generated.

Symbols

A	:	sound absorption in nacelle	$[m^2]$
D	:	diameter wind turbine rotor	$[m]$
$L_w(tot)$	:	predicted s.p.l. wind turbine	$[dB(A)]$
$L_w(G)$	:	s.p.l. generator	$[dB(A)]$
$L_w(GB)$	:	s.p.l. gear box	$[dB(A)]$
$L_w(GB+G)$	:	combined s.p.l. generator, gear box	$[dB(A)]$
$L_w(1)$	:	s.p.l. air noise through openings	$[dB(A)]$
$L_w(2)$	:	s.p.l. air noise nacelle walls	$[dB(A)]$
$L_w(3)$	:	s.p.l. contact noise nacelle walls	$[dB(A)]$
$L_w(4)$	:	s.p.l. contact noise by tower	$[dB(A)]$
$L_w(5)$	:	s.p.l. aerodynamic noise	$[dB(A)]$
m_g	:	mass gear box	$[kg]$
m_n	:	mass nacelle, including components	$[kg]$
m_t	:	mass tower	$[kg]$
R_{250}	:	air noise isolation factor at 250 Hz	$[dB(A)]$
S_A	:	surface sound absorption material on nacelle walls	$[m^2]$
S_g	:	surface gear box walls	$[m^2]$
S_n	:	surface nacelle walls	$[m^2]$
S_o	:	surface openings in nacelle walls	$[m^2]$
S_t	:	surface tower (minimal 15 m^2)	$[m^2]$
V_t	:	tip speed	$[m/s]$
α_{500}	:	sound absorption coefficient at 500 Hz	$[-]$

explanation: s.p.l. : sound power level

Model

The noise prediction model is meant for wind turbines in the range from 50 to 500 kW. The formulas are based on measurements (aerodynamic noise and contact noise) in combination with descriptions given in literature for other fields of technique. The model consists of the following formulas:

$$L_W(1) = L_W(GB + G) + 10 \log (S_o/(A + S_o)) \qquad (1)$$

Sound power levels of gear box and generator can be obtained from graphs given in literature (4,5) where they are presented as function of the nominal power of the component or from component manufacturers.

$$A = \alpha_{500} \cdot S_A + c_1 (S_n - S_A) \qquad (2)$$

For air noise 500 Hz is the frequencie with the biggest impact on the sound power level, therefore the α_{500} value has to be taken.

$$L_W(2) = L_W(GB + G) + 10 \log (S_n/(A + S_o)) - R_{250} \qquad (3)$$

By air noise through openings 250 Hz is the most important frequencie, therefore the R_{250} value has to be taken.

$$L_W(3) = L_W(GB) + 10 \log (n_g/n_n) + 10 \log (S_n/S_c) - c_2 \qquad (4)$$

In case of contact noise only the sound power level of the gear box is taken into account because normally this is the biggest source for vibration in the nacelle. Assumed is a relation between the sound power level and the vibration level. The behaviour of the construction is translated in a factor of c_2.

$$L_W(4) = L_W(GB) + 10 \log (n_n/n_t) + 10 \log (S_t) - c_3 \qquad (5)$$

This formula is derived for tube towers. Normally the whole surface of the tower is taken into account except when the tower is flanged, then only the upper part is taken into account up to the highest flange.

$$L_W(5) = 50 . \log (V_t) + 10 \log (D) - c_4 \qquad (6)$$

The constants c_1 to c_4 are dependent on the wind turbine configuration. By means of a logaritmic addition the predicted sound power level of the wind turbines can be derived.

$$L_W(tot) = 10 \log \left[\sum_{i=1}^{5} 10^{0,1 . L_W(i)} \right] \qquad (7)$$

VALIDATION

In order to test the behaviour of the model a validation has been carried out (6). Therefore wind turbine manufacturers in the Netherlands and abroad were asked to supply information of there design(s) in combination with results of noise measurements. In this way the model has been evaluated for in total 16 wind turbines ranging from 50 to 3000 kW installed power. A detailed validation was not possible because there were no measurements available that describe the model completely. However a general impression, presented in figure 3, can be given. The figure shows that there is a reasonable correlation (correlation coefficient 0,85) between the measured value L_W and the predicted value L_W (tot). However deviations exist from +7 dB(A) to -7 dB(A). Although the model predicts that aerodynamic noise is in many cases (10 out of 16) the most important source, in practice mechanical noise is also often a problem. This is, amongst other things, stated by vibration measurements carried out in a few wind turbines during the validation.

CONSTRUCTION MANUAL

An effective approach of wind turbine noise can only be realized when this matter is taken into account during the design phase of a wind turbine. Then the costs will be relatively low and the effect of the noise reduction can be relatively high compared to improvements realized with existing wind turbines. The tools that are needed for the design of a silent wind turbine are:

1. Prediction model sound power level
2. Overview of measures
3. Information of costs of measurements

With these tools a design process can be executed according to figure 4.

Measures noise reduction

In the design manual the measures are presented in the form of graphs. Together with these graphs information is presented about costs, obtainable noise reduction and the feasibility of the measures, see figure 5. In this way altogether 25 measures are presented, varying from the adaption of the tip speed ratio to the realization of isolating vibration.

TURBNOISE

Together with the construction manual a computer code was developed in order to make a quick and easy optimization possible of a design for wind turbine noise. Within the program this section for the calculation of sound power levels is combined with a section for the calculation of sound pressure levels in the surroundings of wind power plants and wind turbines. In figure 6 the results of the latest are presented. Background emission and soil absorption can be taken into account.

FUTURE

It is expected that in the near future the wind turbines in the Netherlands will be more silent. This is due to a growing experience amongst wind turbine manufacturers, the availability of specialized literature and the subsidy that for some time is offered by the Dutch government for silent wind turbines. The intension is to emphasize research in the field of mechanical noise in order to make a more detailed validation and description of the wind turbine noise model possible. There are plans to erect a wind turbine, as a demonstration unit, where special attention has been paid to wind turbine noise in order to investigate the economical and technical possibilities of noise reduction. By means of these efforts wind turbine noise is hopefully becoming a topic of less importance in the near future.

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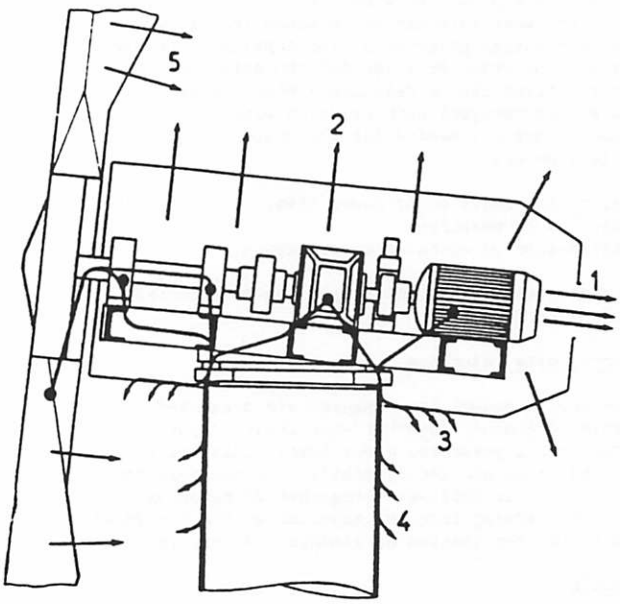


Figure 1 Origine wind turbine noise
●: noise sources
mainly rotor, gear box and generator
1: air noise through openings
2: air noise emitted by nacelle walls
3: contact noise emitted by nacelle walls
4: contact noise, emitted by tower
5: aerodynamic noise

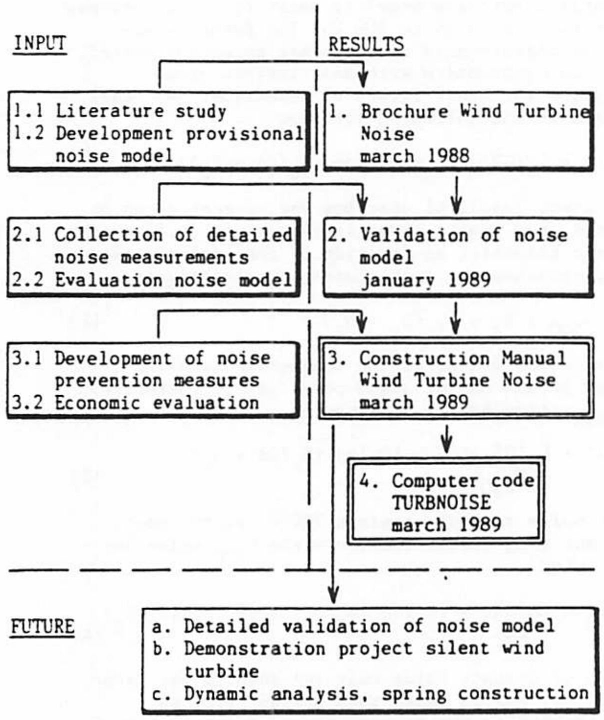


Figure 2 Development of design tools for the reduction of wind turbine noise
explanation: ▭ main tools for wind turbine designers

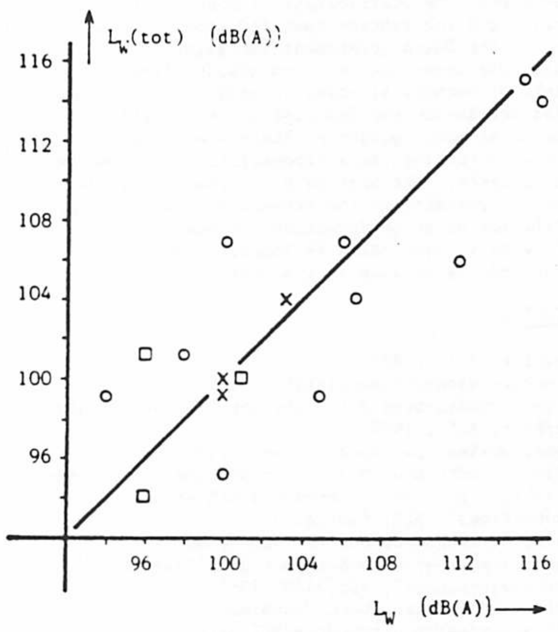


Figure 3 Correlation between measured sound power level L_W and predicted sound power level L_W (tot). The symbols indicate which source is the mean source, according to the model
explanation: o aerodynamic
□ mechanical
× both aerodynamic, mechanical

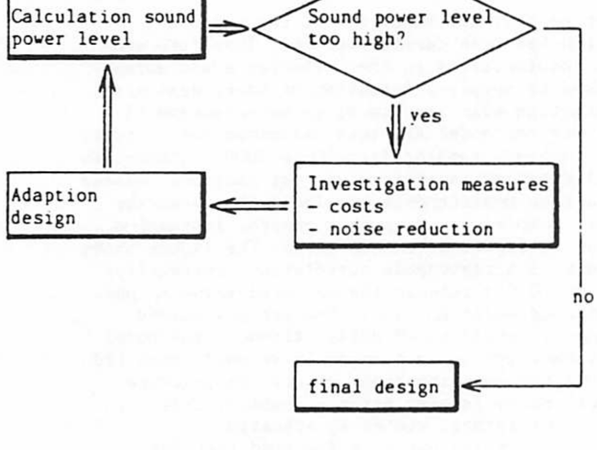
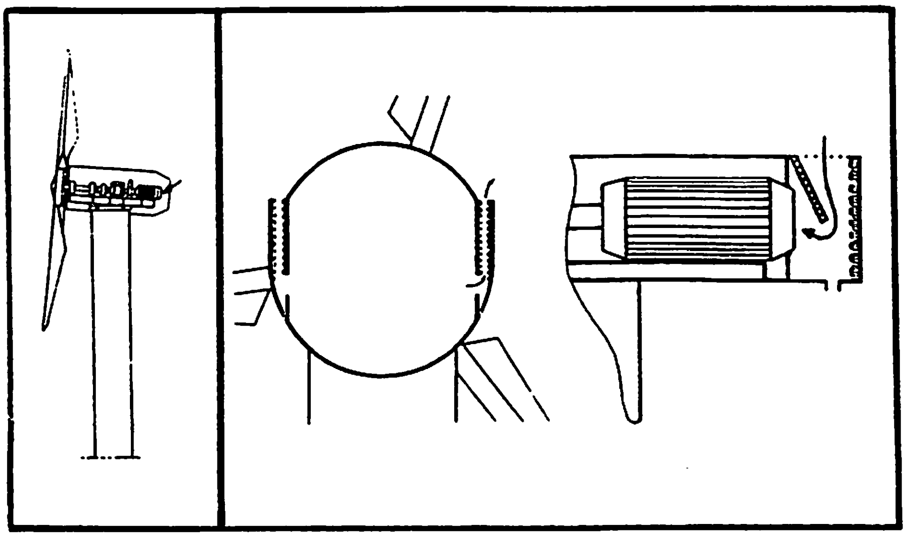


Figure 4 Design process silent wind turbine



Cooling air supply

Reduction sound power level maximum 10 dB(A) on $L_W(1)$: air noise through openings.

Costs approximately dfl 500,-- to dfl 1.500,--.

Description: By means of situating the in- and outlet of the cooling air on the upper side of the nacelle instead of the bottom emission of noise directly to the ground is prevented. It is preferred to attach sound/absorption material on the in- and outlet. A good draining system for rain-water is required.

Figure 5 Example of noise reduction measure

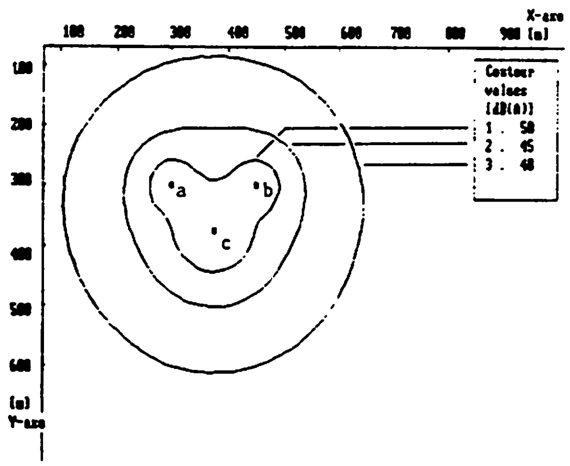


Figure 6 Sound pressure levels around a wind power plant
Conditions: Measuring height 5 m, soft soil
Background noise : 20 dB(A)
Sound power level a,b: 90 dB(A)
Sound power level c : 95 dB(A)

EXPERIENCES DEMONSTRATION PROJECT

In order to investigate the economical and technical possibilities of noise reduction of wind turbines in practice a project has recentely started with the aim to build a silent wind turbine based on the construction manual wind turbine noise. Starting in the design stage of a wind turbine measure are developed and incorporated based on common technology. A 500 kW wind turbine, pitch regulated and with a constant rotor speed is chosen as the basis for the demonstration unit. In the Netherlands wind turbines of 300 to 500 kW with a constant rotor speed are becoming more and more common. At the moment (december 1989) the wind turbine is being erected. A measuring program is expected to be executed in February 1990.

Aims

The demonstration project has the following aims:

1. Validation of mainly the empirical relations for the prediction of the mechanical noise as presented in the construction manual wind turbine noise.
2. Examination of the impact of different measures on the sound power level of the wind turbines. Most of the measures can therefore be bridged in the wind turbines.
3. Evaluation of the subsidy on silent wind turbine. The cost and the effect of the different measures will be evaluated. Questions as can measures be easy combined with other requirements of the design and on which sound power level in relation to the rotor diameter should the subsidy be based at the moment and in the future, are subject of the evaluation.
4. Creation of an example of a large silent wind turbine based on common technology.

Measures

As result of the constant rotor speed concept it is expected that the aerodynamic noise will cause the most hindrance. Therefore one of the main measures was a rotor speed reduction from 42 rpm to 38 rpm. This reduction will cause almost no loss in energy production. On the other hand a bigger gear box is required which causes an increase in the price of the gear box of approximately 10%.
In order to prevent the occurence of pure tones and contact noise the following measures have been taken:

1. Integrated drive train; generator and rotor are attached to gear box. Only the gear box is attached to the nacelle frame.
2. The gear box and thereby the whole drive train is vibration isolated by means of a cork composite from the nacelle.
3. The gear box is a high quality version (planetary, welded round housing, stiffening ribs and fine grinding of wheels).

Because of the great dimensions going with a 500 kW wind turbine is hardly possible to realize a low eigen frequency vibration isolation by means of rubber. Therefore the cork composite has been choosen. After erection some other measures will be taken:

1. Sound silencers placed in the ventilation openings.
2. Isolation of nacelle walls.
3. Closing of gaps and narrow openings.

Expectations

The following value of the different sound power levels are expected based on a wind speed of around 8 m/s:

item	sound power level dB(A)
wind turbine before measures	104
wind turbine after measures	101
gear box	93
generator	94
subsidy level	103

The greatest impact is expected from the rotor speed reduction because the aerodynamic noise will be probably the biggest sound source. Predictions are however difficult to make. Not only because the used models are empirical and simple but also because it is uptil now impossible to predict the influence of pure tones.

MEASURING SOUND PRESSURE LEVELS IN A HARD PLATE ON THE GROUND

J. van der Toorn

1 MICROPHONE IN A HARD SURFACE

IEA document 'Measurement of noise emission from wind turbines' [1] recommends deriving acoustic source strengths of wind turbines from sound pressure levels measured with a microphone in or on a hard surface on the ground.

The method is supposed to have advantages over measuring on a height of 1,5 m or 5 m above the ground:

- The ground effect is normalized on 6 dB (the sound pressure levels on an acoustically hard surface are 6 dB higher than in the free field at the same distance).
  When measurements are performed at a height of for example 1,5 m, the effect of ground reflections is a function of frequency. It depends on the heights of the wind turbine and the microphone, on the distance between them and on the acoustical impedance of the ground [2].
  The impedance of the ground varies with place and is hard to determine.
- Wind induced noise on the microphone is minimal.
  Because wind speed is minimal at ground level, less often special techniques to suppress wind noise [3] are needed when the microphone is placed at ground level and measurements are possible at higher nominal wind speeds.

- Measurements are less sensitive to ambient sound of low sound sources such as road traffic. Damping due to the ground effect [2] for such sources becomes more effective when the microphone height is decreased.

Measuring with a microphone in a hard plate on the ground also has disadvantages:
- A ground plate is not easy to handle.
- The measuring direction is slightly less relevant with respect to remote observers. The severity of this drawback depends on the (unknown) directivity of the windturbine in the vertical plane and on the distance between the relevant observation points and the wind turbine.

Although the method is very promissing, only a few measurement results were shown to support the method [1]. Reproducibility and repeatability of measurements probably depend on the situation. Therefore we did some additional measurements before bringing this method into use.

2 ADDITIONAL MEASUREMENTS

2.1 Sound source

We performed measurements using a loudspeaker as sound source. The strength of the source was controlled by keeping the current through the coil constant.

The source strength was determined by measuring the sound level $L_1^{4,4m}$ of the loudspeaker in an anechoic room on the axis of the loudspeaker, at a distance of $4,4\sqrt{2}$ m (= 6,2 m).

The room is virtually free of reflections for frequencies above 100 Hz.

2.2 Outdoor measurements with a flush mounted microphone in hard plates

The sound of the loudspeaker was also measured outdoors, with a microphone that was flush mounted near the centre of different rectangular, acoustically hard plates, made of plasticized chipboard with a thickness of 18 mm. The other dimensions of the plates are listed in table 1.

The outdoor measurements were performed on flat, grass covered ground (a sports field). The loudspeaker was lifted to 15 m above the ground with a tower waggon and directed towards the microphone, which was positioned 15 m from the projection of the loudspeaker on the ground (the line 'source-microphone' made an angle of 45° with the normal on the upper surface of the plate; the distance between source and microphone was $15\sqrt{2}$ m). The measured sound level is denoted as L_2^{15m}.
The longer sides of the plates pointed into the direction of the projection of the loudspeaker on the ground.

3 HYPOTHESIS

We tested the hypothesis

$$L_{in\ plate} - L_{free\ field} = 6 \text{ dB} \qquad (1)$$

or

$$L_2^{15m} - L_1^{4,4m} + 20.\lg(15/4,4) + C_{vv} - 6 = 0 \qquad (2)$$

where:

L_2^{15m} = sound pressure level measured in the hard plate in the outdoor experiment [dB re 20 µPa],

$L_1^{4,4m}$ = sound level measured in the anechoic room [dB re 20 µPa],

$20.\lg(15/4,4)$ = correction for the difference in measurement distances indoors and outdoors [dB],

C_{ff} = free field correction for the microphone used in the anechoic room [dB] and

6 = the expected enhancement of the sound pressure level due to pressure doubling on the the hard surface [dB].

4 RESULTS

For the plate of 2 m x 2 m deviations from the expected enhancement of 6 dB are less than 3 dB in all 1/3 octave bands (see tables 1 and 2).

5 DISCUSSION

For a plate of 2 m x 2 m the results are similar to Andersen's results, shown in appendix 2 of the recommendations for measurement of noise emission from wind turbines [1].

When measuring with a microphone in a hard plate of sufficient size, the inaccuracy of the measured source strength of a wind turbine is expected to be smaller than in ther case of measurements performed above an absorbing ground for which 'standard' properties are assumed.

Our experiment is not a complete validation, because only the dimensions of the plate have been varied. Effects of inaccuracy of the position of the microphone or of a (narrow) slit around the microphone in the plate have not been studied and we did the experiment on only one ground location (one impedance). In practice, on churned up grounds for example, bigger impedance jumps can occur at the boundary of the hard plate, giving bigger diffraction effects.

6 CONCLUSION

A big hard plate is uneasy to handle, but its use is worthwhile because it significantly improves the quality of the results. Validation experiments on other locations and different types of ground are recommended.

REFERENCES

[1] Ljunggren, S., A. Gustafsson (editors). Recommended practices for wind turbine testing. 4. Acoustics. 'Measurement of noise emission from wind turbines', 2. edition 1988.

[2] Jong, B.A. de, A. Moerkerken, J.D. van der Toorn. J. Sound Vib. 86 (1), 1983, pp. 23-46. 'Propagation of sound over grassland and over an earth barrier'

[3] Bruijn, A. de, F.H. van Tol, K. Verhulst. Proceedings Inter Noise 85, pp. 1207-1210. 'Acoustic aspects of wind turbines; Suppression of wind-induced noise of microphones'.

Table 1

Lengths and widths of the hard plates involved in the experiment; the centre frequency [Hz] of the 1/3 octave band in which the maximum value of $\Delta L = L_{in\ plate} - L_{free\ field} - 6$ appears and the maximum value of ΔL for a 1/3 octave band, for each plate.

length x width of the different PLATES	centre frequency 1/3 octave band [Hz]	maximum of ΔL [dB]
0,5 m x 0,5 m	1000	3,6
1 m x 0,5 m	1000	5,5
1 m x 1 m	630 1000	3,0 3,0
1 m x 2 m	630	3,7
2 m x 2 m	1600	2,6

Table 2

$\Delta L = L_{in\ plate} - L_{free\ field} - 6$,
for a plate of 2 m x 2 m,
for 1/3 octave bands with centre frequency f.

f [Hz]	ΔL [dB]
63	-0,9
80	1,3
100	-0,7
125	-1,2
160	0,1
200	0,0
250	-0,2
315	2,0
400	0,7
500	1,3
630	2,1
800	1,9
1000	0,5
1250	1,2
1600	2,6
2000	0,8
2500	2,1
3150	1,3
4000	1,8
5000	1,7
6300	1,7
8000	1,5
10000	1,3

RESEARCH ON PREDICTION OF WIND TURBINE ROTOR NOISE

-VIEWGRAPHS-

H. van den Wal

1 Predicton models for turbines with

- horizontal axis
- upwind rotor
- rotor blades not stalled

2 NLR model RHOAK, based on GROSVELD, with modifications

3 Validation measurements by TPD and SPE on a 25m HAT

4 NLR model RHOAK-2, adjusted to fit the 25 m HAT measurements

5 Future validation of RHOAK/RHOAK-2 on turbines of different types

SOURCE MECHANISMS

1 Turbulent boundary layer / rotor blade trailing edge - interaction

2 Vortex shedding from a blunt trailing edge

3 Atmospheric turbulence / rotor blade - interaction, resulting in:
   - 3.1 loading noise, due to fluctuating lifting forces
   - 3.2 thickness noise, due to fluctuating resistance forces

4 Tip vortex noise

5 Other mechanisms, such as noise from slits, holes, etc.

SOURCE STRENGTH

$$L_W = 10 \cdot \log B \cdot \sum_{k=1}^{n} W_k \cdot D_k \cdot F_k \cdot K_k$$

L_W : Rotor noise source strength in some direction in dB re 10^{-12} W

B : Number of blades

n : Number of contributing source mechanisms

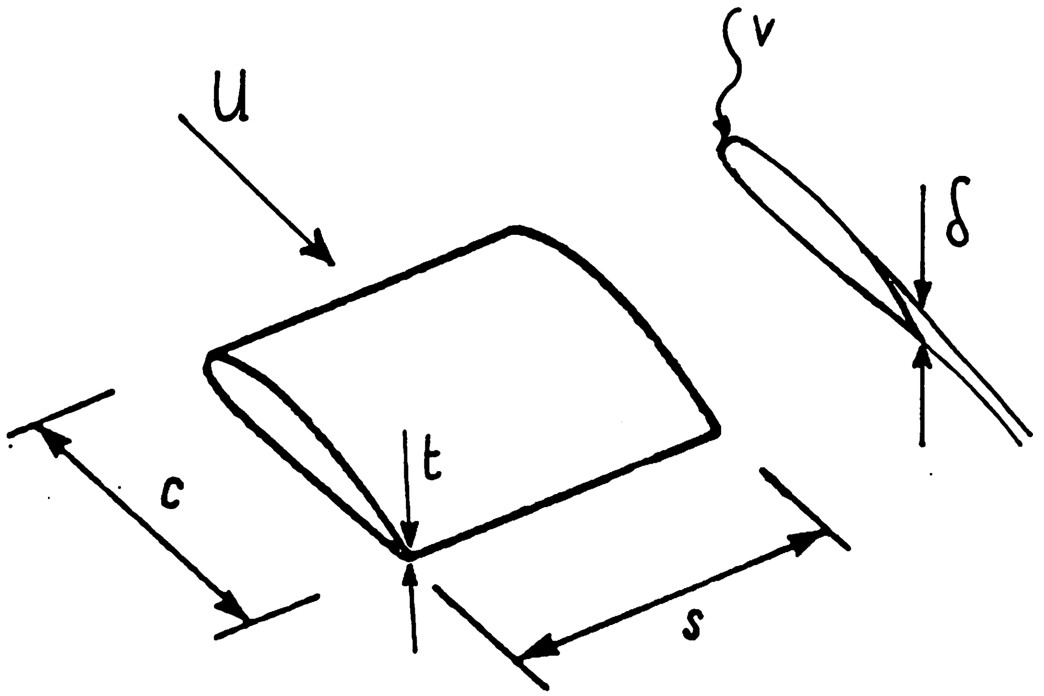
W_k : Unscaled sound power

D_k : Directivity

F_k : Frequency dependent scale factor

K_k : Constant scale factor, determined from empirical data

SOUND POWER GOVERNING PARAMETERS



U_T : tip speed

U : free stream velocity

c : chord

s : Length of blade element

t : thickness of trailing edge

δ : trailing edge boundary layer thickness

v : rms value of axial turbulent velocity fluctuations

UNSCALED SOUND POWER W_k

$$W_1 = U^5 \cdot \delta \cdot s$$

$$W_2 = U^{5.3} \cdot t \cdot s \qquad (\text{for } t < \delta/\ell)$$

$$W_3 = U^4 \cdot v^2 \cdot s \cdot c$$

$$W_4 = W_3$$

$k = 1$: trailing edge / boundary layer

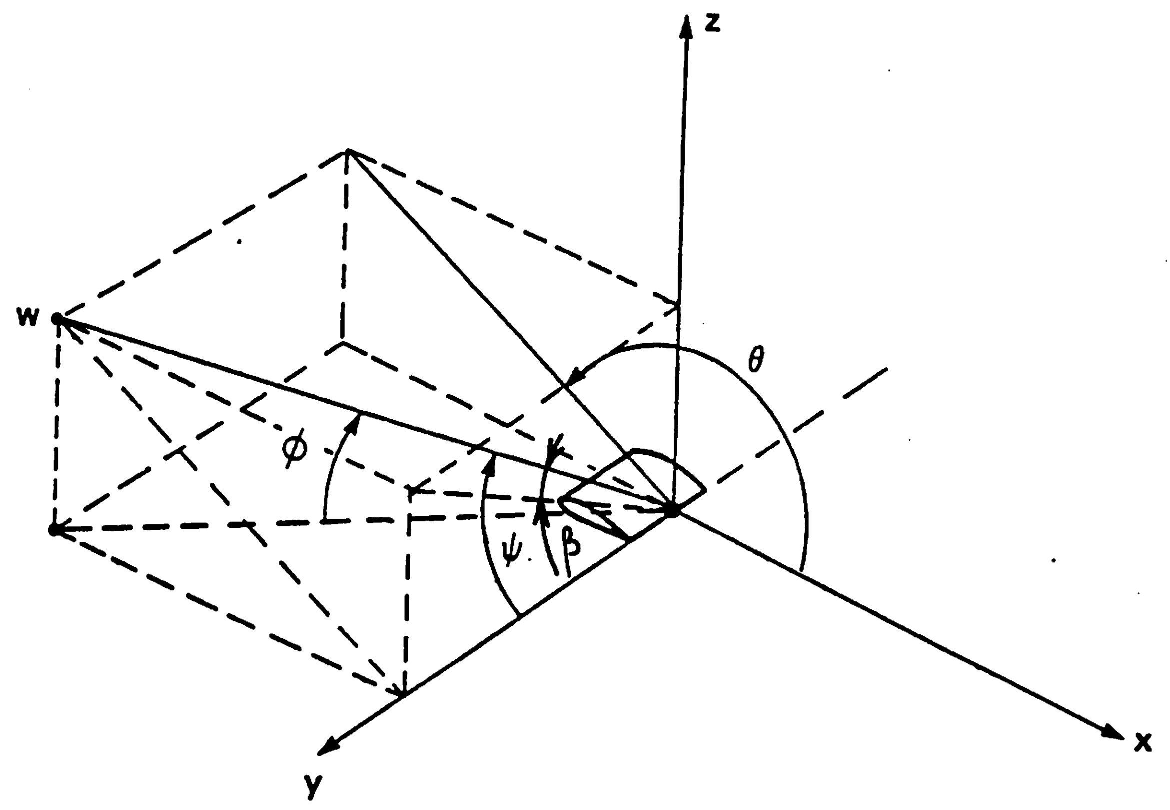
$k = 2$: blunt trailing edge

atmospheric turbulence :

$k = 3$: loading noise

$k = 4$: thickness noise

DIRECTIVITY D_k



$$D_1 = \sin^2\psi \, . \, \sin^2(\theta/2) \, . \, f_1(M, \theta)$$

$$D_2 = \sin^2\psi \, . \, \sin^2(\theta/2) \, . \, f_2(M, \theta)$$

$$D_3 = \sin^2\phi$$

$$D_4 = \sin^2\beta$$

$M = \dfrac{U}{V_{Sound}}$: Mach Number.

FREQUENCY DEPENDENT SCALE FACTOR F_k

$$F_1 = \left[\sqrt{10 \cdot f \cdot \delta / U} + 0.5 / \sqrt{10 \cdot f \cdot \delta / U}\right]^{-4}$$

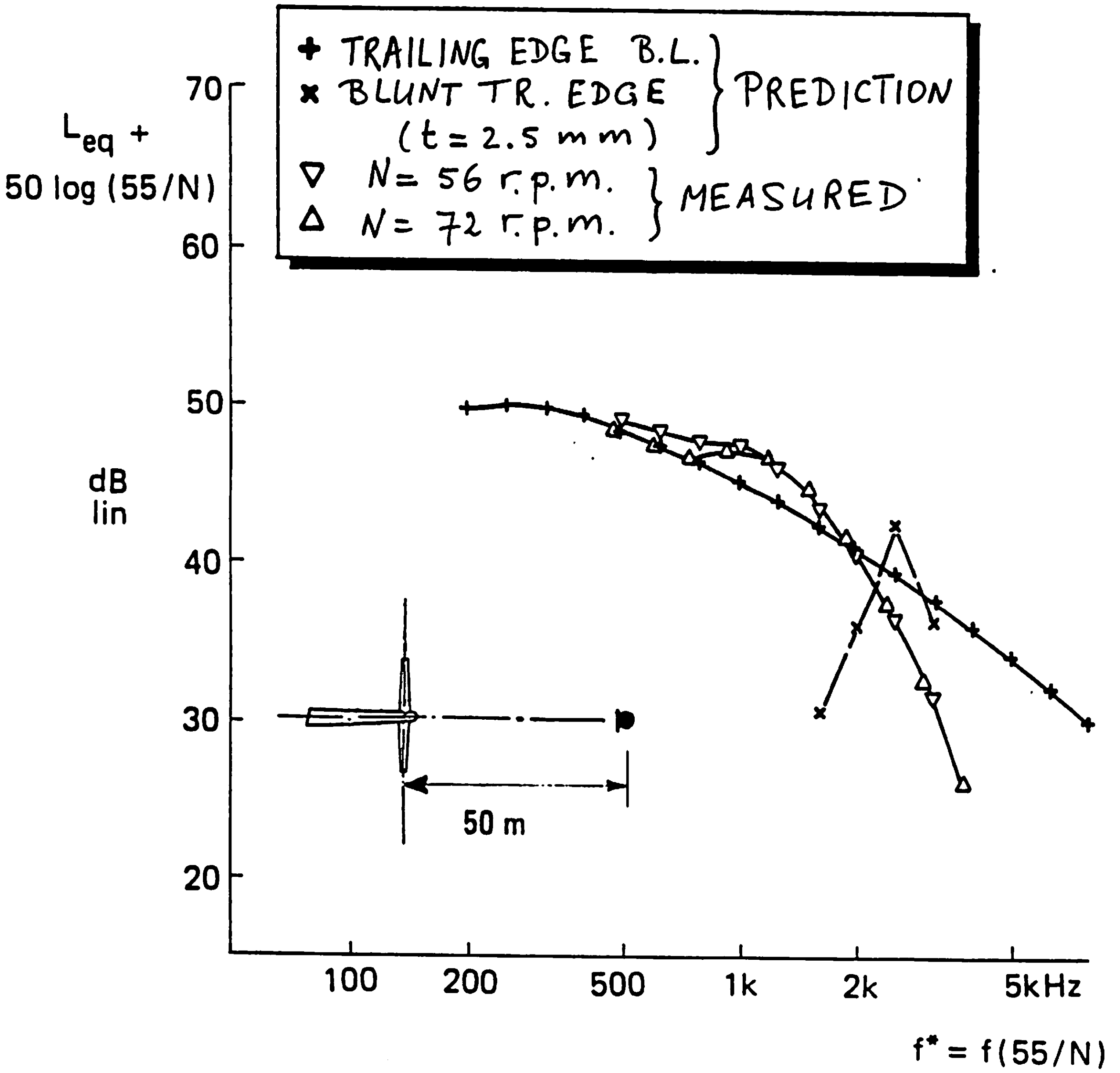
$$F_2 = \begin{cases} 1 & \text{for } f \approx 0.1 \cdot U / t \\ 0 & \text{elsewhere} \end{cases}$$

$$F_3 = \left[f \cdot (H - 0.7 \cdot R) / U_T\right]^{-2}$$

$$F_4 = \begin{cases} F_3 & \text{for } b = 4 \cdot f \cdot c / V_{sound} \leq 1 \\ F_3 + 10 \cdot \log b & \text{for } b > 1 \end{cases}$$

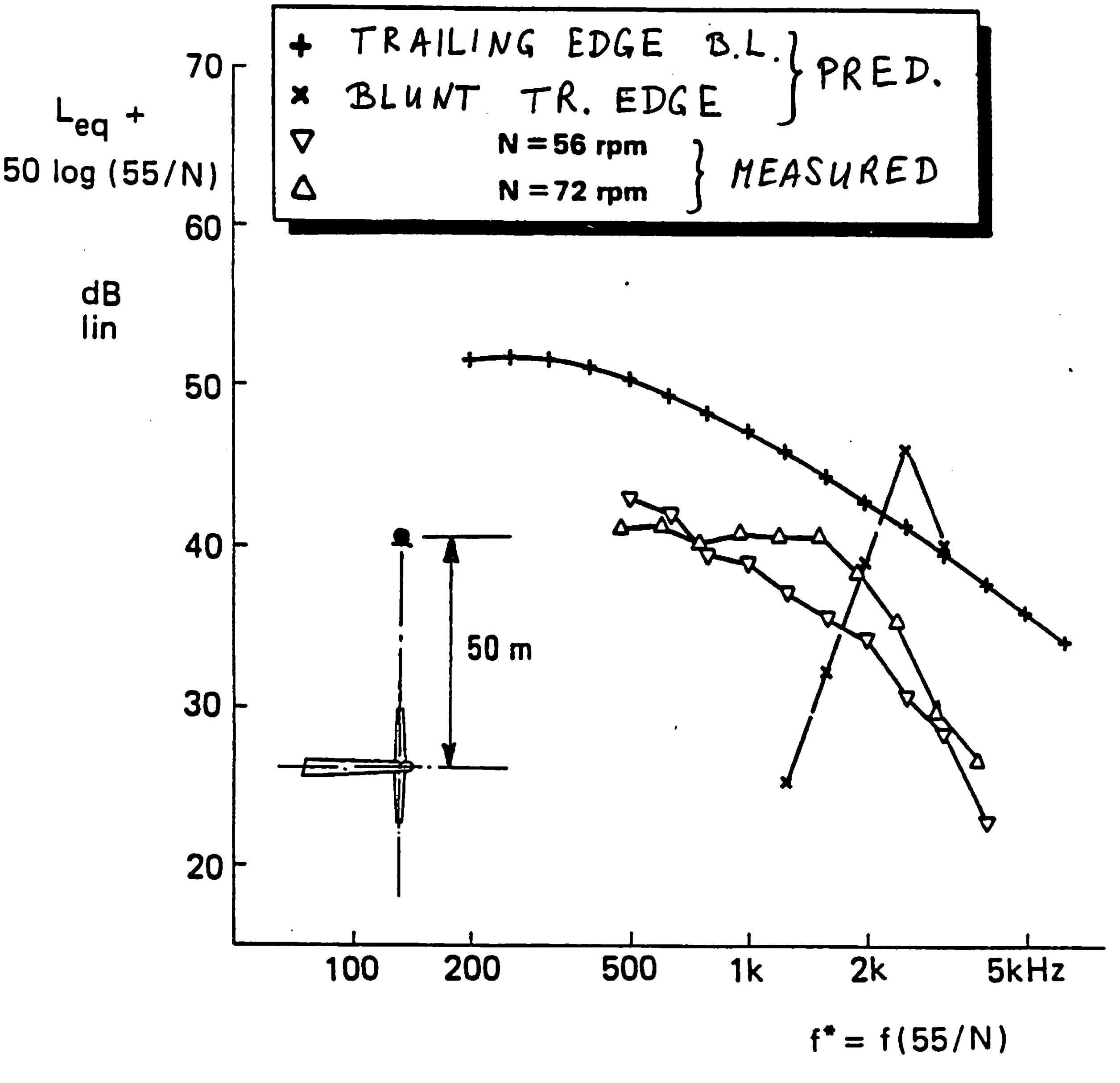
TRAILING EDGE NOISE

PREDICTION VS MEASUREMENT



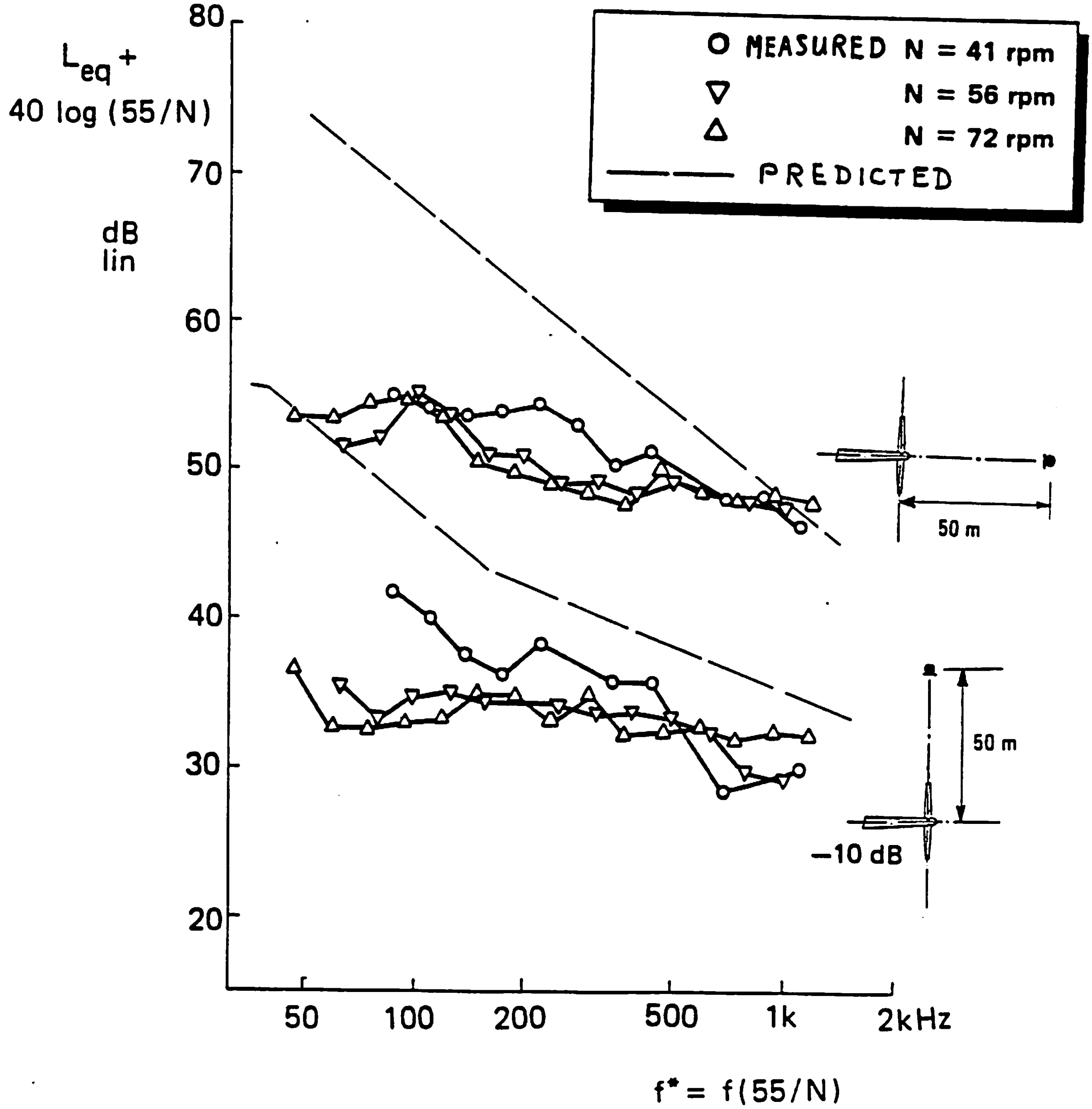
TRAILING EDGE NOISE

PREDICTION VS MEASUREMENT



TURBULENCE NOISE

PREDICTION VS MEASUREMENT



PREDICTION MODEL ADJUSTMENTS

I REMOVED:

1 Vortex shedding from a blunt trailing edge (for the time being) ($k=2$)

2 Thickness noise due to atmospheric turbulence: included in loading noise ($k=4$)

3 Tip vortex noise

4 Directivities D_1 and D_3 set to 1

II UNCHANGED:

Unscaled sound powers W_k
($k=1$, $k=3$)

PREDICTION MODEL ADJUSTMENTS (cont'd)

III MODIFICATIONS:

1 Directivity for observer points in the rotor plane: determined from the 25 m HAT measured data:

$k=1$: $\quad 1/4 \leq D_1 = f \cdot c_{85\%} / (40 \cdot U_T) \leq 1$

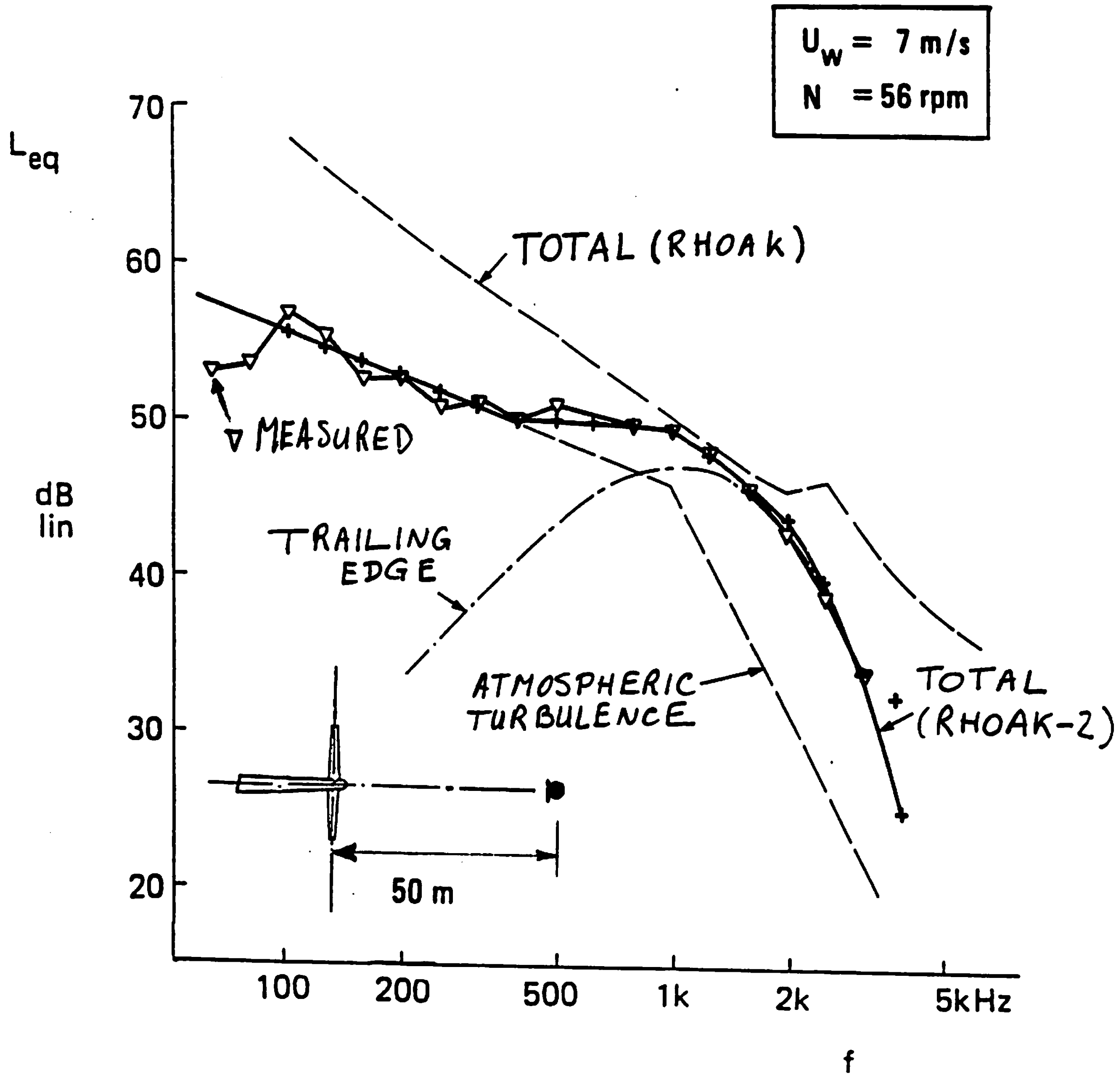
$k=3$: $\quad D_3 = 0.25$

2 Frequency dependent scale factors F_k: adjusted to fit the measured data.

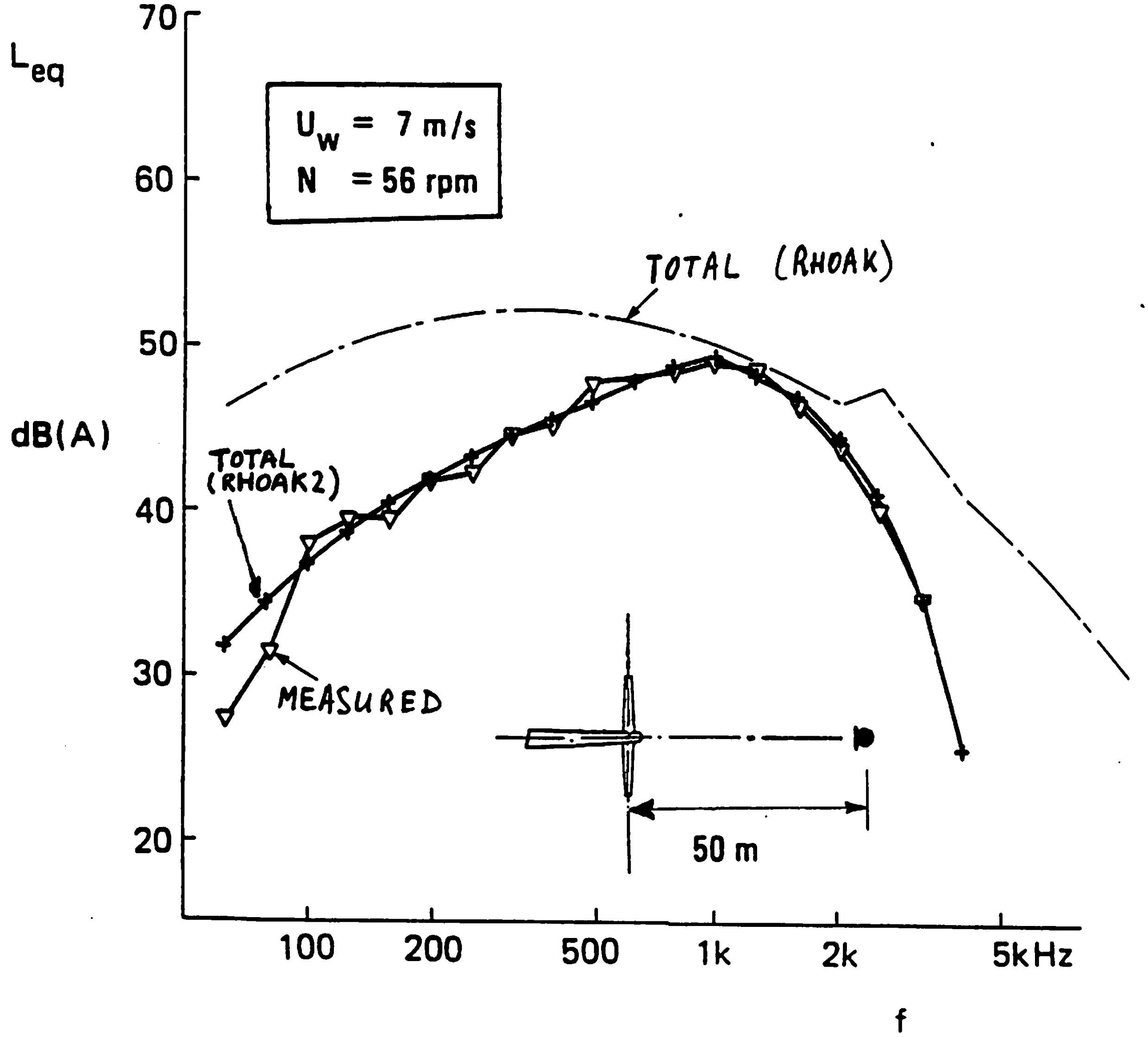
$k=1$: $\quad 10 \cdot \log F_1 = -170 \cdot [\log(2.5 \cdot f \cdot \delta / U)]^2$

$k=3$: $\quad F_3 = [f \cdot (H - 0.7 \cdot R) / U_T]^{-1}$

3 Constant scale factors K_k: adjusted to fit the measured data

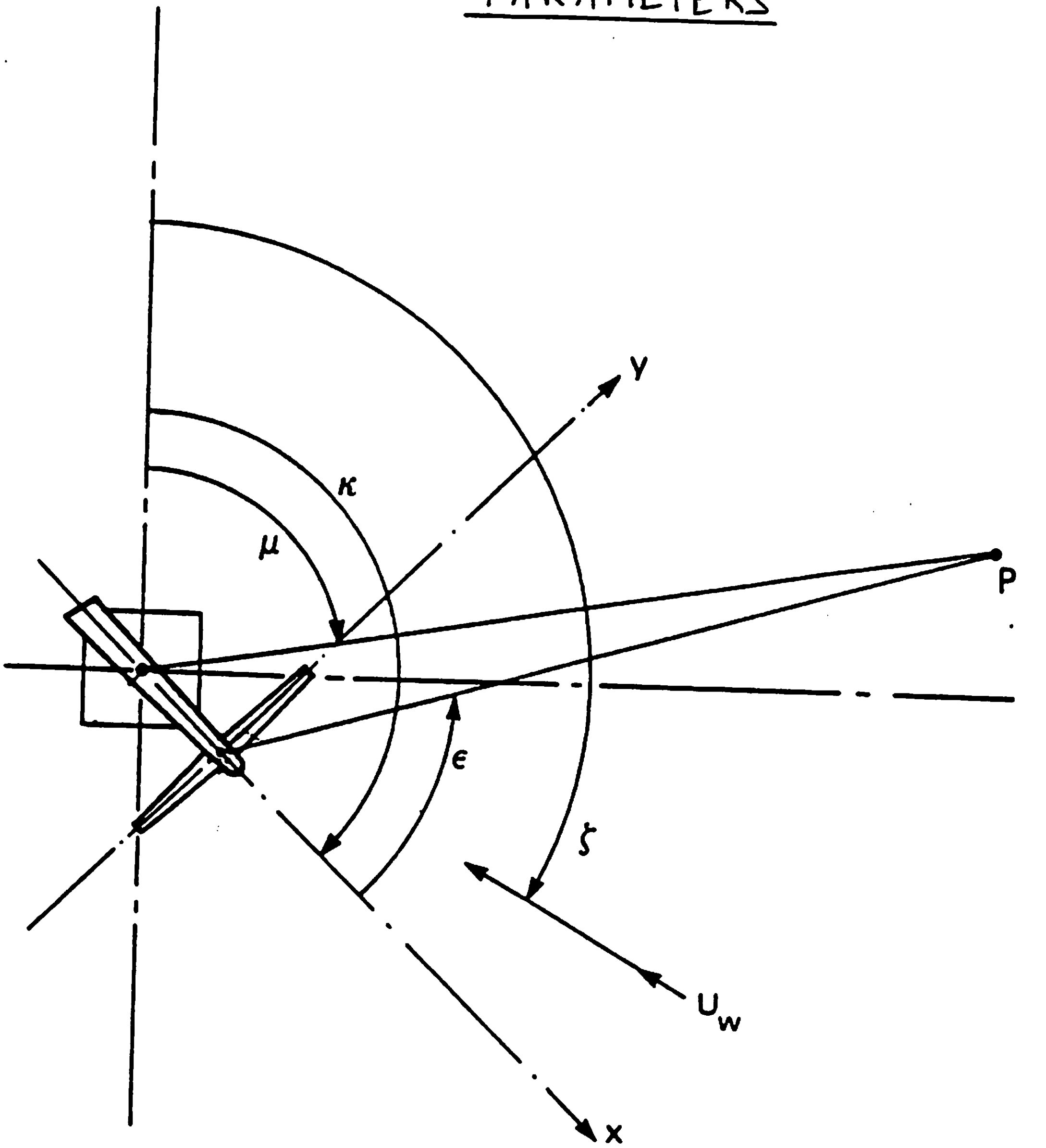


U_w = 7 m/s
N = 56 rpm
L_{eq}
dB lin
70
60
50
40
30
20
TOTAL (RHOAK)
MEASURED
TRAILING EDGE
ATMOSPHERIC TURBULENCE
TOTAL (RHOAK-2)
50 m
100
200
500
1k
2k
5kHz
f



70
60
50
40
30
20
L_{eq}
dB(A)
U_W = 7 m/s
N = 56 rpm
TOTAL (RHOAK)
TOTAL (RHOAK2)
MEASURED
50 m
100
200
500
1k
2k
5kHz
f

GEOMETRICAL PARAMETERS



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IEA - Implementing Agreement LS WECS
Expert Meetings

1. Seminar on Structural Dynamics, Munich, October 12, 1978
2. Control of LS-WECS and Adaptation of Wind Electricity to the Network, Copenhagen, April 4, 1979
3. Data Acquisition and Analysis for LS-WECS, Blowing Rock, North Carolina, Sept. 26-27, 1979
4. Rotor Blade Technology with Special Respect to Fatigue Design Problems, Stockholm, April 21-22, 1980
5. Environmental and Safety Aspects of the Present LS WECS, Munich, September 25-26, 1980
6. Reliability and Maintenance Problems of LS WECS, Aalborg, April 29-30, 1981
7. Costings for Wind Turbines, Copenhagen, November 18-19, 1981
8. Safety Assurance and Quality Control of LS WECS during Assembly, Erection and Acceptance Testing, Stockholm, May 26-27, 1982
9. Structural Design Criteria for LS WECS, Greenford, March 7-8, 1983
10. Utility and Operational Experiences and Issues from Mayor Wind Installations, Palo Alto, October 12-14, 1983
11. General Environmental Aspects, Munich, May 7-9, 1984
12. Aerodynamic Calculational Methods for WECS, Copenhagen, October 29-30, 1984
13. Economic Aspects of Wind Turbines, Petten, May 30-31, 1985
14. Modelling of Atmospheric Turbulence for Use in WECS Rotor Loading Calculation, Stockholm, December 4-5, 1985
15. General Planning and Environmental Issues of LS WECS Installations, Hamburg, December 2, 1987
16. Requirements for Safety Systems for LS WECS, Rome, October 17-18, 1988
17. Integrating Wind Turbines into Utility Power Systems, Herndon (Virginia), April 11-12, 1989
18. Noise Generating Mechanisms for Wind Turbines, Petten 7-28, 1989 (The Netherlands), November 27-28, 1989
19. Wind Turbine Control Systems, Strategy and Problems, London, May 3-4, 1990