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STUDY OF AEROACOUSTIC PHENOMENONS AT DUCTED COOLING FANS

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ABSTRACT
Germany's vehicle industry is greatly concerned with advancing their engines to make them environmentally compatible. This often leads to higher temperatures which have to be reduced by improved fans. However, less experience exists for high efficient fans of large diameters and power as used for trucks and busses. Therefore, two ducted full-scale fan models have been investigated to reduce noise and energy losses. Aerodynamic design parameters were planform, blade number, hub to tip ratio and the size of the slit. The experiments were conducted with a sucking wind tunnel and afterwards with the bus. The first model fan was designed such that the temperature is reduced by 3 degree and the noise by 1.3 dB(A) when mounted inside the bus. However, more than 6 dB(A) less noise was obtained with a second model due to an improved airfoil and a special localization inside the duct. The results indicate some important guidelines for the optimization of fans in the future.

1 - INTRODUCTION
High technologies, developed at Aerospace Research Centers can also be used for other industrial branches as e. g. process engineering or automotive industry. Here, especially the transfer of numerical design tools for propellers, rotors or fans from aviation to ground-bounded vehicles or machines has been effective. During the last decade, several fan blade planforms were optimized to obtain quieter and more efficient cooling fans. At the early eighties the technology transfer started with a 6 dB(A) noise reduction design for a customer of the German car industry [1]. Some further successful designs followed which all led to prototypes that are used in cars or fork lifts today. The computation of the complete cooling system is very complicated. Only isolated fans with an according disturbed onset flow are optimized at the state of art. The aerodynamics of such isolated ducted fans are computed with classical 2-D cascade flow methods [2], modern 3-D singularity or field methods [3]. For the computation of the periodic noise field a modified acoustic analogy method is applied whereas for the stochastic noise field a half-empirical method is developed which is based on turbulence models. All methods are validated with experiments before coupled with an optimization code. The goal of this work is a noise reduction of 4 dB(A) and an increase of the flow rate of 15 %. Reference fan is an 8-bladed conventional fan. All possibilities to increase the performance were ladled out which led to the following design data:

	diameter [mm]	hub/tip ratio	blade number	l [mm]	M_{rot}	φ	ψ
reference fan	750	0.40	8	0.76	0.31	0.17	0.20
fan to be designed	750	0.48	12	0.87	0.31	0.21	0.23

Table 1: Design parameter (M_{rot} rotational Mach number).

In order to obtain a quiet fan it was decided to use laminar airfoils realized by a 4-digits NACA 65 thickness distribution on a circular meanline or by a numerical airfoil design.

2 - DESCRIPTION OF TESTS

For a maximum power of 37 kW and the large fan diameter of 750 mm a sucking wind tunnel with a low speed chamber was built at the Fachhochschule Braunschweig-Wolfenbüttel. All requirements of DIN 24 136/2 were met to obtain a smooth flow with no swirl (see Fig. 1).

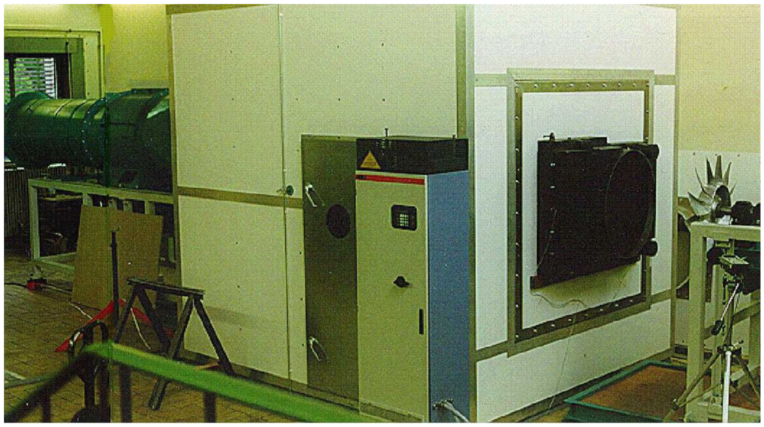


Figure 1: Test rig with wind tunnel and fan 1.

The low speed chamber on the right side has a size of 2.7 × 3.3 × 2.9 m and consists of hard wooden surfaces. The cooler with the housing or the conical inlet is mounted at the right wall side where the fan sucks air form the tube. The static pressure of the fan performance was measured with Betz manometer inside the tube on the left side of Fig. 1, where the flow is controlled by plates. A second fan was installed to compensate the losses. The microphone is mounted 1.45 m above ground and in a distance of 800 mm from the fan center. The angle between the fan axis and the microphone radius is 45 degrees. Acoustic tests around the fan inlet inside the chamber have shown that the sound pressure level is approximately constant within 2 dB. Only relative sound pressure level were considered, so that acoustic reflection effects could be ignored. The acoustic test results have been recorded as dB(A)-weighted Overall Sound Pressure Level (OASPL) or narrow band spectra. As test conditions three operating speeds of 1700, 2400 and 2700 rpm's were selected. Furthermore, the slit size and the location of the fan inside the duct were varied (see Fig. 2).

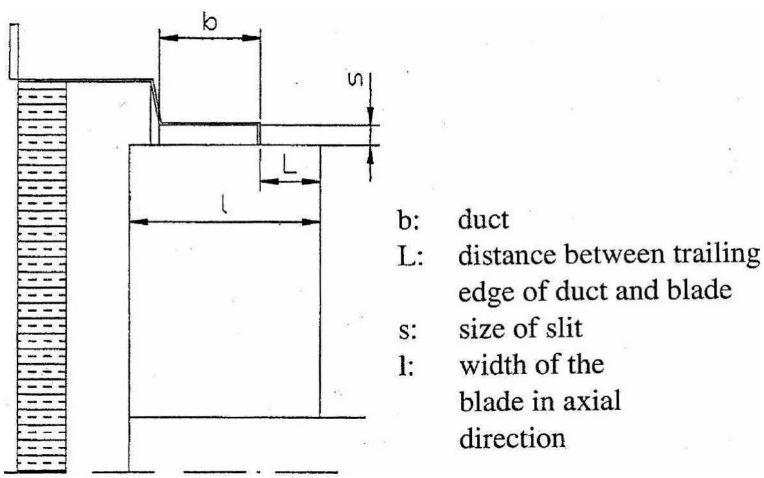
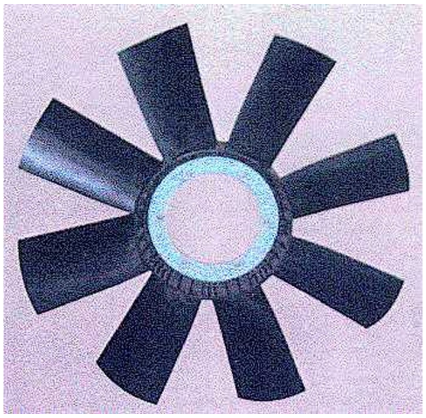


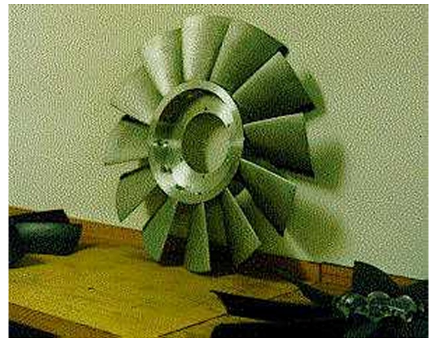
Figure 2: Geometric parameter definition.

3 - AERODYNAMIC DESIGN AND FIRST EXPERIMENTAL RESULTS

Model 1 is a pure aerodynamic design and was computed by the Technical University of Braunschweig, Institute of Fluid Mechanics. The design method used combines the radial-equilibrium calculations for the fan trailing edge plane with blade-to-blade calculations on non-axisymmetric, twisted stream surfaces [2]. The result of the computation is a 12-bladed fan which was built as a full-scale model 1 from aluminum and is shown in Fig. 3.



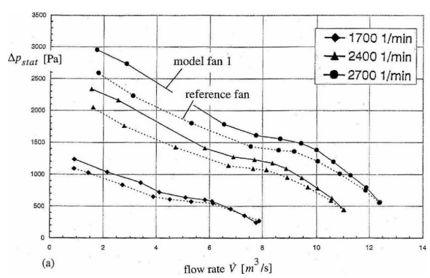
(a): 8-bladed reference fan.



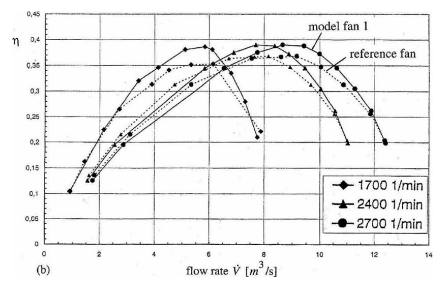
(b): 12-bladed designed fan.

Figure 3: Used full-scale fan models.

The fan models were measured first as isolated fans. The maximum rpm was 2600. To obtain the performance at 2700 rpm, the flow rate and the static pressure were multiplied by a factor of 1.038 respectively its square, which is obtained from the ratio of the rpms (see Fig. 4a).



(a): Static pressure.



(b): Efficiency.

Figure 4: Static pressure performance of the isolated reference fan and fan model 1.

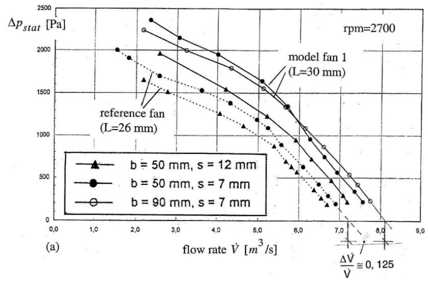
The size of the slit is 12 mm which is 3 mm smaller than inside the bus. At 2700 rpm the reference fan produces a flow rate of 8 m^3/s at the design point, whereas model 1 generates a flow rate of 10 m^3/s there. This is a flow rate increase of 25 %. At the same point the efficiency of model 1 is about 5 % higher than that of the reference fan (see Fig. 4b). The effect still increases with decreasing rpm. But, the efficiency, approximately

$$\eta_{stat} = \frac{\psi_{ts}}{\psi_{tt}} = 1 - \frac{\psi_{tt}}{4\xi^2} - \frac{\varphi^2}{\psi_{tt}}$$

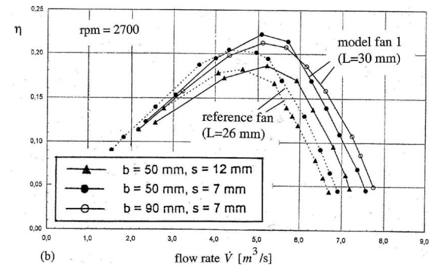
with $\xi = r/R_{tip}$, t for total and s for static is reduced due to the large flow and pressure coefficient. This limits the possibilities here to increase the efficiency by more than 5 %.

Fan and cooler can be considered as a module. Because the losses of the cooler have to be added to the losses of the fan, the efficiency is about the half of the efficiency of the isolated fan. This explains why the efficiency and the flow rate in Fig. 5 are smaller than in Fig. 4.

Around the design point the experimental values of module 1 are larger than those of the reference module. A further remarkable increase of the performance data could be obtained by reducing the slit from 12 to 7 mm. However, the flow rate increase due to the better design becomes smaller when the cooling modules instead of the isolated fans were compared. Instead of 25 % here only the half respectively 12,5 % were obtained. 4 dB(A) noise reduction were obtained when the fan model 1 was measured. Also the acoustic results got worse with the cooler. Only 2 dB(A) instead of 4 dB(A) noise reduction were observed. Measurements inside the bus have shown that the temperature could be reduced by 3 K due to the increased airflow of fan model 1. However, the noise reduction was only 1.3 dB(A).



(a): Static pressure.

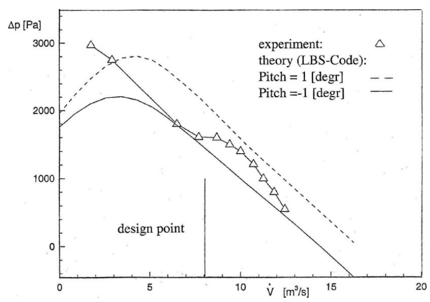


(b): Efficiency.

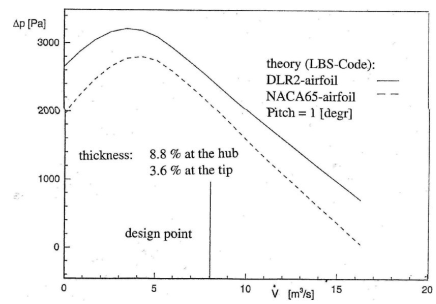
Figure 5: Performance of the reference fan and fan model 1 with cooler at different slit sizes.

4 - ACOUSTIC DESIGN AND COMPARISON OF EXPERIMENTAL RESULTS

To reduce the noise by further 2 dB(A) another laminar airfoil was selected. This airfoil was numerically designed especially for fans and has less camber than the NACA 65 airfoil [1]. The difference in camber was added to the twist to obtain the same performance. Then the Lifting Surface Code LBS was validated with the experimental results of model 1 and applied for computing the static pressure of model 2 with the new airfoil (see Fig. 6).



(a): Validation of the LBS-Code.



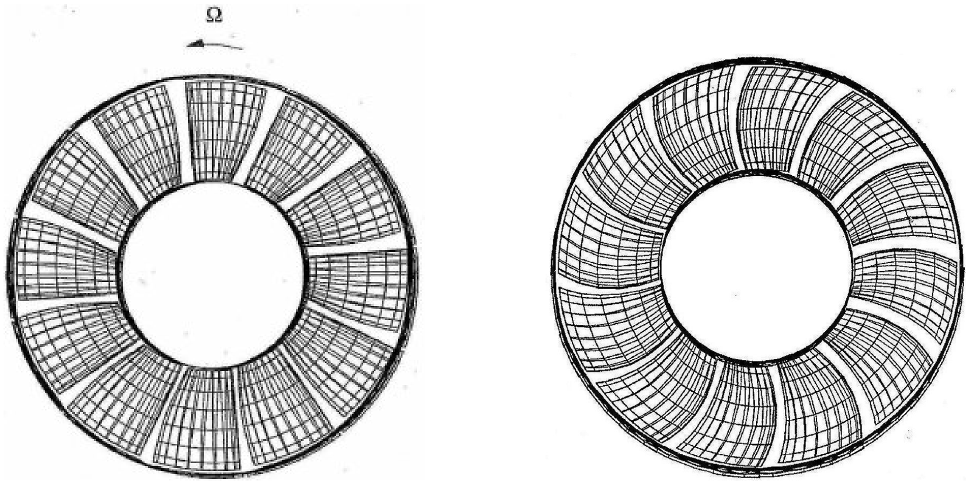
(b): Computation of static pressure curves.

Figure 6: Performance computations with the DLR Lifting Surface Code LBS.

In a second computation the planform was optimized in aeroacoustics to obtain a further noise reduction of 3 dB(A) by sweep (see Fig. 7). However, this optimized planform was not investigated due to the strong centrifugal forces which appear up to 1000 kp at an rpm of 2600.
Fig. 8 shows, that the fan module 2 is aerodynamically equivalent to module 1 at the design point. Outside the design point the efficiency of module 2 is very much smaller.
The reason for the losses at lower flow rates is probably a bubble on the upper side of the airfoil as could be shown by laminar RANS CFD-computations [3], (see Fig. 9).
The acoustic results of module 2 confirm the earlier results of the DLR2-airfoil: Fan model 2 shows a noise reduction of 4 dB(A) for both, with and without the cooler (see Fig. 10). From simple analyses it follows − under consideration of the experimental values − that the noise level of the cooler is 11 dB higher than the noise level of the fan. Therefore, in the case of a module the noise reduction at the fan has to be more than 4 dB(A). This indicates that the fan model 2 generates another mass flow which reduces the noise of the cooler.
To reduce the peak level at the blade passage frequency, the position of the fan model 2 inside the duct was changed till a minimum noise level at the fan location L = 13 mm was found (see Fig. 11).
The difference between the minimum tone noise level and the noise level at L= 33 mm is about 8 dB. This reduces the Overall Noise Level by about 2,5 dB(A) as can be seen in Fig. 12a. Thus, all together the total noise reduction of fan module 2 is more than 6 dB(A). On the other side, the change of the performance with L is relatively small and can be accepted. E.g., the static pressure difference between the two different fan positions of L is 6 % only (see Fig. 12b).

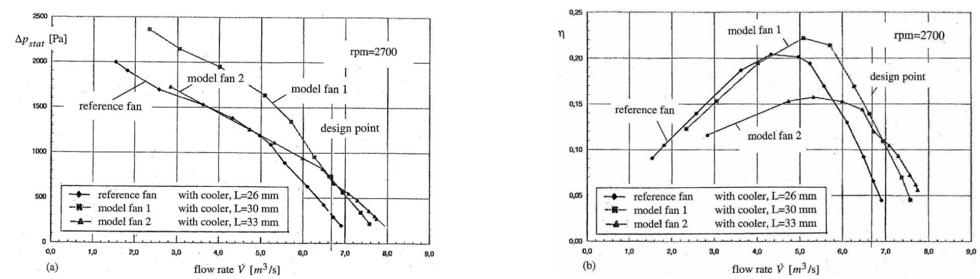
5 - CONCLUSIONS

A test rig was put up which fulfills the requirements of DIN 24 136/2. To improve the efficiency and the aeroacoustics of a cooling fan for busses two full-scale model fans were designed and measured. The 1^{st}



(a): Fan model 2 with DLR2-airfoil.
(b): Fan model 2 with optimized sweep.

Figure 7: Geometry modifications for noise reduction.



(a): Static pressure.
(b): Efficiency.

Figure 8: Performance of the fan modules − reference fan, model 1 and model 2.

model was designed with the radial equilibrium method. As a module it delivers 15 % more flow rate at the design point which increases the engine cooling temperature by 3 K. However, because of the high flow rate the cooler is about 11 dB(A) more noisy than the fan. Therefore, the module is still to noisy by 2 dB(A). To improve the acoustics of the fan, the NACA65 airfoil was exchanged by a proven laminar DLR airfoil for fans with less camber. The loss in camber was compensated by an according increase of the twist. Finally, at the design point the module 2 shows the same aerodynamic performance as module 1 but the noise was reduced by more than 4 dB(A), probably caused by a better mass flow. Further 2,5 dB(A) noise reduction could be obtained by optimizing the location of the fan inside the duct. Because of the high flow rate the efficiency could be increased by 5 % only. The benefit was obtained due to a better fan blade design and halving the size of the slit.

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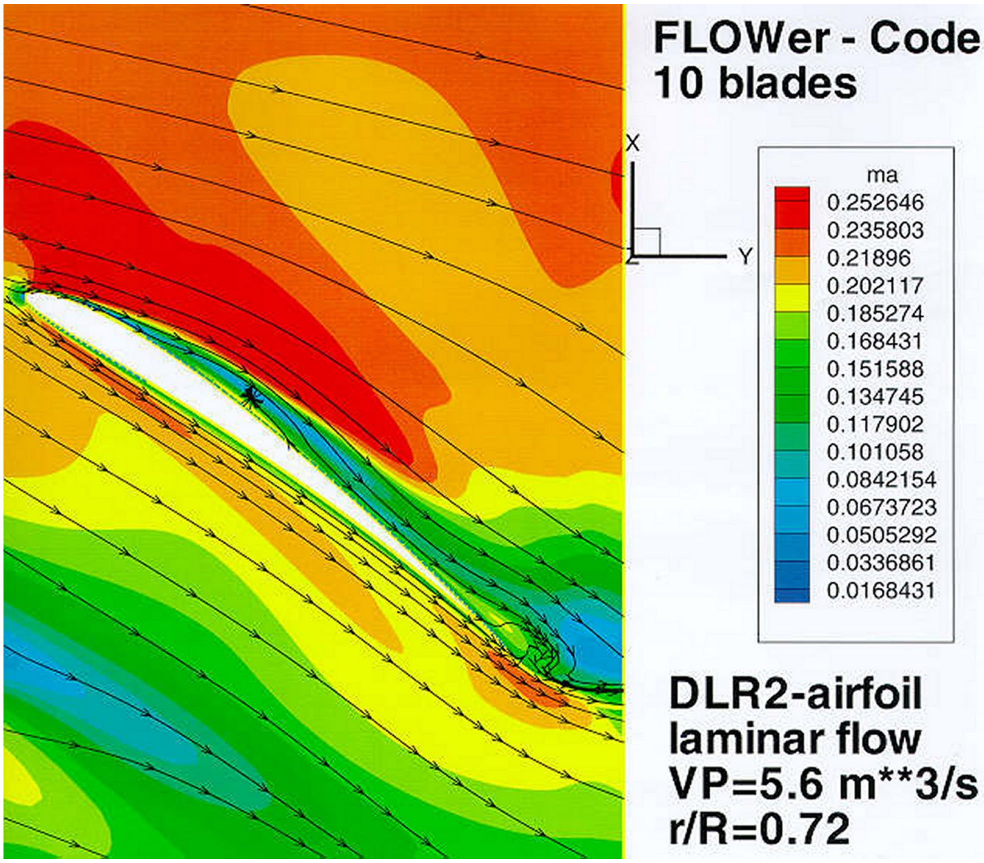

Figure 9: Bubble causing losses at 5.6 m^3/s.

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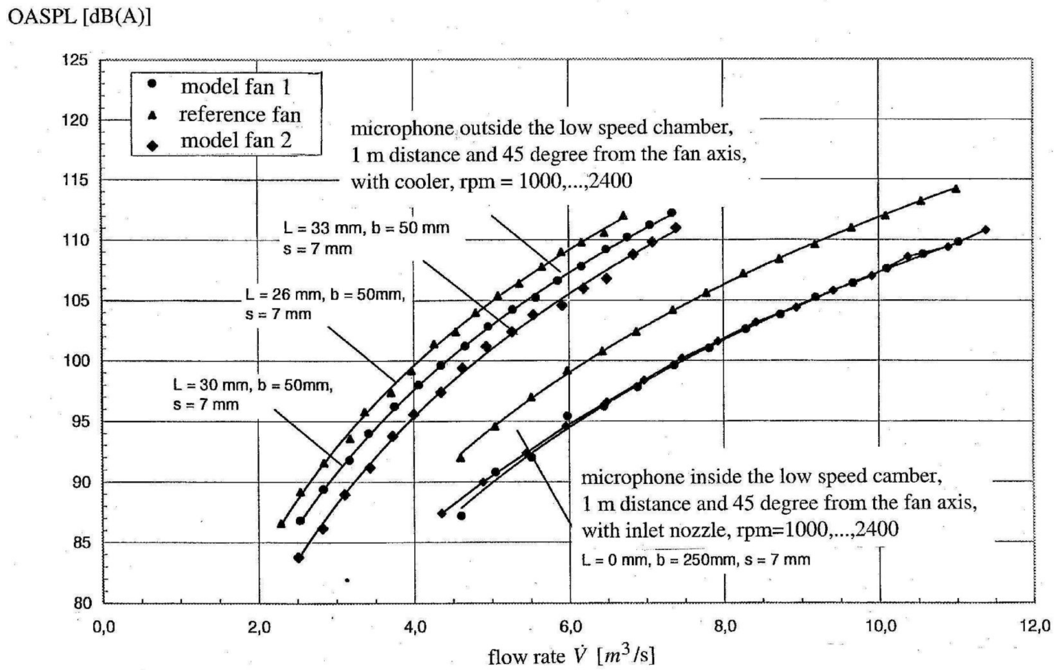
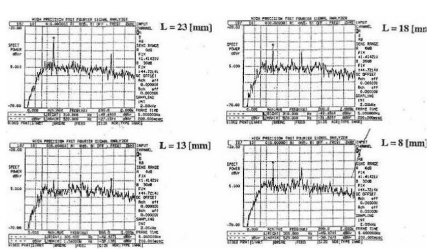
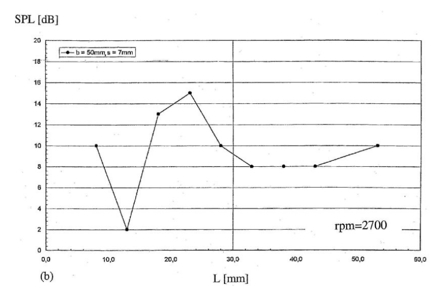


Figure 10: OASPL of the 3 models.

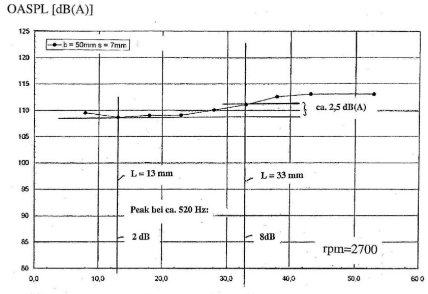


(a): Spectra.

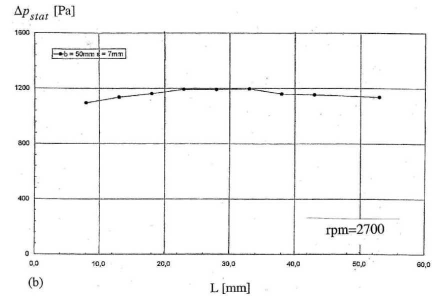


(b): Peak level of BPF over L.

Figure 11: BPF noise level at different fan positions L.



(a): OASPL variation over L.



(b): Static pressure variation over L.

Figure 12: OASPL and static pressure at different fan positions L.