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EFFECT OF VORTICITY DISTRIBUTION ON THE BLADES ON FAN NOISE

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Abstract

Tests have been performed to determine the connection between noise emission of radial flow fans, impellers, with different inlet design, and vorticity distribution on the blades. An inlet cone protruding into the impeller was found to reduce significantly the radiated sound power level. Measurements showed that for the tested impellers about the duty point corresponding to maximum efficiency, vorticity distribution on the blades has little effect on the sound power level.

Keywords: radial flow fan, noise, vorticity distribution.

Introduction

A variety of problems arise in practice concerning air flow. Air is often supplied by fans, causing noise — which should be minimized because of its impact on people. Engineers need therefore parameters helping them to design fans with specified flow characteristics at the lowest noise level possible.

Data in the literature on acoustics of axial flow fans are more extensive than those for radial ones. Apart of their scarcity, they are not informative enough neither for researchers nor for practical designers partly due to the complexity of the flow pattern in radial fans. Another difficulty is due to the fact that the impeller is rotating in a spiral casing, causing an inadequately known acoustic effect. All these prompted a research program on the noise of radial flow fans containing experiments concerning the acoustic consequences of changing certain parameters. In the following, measured data obtained from tests made at the Department of Fluid Mechanics, TU Budapest, and conclusions drawn therefrom, will be presented.

Noise measurements of radial flow fans showed that the noise emitted consists mostly of broad-band noise. In usual operation ranges the effect of rotational noise is negligible because of the low peripheral velocity. Neither will the discrete frequency, known as cut-off noise prevail if certain aspects of constructing spiral casings are respected [1]. A more difficult problem for the acoustic designer will be to reduce the broad-band noise component, or that one source may be reduced to flow turbulence. According to observations and estimations, noise of the turbulent boundary layer is negligible [2]. The other source of broad-band noise is due to the pressure fluctuation on the blade surface caused by the free stream velocity of variable direction and magnitude due to the turbulence of the oncoming flow [2]. Third source of the broad-band noise is the so-called wake noise [2], [3] resulting from the shedding of vortices behind the trailing edge of the blades. Vortices continuously born and flowing away behind the blades cause circulation change around the blade, in conformity with Thompson's law. Since the circulation periodically changes on the blade, the velocity and pressure fields vary periodically as well. A separation of flow along the impeller shroud may cause also broad-band noise.

Aim of Measurements

Measurements to be described below have been concerned with two phenomena often dominant among radial flow fan noise components: wake noise, and the noise generated by the random flow behind the separation zone due to the improper design of the inlet.

The goal was therefore to test the influence on the noise of the flow pattern depending on the form of the so-called inlet gap between inlet cone and impeller, and the vorticity distribution on the blade.

Test Procedure

Measurements were made on impellers rotating free, without a spiral casing, in order to eliminate the acoustic effect of same, hence to test the noise caused by the impeller alone. No reference has been found in the literature on similar systematic measurements. It has been taken into account that the flow pattern developing in an impeller rotating free is somewhat different from that in a spiral casing. Its effect on the broad-band noise source is, however, assumed small enough not to cause any qualitative changes in the noise source action.

Tested Impellers

The set of tested impellers aimed to compare impellers, designed for the same duty point, with same dimensions but with different vorticity distributions on the blade. It should be kept in mind that the effect of the vorticity distribution on the blade depends heavily on the required design duty point. The blade load, depends of the number of blades and so the blade vorticity determines the velocity fields around it. For high blade loads, considerable velocity changes may occur on both (pressure and suction) sides. High decelerations namely may lead to flow separation.

Impellers with low blade loads are probably less sensitive to the kind of vorticity distribution on the blade. In this case, within certain design point limits impellers with equally favourable flow pattern may be produced by blades with different velocity distributions. For impellers with medium blade loads, however, the selection of the proper vorticity distributions is a delicate matter, since here the shape of the velocity distribution on the blade causes important differences in impeller flow parameters.

Again, for impellers with high blade loads, vorticity distribution on the blade is of lesser importance. For such impellers, poor efficiencies are frequently experienced, because no kind of vorticity distribution can keep the flow from separation.

In the first test series, impellers with relatively increased blade widths and medium blade loads were aimed at. The following non-dimensional parameters were chosen:

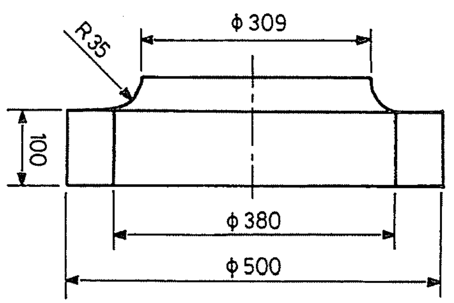

Fig. 1. Main dimension of the impellers

flow coefficient:

$$\phi^* = \frac{q_v}{D_2 \cdot \pi \cdot b \cdot u_2} = 0.28\,,$$

ideal head coefficient:

$$\phi_{\mathrm{id}} = \frac{\Delta p_{t,\mathrm{id}}}{\frac{\rho}{2} u_2^2} = 0.82\,,$$

where q_v [m^3/s] theoretical rate of flow,
$\Delta p_{t,\mathrm{id}}$ [Pa] ideal total pressure rise,
D_2 [m] outer diameter of impeller,
b [m] impeller width,
ρ [kg/m^3] density,
u_2 [m/s] peripheral velocity at diameter.

Fig. 1 shows the main dimensions of the impellers.

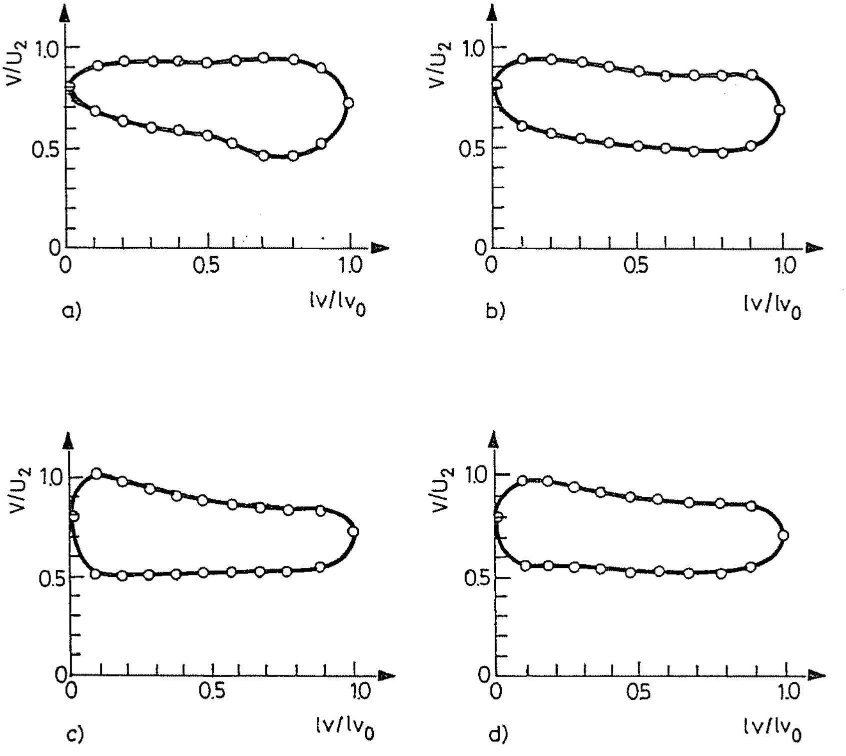

Fig. 2. Velocity distribution on blades of impellers A, B, C and D
a.) Velocity distribution on the blade of impeller A (backward drawn vorticity distribution)
b.) Velocity distribution on the blade of impeller B (constant vorticity distribution)
c.) Velocity distribution on the blade of impeller C (strongly forward drawn vorticity distribution)
d.) Velocity distribution on the blade of impeller D (mildly forward drawn vorticity distribution)

$N = 14$ blades have been chosen, and their shape have been determined by Gruber's method [4]. According to it transforming the circular cascade to an infinite straight one, certain vorticity distribution was taken in this straight cascade along the chord length. Based on earlier observations, four alternatives have been developed: velocity distributions on blades of impellers A, B, C and D are seen in *Fig. 2* on which v/u_2 stands for the relative velocity and lv/lv_0 for the relative camber line length.

Two impellers (A and B) with different velocity distributions, and with axial and radial gaps, resp. at the inlet have been tested first. In one of the alternatives with radial gap, the axial distance is zero. Namely the fitting surfaces are exactly in the same plain. The two kinds of inlet gaps are sketched in *Fig. 3*.

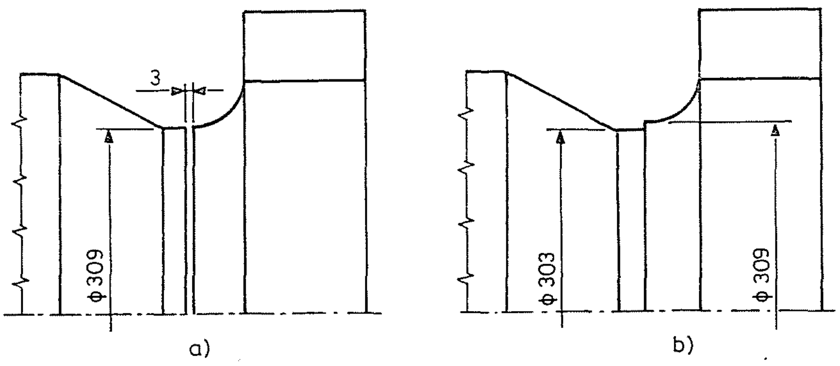

Fig. 3. The two kinds of inlet gaps. a.) axial gap inlet, b.) radial gap inlet

The second part of the tests was to measure all the four impellers (A, B, C and D) with different vorticity distributions.

The Test Arrangement

The tested impeller has been fitted directly on to the shaft of a DC balance motor of variable rpm. The suction duct connection was designed so that the inlet cone was easy to change. Nothing was in the path of the air flow from the impeller not to cause noise by interaction. The test rig was placed in a reverberant testing room of 218 m^3, because of two reasons. Firstly, because the free-field noise output determination was hindered by the difficult directivity pattern of the noise source, and by the air flow from the impeller, affecting the microphone. Secondly, because the different duty

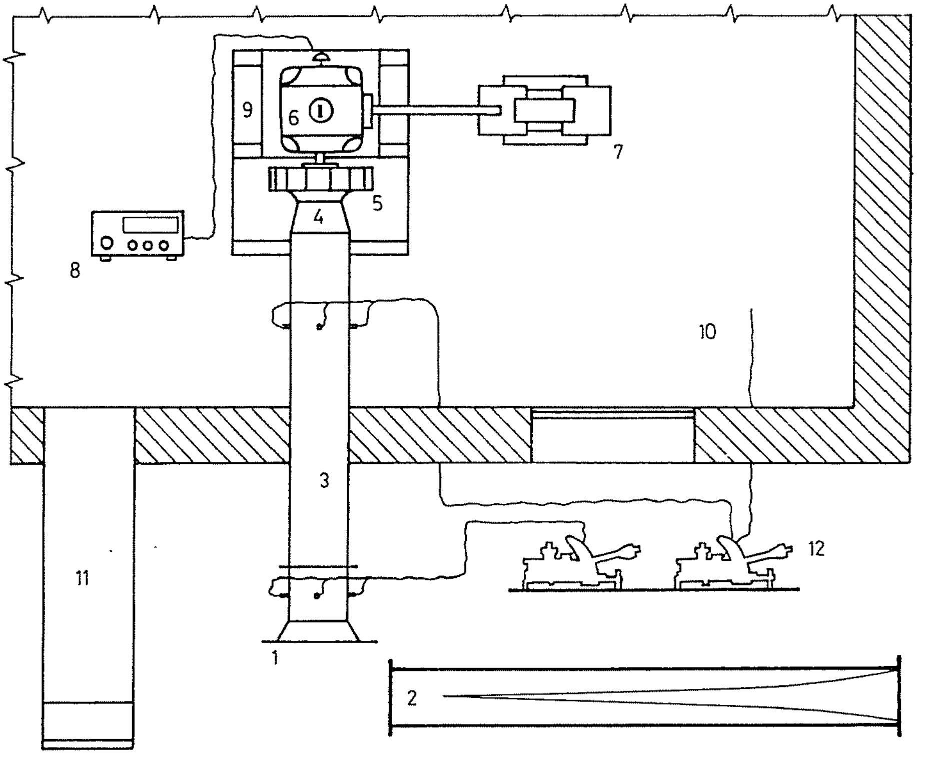

Fig. 4. Scheme of the test arrangement
1.) Inlet cone for flow rate measuring, 2.) Anechoic termination, 3.) Inlet duct, 4.) Inlet cone, 5.) Impeller, 6.) Balance motor, 7.) Balance, 8.) Revolution counter, 9.) Machine stand, 10.) Reverberating test room, 11.) Control unit, 12.) Micromanometers

points could be performed otherwise only by throttling in the inlet duct, and the throttling device would have caused considerable wave reflection altering perhaps the noise source itself, in addition becoming an intensive noise source also. Actually, the aperture in the wall of the testing room was connected to a silencer lined with sound absorbing material, ending in a throttling device for controlling the air flow from the room. The balance motor was placed in the testing room, therefore the emitted background noise had to be determined. Comparing the measured noise of the impeller and of the DC motor, the motor proved in all frequency bands of interest to be less noisy by at least 10 dB. Scheme of the test arrangement is seen in *Fig. 4*.

For each variety, flow and acoustic parameters have been separately measured.

Determination of Flow Characteristics

Every test started by measuring the flow characteristics of the fan and plotting the non-dimensional characteristics, in this case the static head coefficient ψ_{st} and static efficiency η_{st} vs. flow coefficient φ. The arrangement did not permit else but to determine static head coefficient and efficiency. (Measurements were ruled by specifications of Hungarian Standard MSZ 11110.):

$$\varphi = \frac{q_v}{\frac{D_2^2 \cdot \pi}{4} u_2},$$

where q_v $[\mathrm{m^3/s}]$ rate of flow,
D_2 $[\mathrm{m}]$ impeller diameter,
u_2 $[\mathrm{m/s}]$ peripheral velocity for diameter D_2,

$$\psi_{st} = \frac{\Delta p_{st}}{\frac{\rho}{2} u_2^2},$$

where Δp_{st} [Pa] static pressure rise,
ρ $[\mathrm{kg/m^3}]$ density.

Flow rate was measured by means of an inlet cone, to help also development of a regular afflux in spite of the relatively short suction pipe. Test arrangements followed the relevant specifications of ISO 5221.

The pressure differences were measured by inclined-tube and Betz micromanometers, the rpm was checked by a frequency counter. Torque measurements were made by the balance motor. In addition to the usual characteristic curve, another characteristic curve p_{amb}–p_{in} versus φ has been plotted, in which p_{amb}–p_{in} is the difference between static ambient and inlet pressures. It helped to obtain φ from the measurable pressure difference in acoustic measurements, while the inlet cone is replaced by an anechoic termination.

Acoustic Measurements

First, the reverberant room was acoustically tested, by measuring reverberation time for each octave band using a white noise generator, filtered by an octave-band filter. In conformity with the relevant standard specification, measurements were made in six different microphone positions. Noise source (impeller) position, microphone positions, and principal dimensions of the reverberant room can be seen in *Fig. 5.*

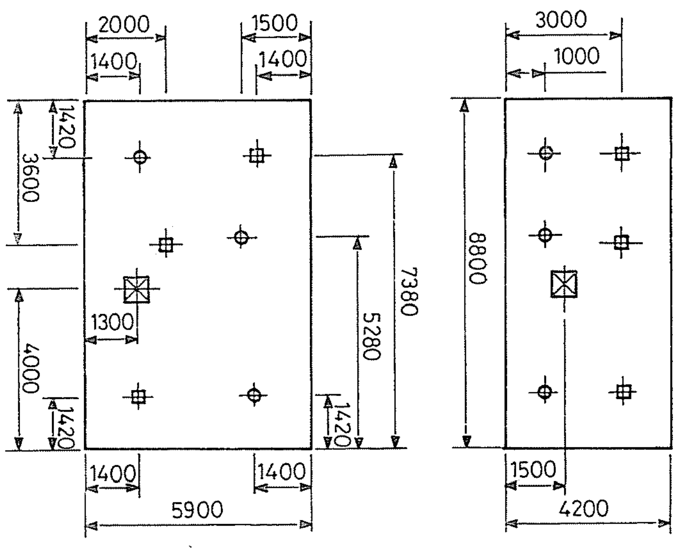

Fig. 5. Noise source position, microphone positions and principal dimensions of the reverberation room.
-⊕- microphone position -⊞- — machine position

Measurements were performed mainly by means of Brüel & Kjaer instruments. To avoid the interference of air flow in the room the 1/2" microphone type 4134 has been fitted by a wind shield type UA 0237. The signal from the microphone passed through a preamplifier type 2615, then to an octave/third octave analyzer type 2112. In acoustic measurements a level recorder type 2305 has been used. Later on, the signal was processed by an FFT analyzer type OMC 105. The computer combined with the analyzer permitted fast processing of measurements. The computer helped to obtain, in addition to the narrow-band measurement results, at the same time, the octave-band spectrum, too.

Every measurement was preceded by microphone calibration. Correlation between sound pressure level and sound power level relied on reverberation times in octave-bands. Thereby sound power level could be calculated from sound pressure levels determined at six points, then averaged:

$$L_{W\mathrm{oct}} = \tilde{L}_{\mathrm{oct}} - 10\lg\frac{T_N}{T_1} + 10\lg\frac{V}{V_1} - 13\,,$$

where $\tilde{L}_{\mathrm{oct}}$ [dB] mean pressure level in the octave band,
$L_{W\mathrm{oct}}$ [dB] octave-band power level,
T_N [sec] reverberation time of the testing room in the octave band,
T_1 [sec] 1 sec,
V [m^3] testing room volume,
V_1 [m^3] 1 m^3.

Experimental Results

For the sake of clearness and illustrativeness, the plotted experimental results have been tabulated. Flow and acoustic characteristic curves of impellers with different inlet gaps and vorticity distributions are seen in *Fig. 6*.

Flow and acoustic characteristic curves of impellers with different vorticity distributions are seen in *Fig. 7*.

Evaluation of Measurement Results

In the first measurement series, the combined effect on inlet gap design and vorticity distribution has been tested. Measurement results compiled in *Fig. 6* lead to the following conclusions:

It is well known that the flow characteristics of an impeller with high flow coefficient having radial inlet gap is much more favourable than of that with an axial one. The advantage of radial gaps can be explained by the favourable impulse of the gap-flow into the main inflow. This flow, by turning into the radial direction is decelerated hence inclined to separation of the shroud. A kind of boundary layer refreshing thereby helps to resist separation. There is, however, an impulse transfer also for axial gaps, but -- due to its wrong direction — with an opposite effect, helping separation

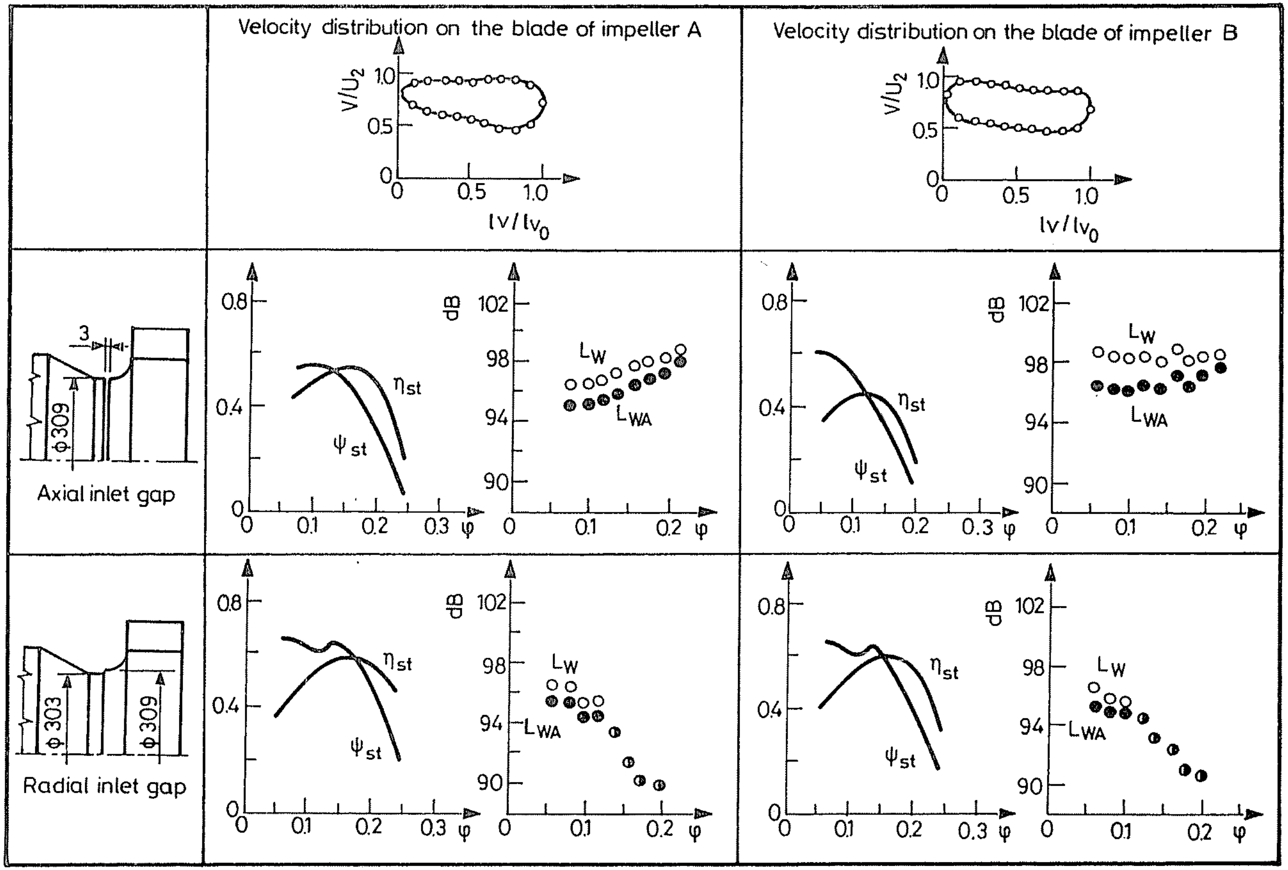

Fig. 6. Flow and acoustic characteristic curves of impellers with different inlet gaps and vorticity distribution

as proved by measurements. A look at *Fig. 6* shows also that the 'gap effect' not only depends on the main dimensions (ratio of inlet diameter to impeller width, or of impeller diameter to width) but also on the vorticity distribution on the blades. Comparing swept-back and constant vorticity distribution impellers (A, B) it is clear that for impellers B radial gaps improve the static efficiency by about 14%. On the other hand, for impeller A, a more 3% of efficiency improvement was observed. Impeller designed for vorticity distribution A may be assumed to be less affected by the inlet gap than is that for B.

Another result of the measurements has been the experienced strong flow pattern dependence on acoustic parameters. Higher sound power outputs on the outlet side were observed for impellers with inlet cones with axial gaps. The sound power level is almost independent of flow coefficient φ in case of impeller B, while for impeller A, the sound power level inexpectedly grows toward the optimum working point.

The radial gap was efficient for reducing impeller noise. Overall sound power levels in function of the φ flow coefficient of the impellers A and B decreased by 8 dB, and by 7.5 dB, respectively (*Fig. 6*). One can say that irrespective of the flow characteristics, from the noise aspect, impellers designed with different vorticity distributions are equally sensitive to the inlet gap design.

Diagrams show that for an axial gap, impeller noise has a significant low frequency content which is not the case for radial gaps, hence here A-weighted and linear overall sound power levels differ by much less. The difference is the most pregnant at and around the optimum duty point. For impellers with axial gap, the marked low-frequency content is certainly due to the separated flow along the full length of the impeller shroud. Pair wise comparison of acoustic characteristic curves shows that sound power level values for both kinds of gaps are nearly equal when throttled. Now, in all four cases, fans worked in a mode of markedly separated flow. Because of the similar (separated) flow pattern, obviously, noises of similar intensity and type arose. With increasing flow coefficient φ, the angle of attack of impeller blades got more favourable, and the flow along the blades smoothened. Thereby noise from the impellers with radial gaps abruptly decreased, while impellers with axial gaps kept noisy. This may point to flow separation on the shroud to continue for axial-gap impellers, and separation would continue to act as noise source in the impeller. In addition to a quieter operation, hence lesser overall sound power, more regular flow in impellers with radial gap resulted in a variation of the sound level spectrum resulting in slight differences between A-weighted and linear overall sound power levels. Missing of the large vortices typical for so-called rough separation significantly reduced low-frequency components. This is

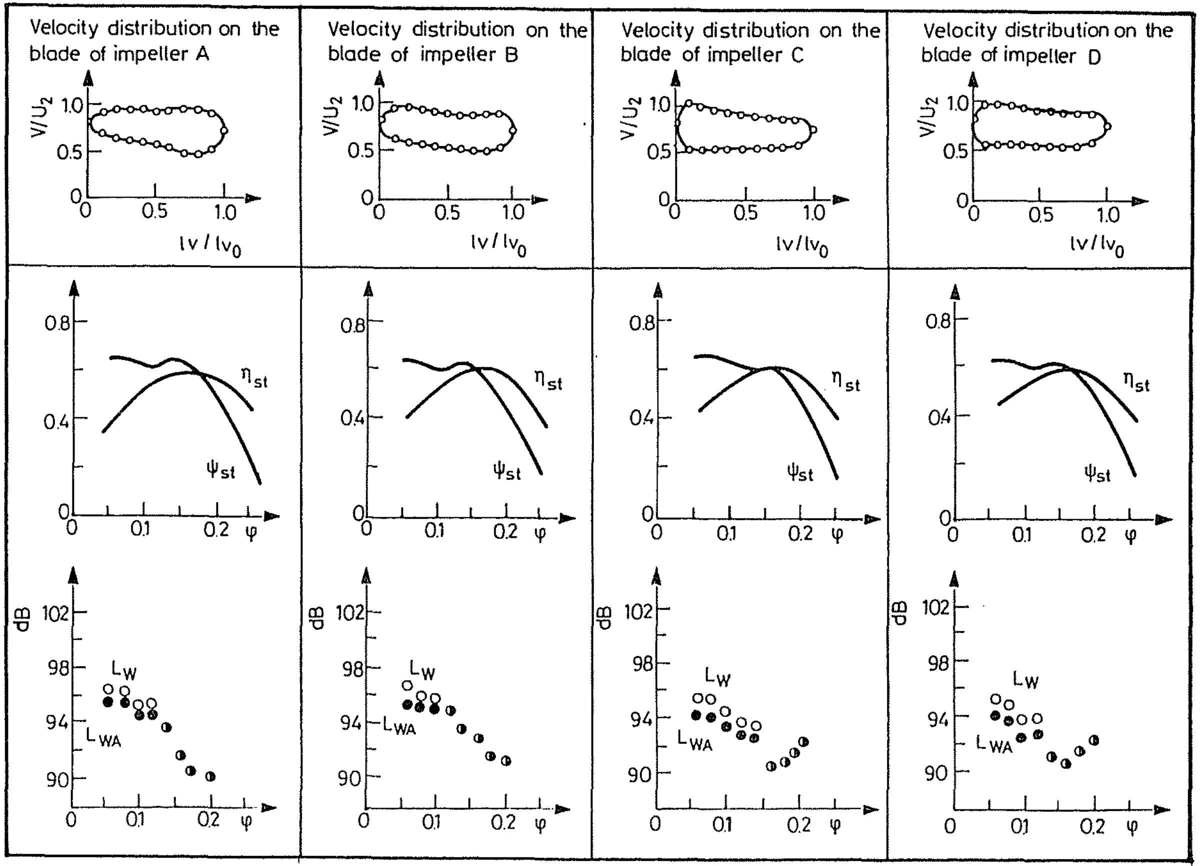

Fig. 7. Flow and acoustic characteristic curves of impellers with different vorticity distributions

of importance also since subsequent damping of low frequency noise usually encounters more difficulties than that of high frequency ones.

In the second test series, exclusively the effect of vorticity distribution on the blade has been investigated, applying radial inlet gaps in all four cases. Measurement results have been recapitulated in *Fig. 7*. In general, measurements showed no essential differences between the four impellers with different vorticity distributions.

Nevertheless, it is worth to consider lesser, seemingly unimportant differences between the characteristic curves. Static efficiencies of different impellers slightly differ. In measurements, impeller A (with drawn back vorticity distribution) proved the poorer, with a maximum static efficiency $\eta_{st} = 58.5\%$, while impeller C (with markedly drawn forward vorticity distribution) had the best efficiency of 62%. Peaks of efficiency curves, similar in shape and in position, were about at flow coefficient $\varphi = 0.165$. Characteristic curves ψ_{st} of the four different impellers were even more similar than those for the efficiency.

The case for the acoustic characteristics of the impellers is similar. Minima of sound power level are equally 90 dB at the outlet sides of the four impellers of different vorticity distributions. But this minimum is at about the optimum flow coefficient of impellers C and D with somewhat drawn forward vorticity distribution while for those A and B (backward drawn and constant vorticity distribution), with increasing φ the sound power level at the outlet side monotonously decreased. Thereby, unusually, the most quiet run of the fan is not about the maximum efficiency but somewhat to the right, at higher flow coefficients.

It can be observed that for all the four types, linear, and A-weighted sound power level values differ little. This is a reassuring fact, namely absence of low frequencies points to the absence of so-called rough separations.

Conclusions

It is well known that a radial gap at the inlet cone of a radial flow impeller much improves its flow characteristics compared to one with an axial gap. Recent tests showed that impellers with a radial gap are less noisy as well.

Recent measurements led to conclude that in impellers tested vorticity distribution on the blade is unimportant for the impellers. No other difference was found but that between the sound power level vs. flow coefficient curve shapes. For impellers with drawn forward vorticity distribution, fan noise is the lowest about the efficiency maximum. While for impellers with constant or backward drawn vorticity distribution, the least noisy operation is at higher than optimum flow rates. Hence, for the two latter types a compromise is needed since flow and acoustic aspects are against each other. It is true also that this difference is not more than 2 dB. Accordingly, it may be stated that at the design of impellers with parameters similar to the tested ones, the designer is free from the acoustical point of view to select the kind of blade vorticity distribution.

It should be noted that obviously such a relatively low number of tested impellers is insufficient for drawing generally valid conclusions. Overall investigation of the acoustic effect of vorticity distribution on the blade requires further investigations on impellers of different main dimensions, design parameters and vorticity distributions. Further tests are needed on the effect of turbulence on impeller noise.

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