Experimental and Theoretical Investigation of Noise Effect in Centrifugal Fan Impeller

Asst. Prof. Dr. Manal Hadi Saleh
University of Baghdad/College of Eng.
Manalhadi2005@yahoo.com

Asst. Prof. Dr. Muna Sabah Kassim
University of Al – Mustansirya/College of Eng.
munahdr@yahoo.com

Amina Hmoud Hnaef
Mechanical Engineering
Sahmali46@yahoo.com

ABSRACT

In this work a study was made in centrifugal fan blower to investigate the effect of impeller blade design on sound pressure level (SPL). Shroud and unshroud impeller of nine blades are used. The sound generation from flow inside the test rig at different positions was displayed by using spectral analyzer. The experiments were carried out in anechoic chamber with small holes in its walls, under ambient condition about (25-27) C ° to avoid the effect of temperature on the sound pressure level. The results showed that (SPL) decreased with the increase of distance from the source about (3-4)dB when distance varied about (0.8-1.06)m, and the (SPL) decreased with the decrease of velocity about (8-12)dB when velocity varied between (13000-2600) r.p.m., and when the velocity remain constant (SPL) increased with the increased of pressure about (7-15)dB when the pressure varied between (36-8)mbar. For the purpose of comparison, two types of impellers were tested under same conditions, the results showed that (SPL) increased when shroud used on the impeller. The mathematical results show good agreement with the experimental results. The study also concluded a spectral analysis of the noise generated using 1/3 octave band filter. The analysis showed that (SPL) increased with frequency range of (0.8-400) Hz. The maximum sound pressure level was appeared clearly in the frequency range between 200 – 400 Hz .

Key words: centrifugal blower; noise; frequency; rotational speed; static pressure; flow rate

دراسة عملية ونظرية لدراسة ظاهرة الضوضاء في منفاخ الطرد المركزي

أ.م.د. منال هادي صالح
قسم هندسةالميكانيك
كلية الهندسة / جامعة بغداد

أ.م.د. منى صباح قاسم
قسم هندسةالميكانيك
كلية الهندسة / الجامعة المستنصرية

آمنة حمود ضايف
قسم هندسةالميكانيك
كلية الهندسة / جامعة بغداد

الخلاصة

في هذا البحث تمت الدراسة على مروحة ذات الطرد المركزي لدراسة تاثير تغيير تصميم ريش بشارة المروحة على مستوى ضغط الصوت الصادر منها. حيث تمت الدراسة على نوعين من البشارات مغطاة وغير مغطاة ذات 9 ريش. تم قياس الصوت المتولد من الجريان داخل غرفة العزل باستخدام محلل طيفي. التجارب اجريت في غرفة عزل صوتي تحوي على ثمان ثقوب مخروطية على جدرانها الخارجية، الغرفة تعمل تحت تاثير الضروف الجوية بدرجة حرارة تتراوح ما بين 25−27 م، تم العمل بهذه الضروف لتجنب تاثير درجة الحرارة على مستوى ضغط الصوت(م. ض. ص.). النتائج اثبتت ان مستوى ضغط الصوت يقل مع زيادة المسافة عن المصدر الصوتي بنسب تتراوح م بين(3−4) ديس بيل ولابعاد تختلف ما بي 0.8م − 1.06م عن مصدر الصوت، ويقل م.ض.ص مع نقصان السرعة بنسب تتراوح ما بين(8−12) ديس بيل عند تغير السرعة ما بين(13000−2600) دورة لكل دقيقة، وعند بقاء السرعة والعوامل الاخرى ثابتة فان مستوى ضغط الصوت يزداد مع زيادة الضغط الاستاتيكي وهذه الزيادة ما بين(7−15)

dB عند تغير الضغط من 8ملي بار الى 36 ملي بار. لغرض المقارنة نوعين مختلفة من البشارات تم استخدامها تعمل عند نفس الضروف، النتائج اثبتت ان مستوى ضغط الصوت يزداد عند استخدام البشارة المغطاة. النتائج النظرية اعطت توافق جيد مع النتائج العملية ، الدراسة ايضا اعطت التحليل الطيفي للضوضاء المتولدة باستخدام مرشح(octave3/1). التحليل بين ان زيادة م.ض.ص مع التردد للقيم مابين 0.8 –400هيرتز . القيمة العظمى لمستوى ضغط الصوت تتحقق عند الترددات ما بين 200–400 هيرتز.

1. INTRODUCTION

In industrial ventilation applications, noise can be a significant concern. High acoustic levels promote worker fatigue. The noise generated by a fan depends on fan type, airflow rate, and pressure. Inefficient fan operation is often indicated by a comparatively high noise level for a particular fan type. If high fan noise levels are unavoidable, then ways to attenuate the acoustic energy should be considered. Centrifugal turbo machines are common devices used in many flow control applications due to their ability to achieve relatively high-pressure ratios in a short axial distance compared with axial fans. They are often found in gas turbine engines, heating ventilation and air-conditioning systems and pumps. Because of their widespread use, the noise generated by these machines causes one of the serious problems. The noise is often dominated by tones at the blade passage frequency and its higher harmonics. This is a consequence of the strong interactions between the flow discharged from the impeller and the cut-off in the casing. In addition to the discrete tones, the broad band noise is generated from the trailing edge due to the fluctuations of the turbulent boundary layers or separated flows on the impeller blade, **Jeona and Leeb, 2002.**

Acoust, J., 1978, studied the Sources of broadband noise in FC (forward curved blades) centrifugal fans experimentally using a 0.28m diameter fan. Strong tangential and axial gradients of mean velocity and static pressure were found in the housing. Very large variations in velocity and velocity fluctuations occurred across the blade passages. Both the inlet and discharge rotor noise showed a continuous reduction in level with flow from maximum flow to the stall region. A miniature microphone mounted at various points on the blade surfaces showed a large increase in pressure fluctuations from the leading to the trailing edge of the blades. The fluctuations on the suction surface were much stronger than those on the pressure surface. **Choi, 1993,** Identify which aspects of the fluid dynamics were associated with noise generation in centrifugal turbo machinery. Research emphasis was placed on the generation of noise at frequencies other than the blade passage tones. In order to avoid noise generated by the interaction of the discharged flow and stationary objects outside of the impeller, experiments were performed on a centrifugal impeller without diffuser and casing. With this discharge configuration, the radiated noise spectra were show to be dominated by harmonically related broad humps at low frequency. These were proven to be generated by the interaction of a coherent unsteady flow structure rotating around the impeller discharge and the trailing edges of the impeller blades. **Xiaoliang, et al., 2008,** reduce the noise of the T9_19No.4A centrifugal fan, whose impeller had equidistant forward swept blades, two new impellers with different blade spacing were designed and an experimental study was conducted both the fans aerodynamic performances and noise were measured when the two redesigned impellers were compared with the original ones. The test results were discussed in detail and the effect of the noise reduction method for a centrifugal fan using impellers with no isometric forward swept blades was analyzed which could serve as a reference for researches on reduction of fan noise. **Wolfram, et al., 2011,** studied a typical acoustic spectrum of a fan consists of both broadband and tonal components; the overall acoustic level was dominated by the tonal part, especially the tone at blade passing frequency (BPF) and higher harmonics. **Hsien, et al., 2011,** presented a new robust multi—criteria optimization method that employs the Taguchi method to design and analyze an ultra-thin centrifugal fan. The objective of this study was to obtain the maximum volume flow rate, static pressure, and minimum noise of a fan. The proposed approach utilizes a combined orthogonal array and computational fluid dynamics method to simulate the internal flow field of the ultra-thin centrifugal fan. This study employs signal-to-noise ratio and analysis of variance to determine the Pareto-optimal robust design solution. This research also identifies the optimal design parameters that affect the cooling performance of the centrifugal fan. The experimental results confirm the effectiveness of this approach The Taguchi method is a highly practical tool for process design, wherein mathematical models for system performance do not exist. The method provides a simple, efficient, and systematic approach to optimize designs for performance, quality, and cost. The Taguchi parameter design can optimize performance characteristics by setting design parameters, and reduces the sensitivity of system performance to the source of variation. **Sasaki, 2012,** presented a preliminary attempt towards the prediction of the broadband noise from the flow features, compatible with industrial constraints. In the case of low-solidity impeller, the wake rapidly expands in a wide outer part of the blades under the influence of the separation forced by the tip vortex. The broadband noise level in the low-frequency domain became large because the wake vortices with large scale in the low frequency domain were shed from the blade. Since the relative flow of the high-solidity impeller at the maximum efficiency point remains attached over the blades, the strength of vortex-shedding in the wake was reduced. Therefore the broadband noise at the maximum efficiency point was substantially decreased. At the off-design point in low flow rate, the number of blades had limited influence on the flow regime in the wake because separation likely occurs from the leading edge.

2. EXPERIMENTAL DEVICE

In this work, an experimental study about the tonal noise sources in a centrifugal fan with backward curved blades had been carried out. Acoustic pressure measurements at the blower exit duct and pressure fluctuation measurements on the volute surface had been made for different flow rates. A correlation study of both pressure signals has been made in order to explain some of the features of the aerodynamic tonal noise generation. A strong source of noise caused by the interaction between the fluctuating flow leaving the impeller and the volute tongue is appreciated. The unsteady forces exerted on the fan blades constitute another noise generation mechanism, which affects the whole extension of the impeller, thus transmitting pressure fluctuations to the entire volute casing. The relative importance of this mechanism compared to the impeller–tongue interaction depends on the flow rate. The fan used in this study is a single-stage machine with shrouded and unshrouded impeller and external volute.

2.1 Test Rig Equipment

The Rig consists of the following main parts as shown in **Fig. 1**.

2.1.1 Anechoic Chamber

Anechoic chamber is a room that has been prepared to minimize sound reflections from walls. The Anechoic Room is used to prevent the undesirable outside noise to effect the test. However, a free-field microphone can be used, **Barlow, et al., 1999, and Holman and Gajda, 1984**.

The anechoic chamber is shown in **Fig. 2** with the size of 1.5m long by 1.0m wide and 0.9m high. It consists of four parts:

1- Wooden structure of 3.5 cm thickness.
2- External wooden cover has a thickness of 8mm.
3- Sheets of cork material inside the wooden structure have a thick of 3.5 cm.
4- Triangular sponge was used to distribute on all sides, ceiling and floor of the room made from wood. Each sponge has base (10cm x10cm) and height 10 cm as shown in **Figs. 3 and 4**.

There are 8 holes. Each hole is in the form of cone have a length of 5cm and two diameters the first diameter is 2.5cm and the second diameter is 1.4cm, was used to put the device of (**SPL**) for measuring purposes.

2.1.2 Pipes

There are two types of pipes used in this work:

1- Rigid pipe: it is a pipe of 6.25 cm of diameter and 2.5m length as shown in **Figs. 5 and 6**. An orifice meter was fixed on the entering pipe with manometers in each side of the orifice meter to measure the determined pressure across the orifice meter for all the cases studied. A gate valve was fixed in the existing pipe to get a specified mass flow rate inside the system.

2- Flexible pipe: it is a pipe of 6.25cm of diameter, and 25cm length as shown in **Fig. 7**. this pipe was used because of its flexibility to insulate and absorb the blower vibration.

2.1.3 Blower

The blower model ct 6007of unshrouded rotor type shown in **Fig. 8** was used in this work; of (a centrifugal type) which has different speed, and the specification of the blower is shown in **Table 1**.

2.1.4 The Impeller

Two type of impellers used for the test shrouded and unshrouded. It is fitted on the rotating shaft which is directly connected to the motor shaft. The blades of this impeller are of backward shape having a thickness of (3mm).

2.2 Measuring Instruments:

2.2.1 Sound level meter

Sound pressure level is the measurement of sound strength on a logarithmic scale (base ten). The sound level meter is shown in **Fig. 9** and has an auto range Rs-232 type K/J.

2.2.2 Sound and vibration analyzer

The **SVAN 957** is digital, Type 1 sound and vibration level meter along with analyzer. The instrument is intended to general acoustic and vibration measurements, environmental monitoring, occupational health and safety monitoring as shown in **Fig. 10**.

2.2.3 Digital manometer:

A p200 UL model of a digital device, used to measure the pressure inside the inlet pipes before, and after orifice plate, as shown in **Fig. 11**, its range are: Low pressure 0_19.99 mbar and High pressure 0_100 m bar.

2.2.4 Interface temperature

The thermocouples used in this experiment were type K and insolated. These average values of the temperature were taken for each flow and pressure measurement as shown in **Fig. 12**.

2.2.5 Digital photo tachometer

This digital photo tachometer was used to measuring the velocity of the blower impeller for each flow and each pressure measurement as shown in **Fig. 13** with the applied target.

EXPERIMENTAL STEPS:

1. Operate the blower at a specified speed
2. Open the gate valve (full open)
3. Record the thermometer reading
4. Use the manometer to measure the maximum pressure at that blower speed.
5. By controlling the gate valve, take four readings of different pressures where the last reading is for the maximum pressure.
6. Record the spectrum sound analysis using the (Svantek pc++) device in the eight openings of the anechoic chamber for each pressure.

Repeat steps (1- 6) for five blower speeds.

3. MATHEMATICAL MODEL

The backward inclined blower wheel design has blades that are slanted away from the direction of wheel rotation**, Kunjur and Krishnamurty, 1997**. The term applied to this type of balding is BI or backward inclined. Centrifugal fans are widely used and the noise generated by these machines causes one of the serious problems, **Beranek, 1960**.

In order to calculate the radiated acoustic field of a centrifugal fan, the modification of the generated noise by the casing should be considered. The unsteady flow is generated due to the rotation of the impeller at a rotation velocity N near a wedge. For the centrifugal fan, the rotation of the impeller results in an inflow across the inlet section. Let Q denote the volume flow rate, represented by a source located at the center of the impeller. Assuming the fluid is incompressible and inviscid, the main feature of the flow considered is the non-zero circulation around every impeller blade. The impeller was assumed to rotate with a variable angular velocity and the flow field of the impeller is incompressible and inviscid. The impeller has (b) number of blades. The inlet flow is modeled by a point source located at the center of the impeller. A major component of fan noise for the large commercial fan currently in service is the tone noise generated by the rotating blades of the fan, as they interact with the stator blades and the struts**, Faulkner, 1976 and Bartlett, 1934.**

Sound pressure level, SPL, is defined by the relation**, Faulkner, 1976.**

$$SPL = 10\log\,[p/p_{ref}]^2 \quad (1)$$

Where:

$P_{ref} = 2 \times 10^{-5}$ r.m.s sound pressure (Pa)

Or

$$SPL = 20\log\left[\frac{p}{p_{ref}}\right] \quad (2)$$

Since the wave fronts generated with each pulsation are always in phase, the resultant wave motion diverges uniformly in a spherical manner. Now, as seen previously for uniform spherical divergence, the sound intensity I at a distance r is given by:

$$I = w/4\pi r^2 \quad (3)$$

Where:

w= acoustical power of the radiating source (W)

$4\pi r^2$ = surface area of a sphere of radius r (m^2)

In addition, it can be shown that for freely propagation plane waves, the intensity are related to the rms sound pressure as follows:

$$I = p^2/\rho c \quad (4)$$

Combining the equations, to get finally, the relationship between sound pressure and sound power:

$$p^2/\rho c = w/4\pi r^2 \quad (5)$$

With a little algebra, the more useful relationship between sound pressure level (SPL) and sound power level L_w is obtained:

$$SPL = L_w - 20\log_{10}(r) - 11 \quad (6)$$

Where

L_w = sound power level of the point source (re10^{-12} W)

r = radial distance from source (m)

The frequency of the discrete tones is given by:

$$F_n = N*b/60 \quad (7)$$

The origin of the discrete tones results from two sources. The first, for each time a blade passes appoint in space, a pressure fluctuation is created due the displacement of air and/or aerodynamic lift if the blade is in airfoil configuration. The broad band aerodynamic noise originates from vortices created at the leading and/or trailing edge of the blades and turbulence imparted to the fluid, usually in the form of eddy like flow. Here again the accurate prediction of noise levels for these fans is at best very difficult, but an empirical approximation which provides good first-order results for the average sound power level in the range of 500 to 400 Hz is:

$$L_w = 10\log Q + 20\log P_t + K \quad (8)$$

Where:

K=constant depending on fan type, 35 for forward -or backward-curved blades and 43 for radial types.

4. RESULTS AND DISCUSSION

4.1 Experimental Results

In this work the spectral analysis of noise emitted from centrifugal blower was studied for different distance from the sound source in the anechoic chamber and the effect of changing the centrifugal blower velocity, types and pressures on the sound pressure level (SPL) was studied.

4.1.1 Spectral analysis of noise at different distances from the source :

Fig. 14 Shows the variation of the sound pressure level versus the frequency in different distances around the anechoic chamber (eight holes), for blower have unshroud impeller with 9 blades, angular velocity of 13000 r.p.m and a pressure of 32 mbar. It is clear that the maximum value of the SPL is in the first position 1(The closest distance from the source) and the minimum value of the SPL is in the position of the hole at position 6 the farther position from the source) The peak amplitude of SPL at position 1and 6 can varied between (3–4) dB, for different types of impellers.

4.1.2 Spectral analysis of noise emitted for variable velocities:

Different types of impeller and different distances from the source was used to explain the effect of the variation of blower velocity on the maximum value of SPL in all cases, the results show that the increase of velocity caused increase of SPL in the range of (8– 12) dB depending on the value of velocity as shown in **Figs. 15 and16**.

4.1.3 Spectral analysis of noise for different pressures:

Spectral analysis of noise at different pressures was studied when other parameters were taken as constant. In all cases the only parameter changed was the pressure. , increasing pressure cause increase of the peak value of SPL at the range of (8-15) dB that depends on the value of specified pressure as shown in **Fig. 17**.

4.1.4 The Effect of the mass flow rate on SPL, and Reynolds number

Five values of velocity were taken and for each velocity the gate valve used to get four values of the mass flow rate. **Figs. 18 and 19,** show that the SPL decrease with the decrease of the mass flow rate.

4.2 Theoretical Results

A computer program was built to solve the sound pressure equation by using mat lab program. The data that controls program is volume flow rate, static pressure and the rotation velocity of flow. The mat lab program was used to simulate the noise propagation **Fig. 20 and Fig. 21** shows the relation between the SPL and position from the source at different velocity in shroud and unshroud impeller.

5. CONCLUSION

In this work many parameters were studied that affected the sound intensity caused by a centrifugal blower in anechoic chamber. The following major conclusions for the experimental and theoretical study can be drawn as follow:

1- An increase in the sound pressure level (SPL) with the velocity of the blower.

2- The noise resulted in the test section was in the frequency range of o.8 – 20000 Hz.

3- The sound pressure level increase with frequency for the range mentioned above.

4- The maximum sound pressure level was appeared clearly in the frequency range of 200 – 315 Hz.

5- The results show that using cover on the blades causes increase in pressure Increase the sound pressure level.

6- For the same number of impeller blades the flow velocity and pressure decrease for a light impeller material

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NOMENCLATUR

Letter	Description	Units
SPL	*Sound pressure level*	*dB*
Fn	*frequency*	*Hz*
p	*Static pressure*	N/m^2
I	*Sound intensity*	W/m^2
w	*Acoustical power of the radiating source*	*W*
ρc	*Mass flow rate per unit area*	$Kg/s.m^2$
Lw	*Sound power level of the point source (re10-12 W)*	*W*
r	*Radial distance from source*	*m*
N	*Fan rotational speed*	*r.p.m*
b	*Number of blades*	–
n	*Harmonic; i.e. ,n = 1(fundamental)*	–
Q	*Volume flow rate*	m^3/s
$\dot{m}$	*Mass flow rate*	*kg/s*

Table 1. Specification of the blower model ct 6007.

Radiated power input	600W
Outlet diameter	10 cm
Type of curved blade	Backward curved blades
Impeller exit diameter	104 mm
The inlet and outlet Pipes diameter	2.5cm
Impeller inlet diameter	15 mm
Number of impeller blades	9
Speed	13000 rpm
Inlet blade angle	β1= 54 from tangential direction
Outlet blade angle	β2=42°from tangential
Blade thickness	2mm
Impeller thickness	3 mm
Maximum thickness, discharge width	24 mm
Length of blade	520mm
pitch	30mm

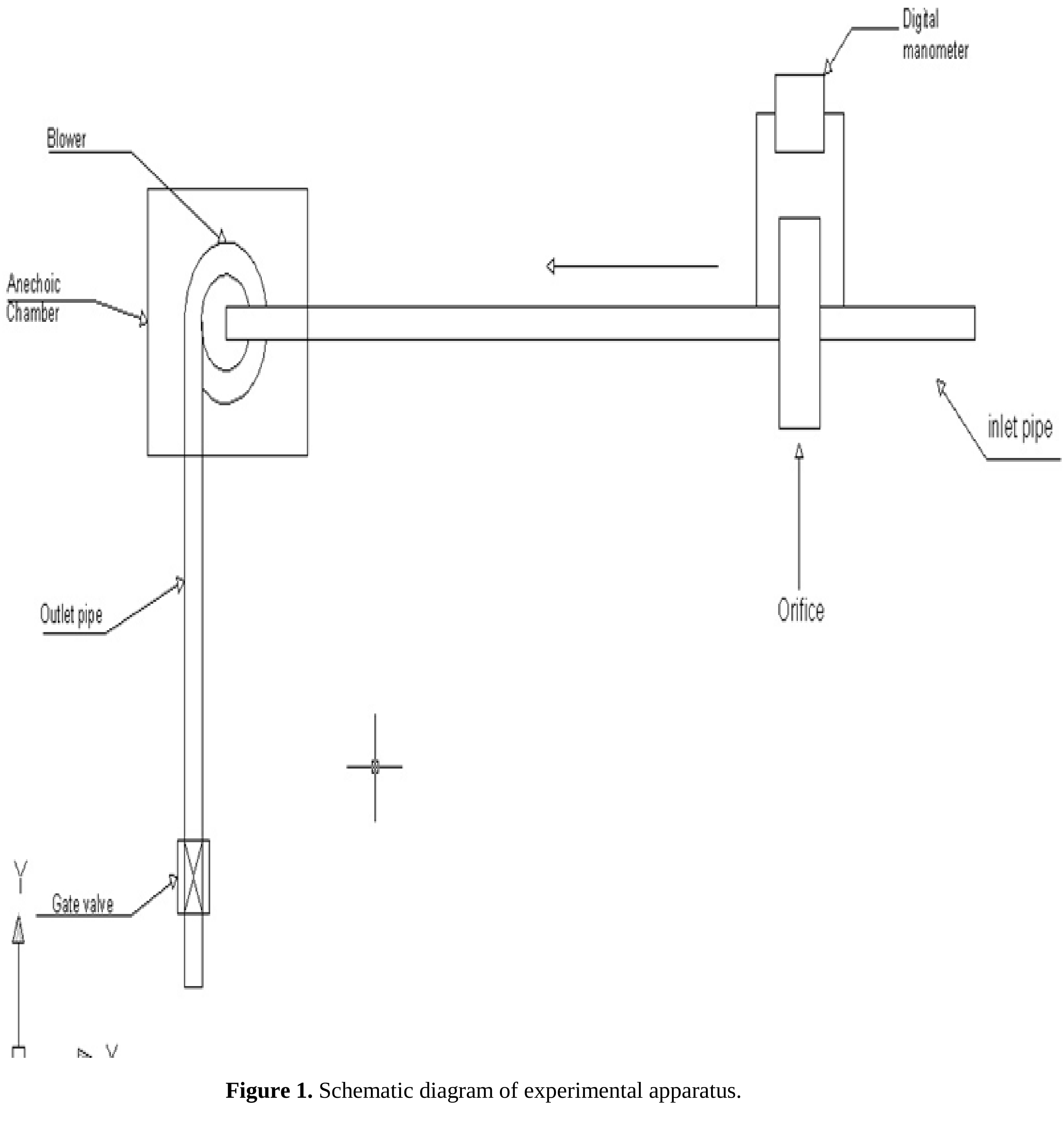


Figure 1. Schematic diagram of experimental apparatus.

Figure2. Outside of anechoic chamber.

Figure 3. Inside of anechoic chamber**.**

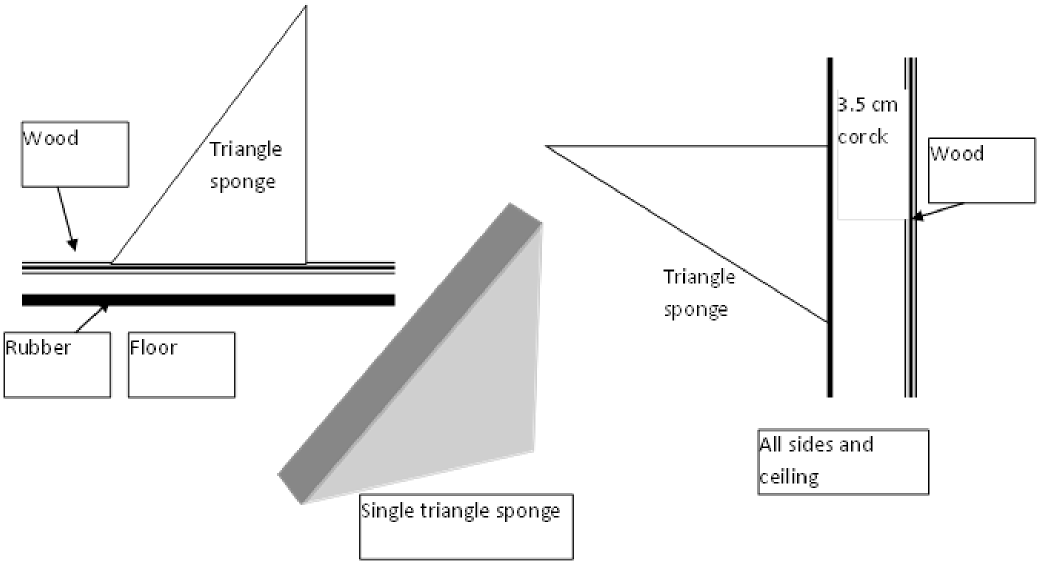


Figure4. Sponge distribution and dimensions.

Figure 5. Inlet pipes with orifice meter.

Figure 6. Outlet pipes with gate valve.

Figure7. Flexible pipes.

Figure 8. The centrifugal blower.

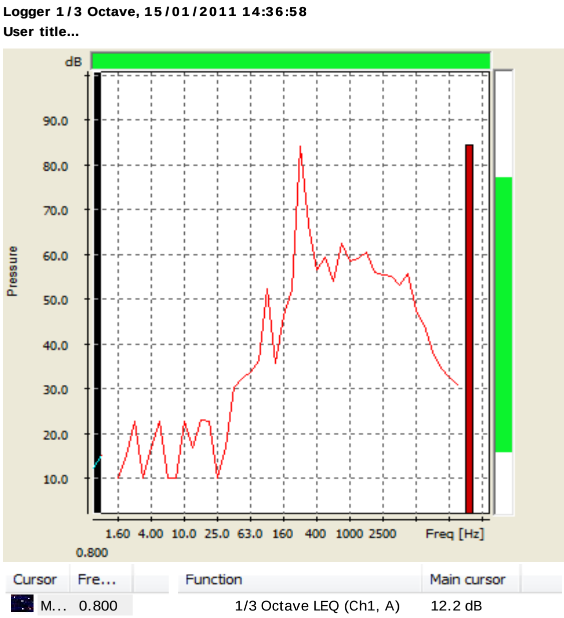
Fiure 9. Sound analyzer.

Figure10. Sound level meter.

Figure 11. Digital manometer.

Figure 12. Digital thermometer.

Figure 13. Digital tachometer.



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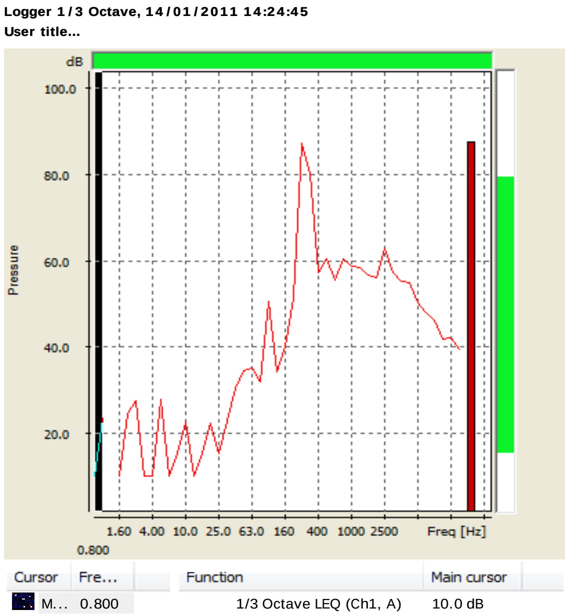

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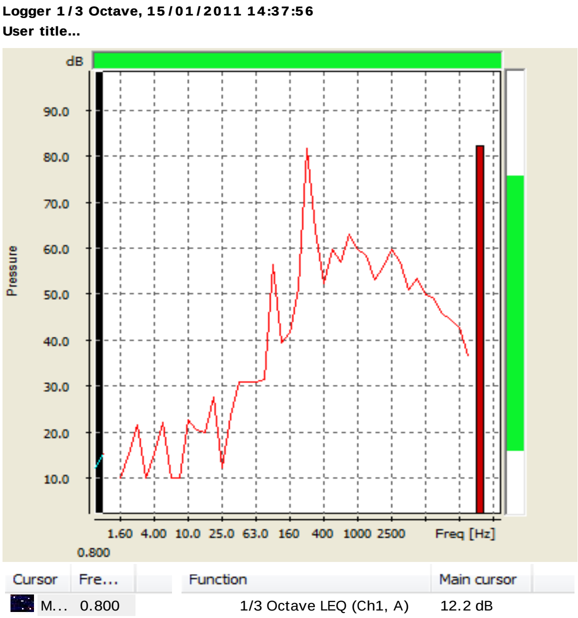

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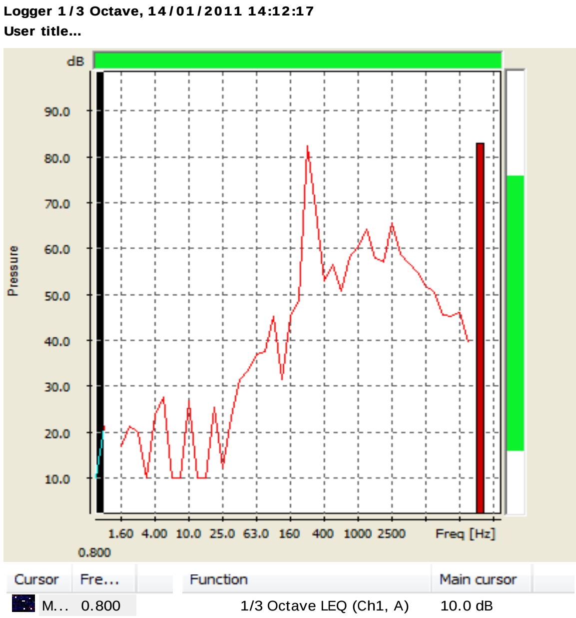

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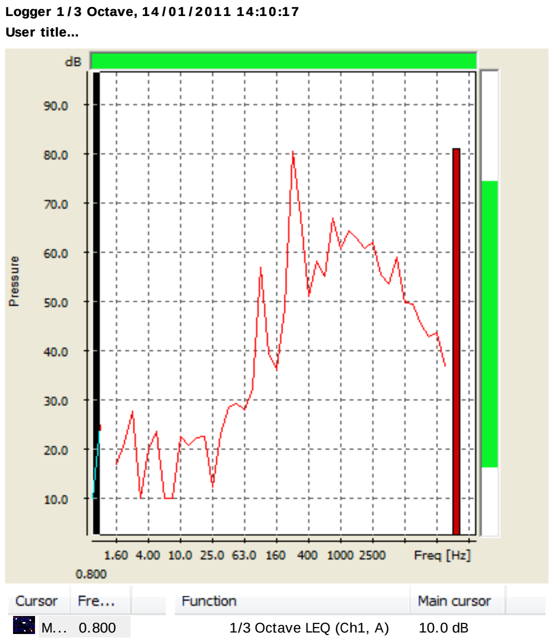

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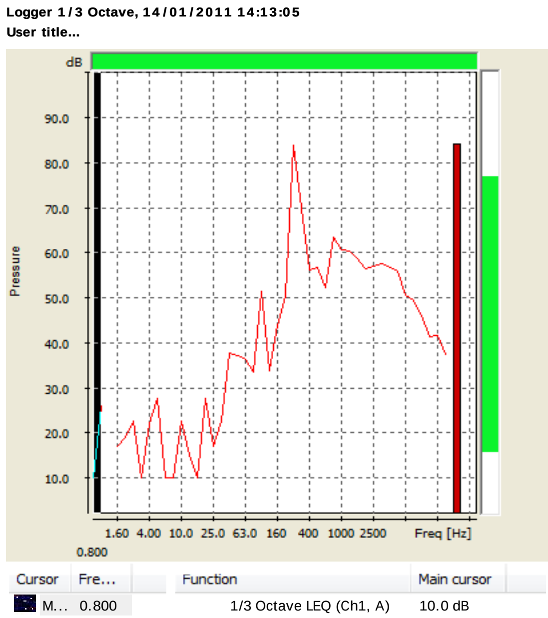

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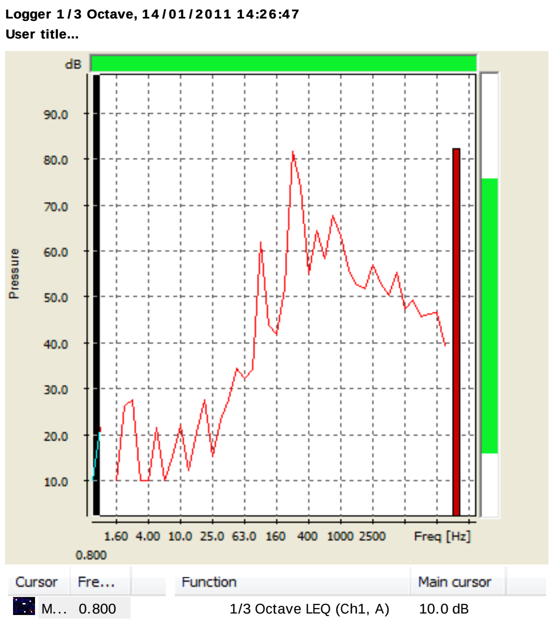

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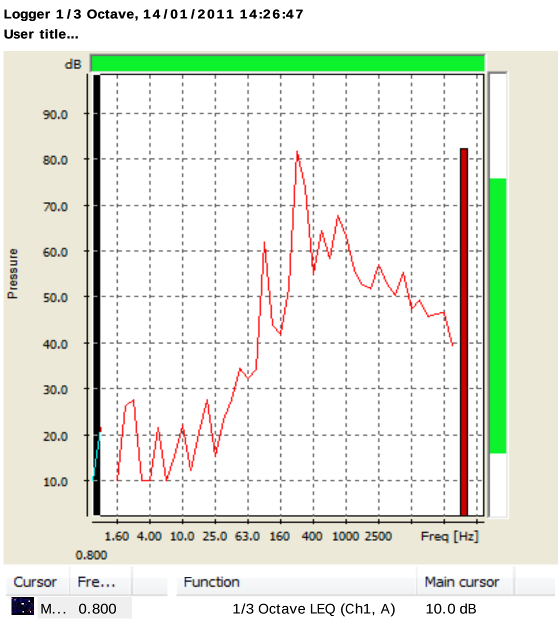
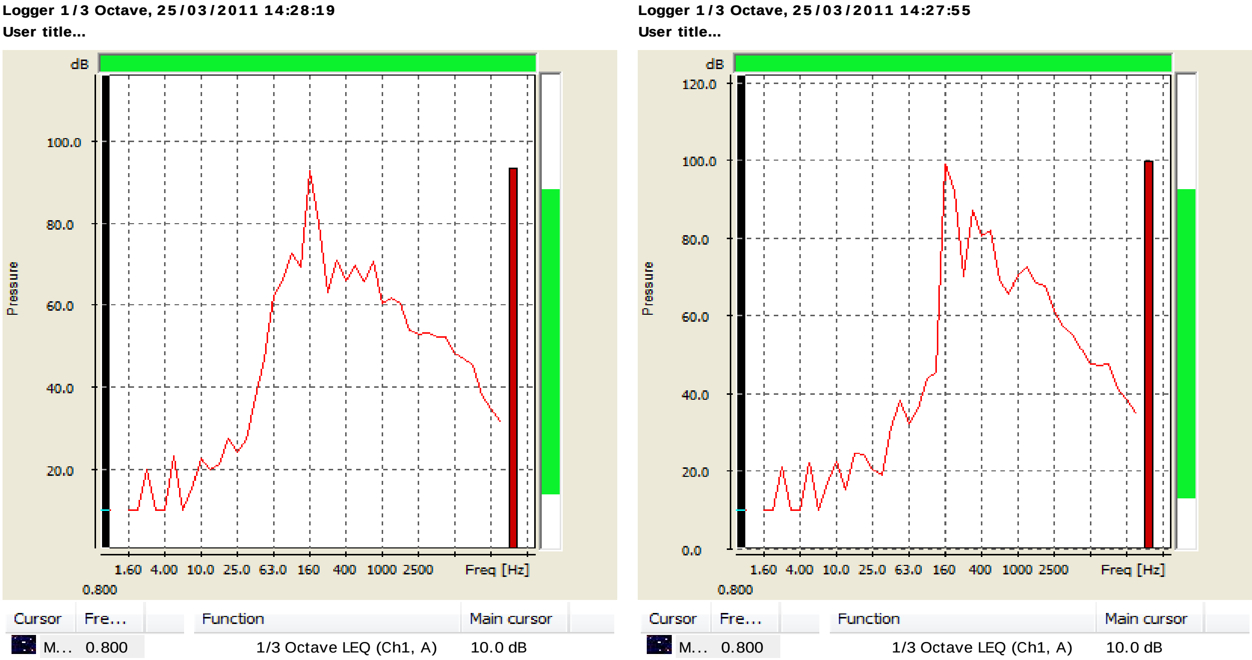


Position 7

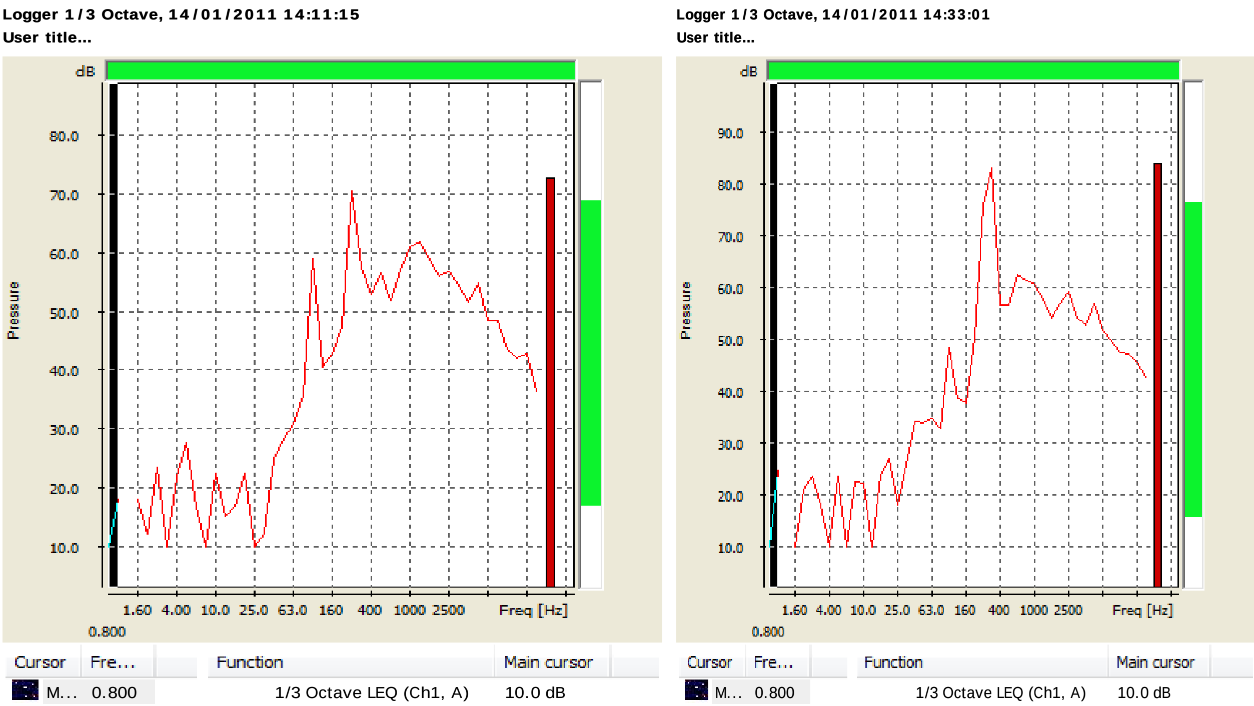
Figure 14. Spectral analysis of noise at unshroud of 9 blades impeller at eight positions around the anechoic chamber, velocity=13000 r.p.m and pressure=32mbar.



Velocity= 2600 r.p.m

Velocity= 13000 r.p.m

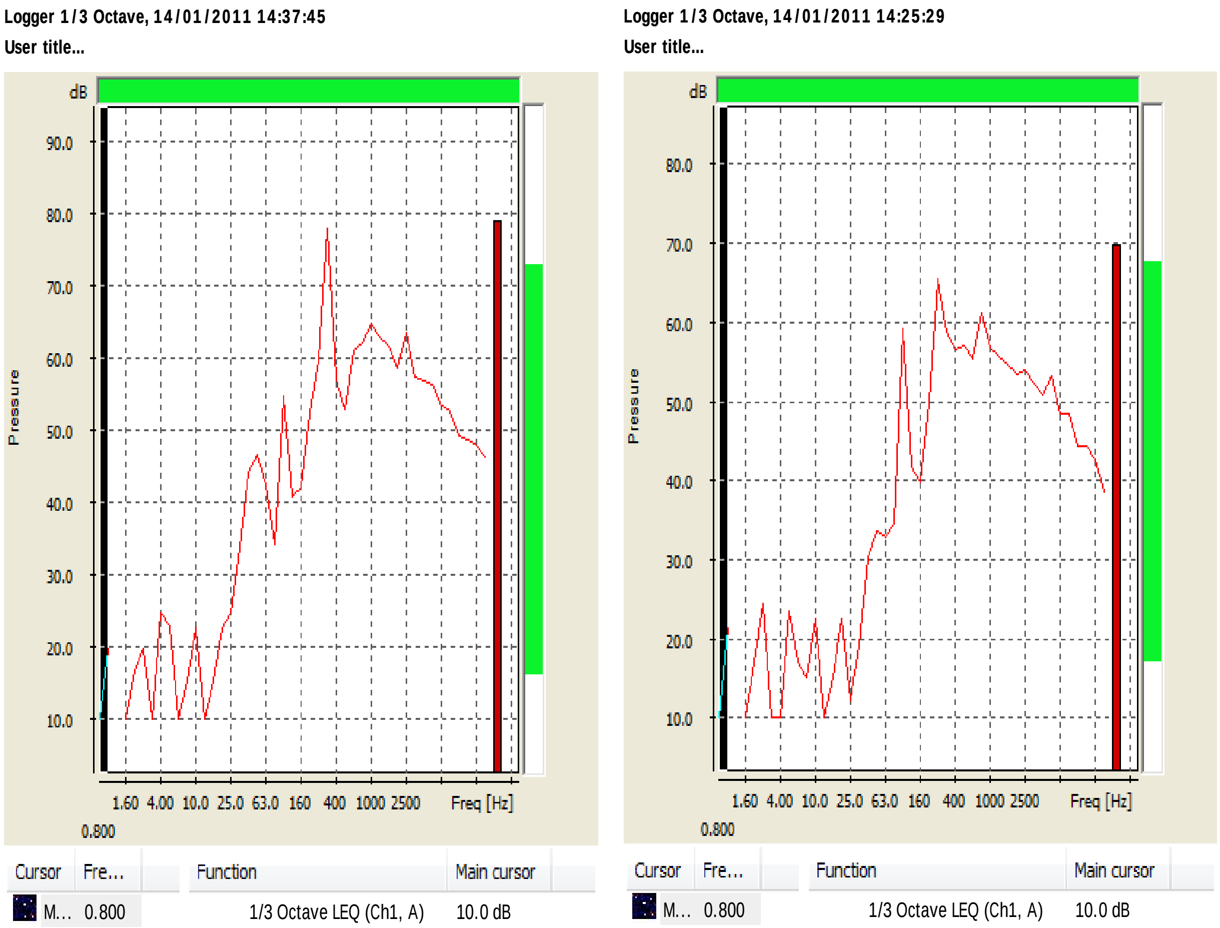
Figure 15. Spectral analysis of noise in shroud of 9 blades impeller at different velocity.



Velocity= 2600 r.p.m

Velocity= 13000 r.p.m

Figure 16. Spectral analysis of noise at unshroud of 9 blades impeller at different velocities.



Shroud impeller
Velocity= 5200 r.p.m
Pressure= 24 mbar

Unshroud impeller
Velocity= 5200 r.p.m
Pressure= 12 mbar

Figure 17. Spectral analysis of noise at different pressures

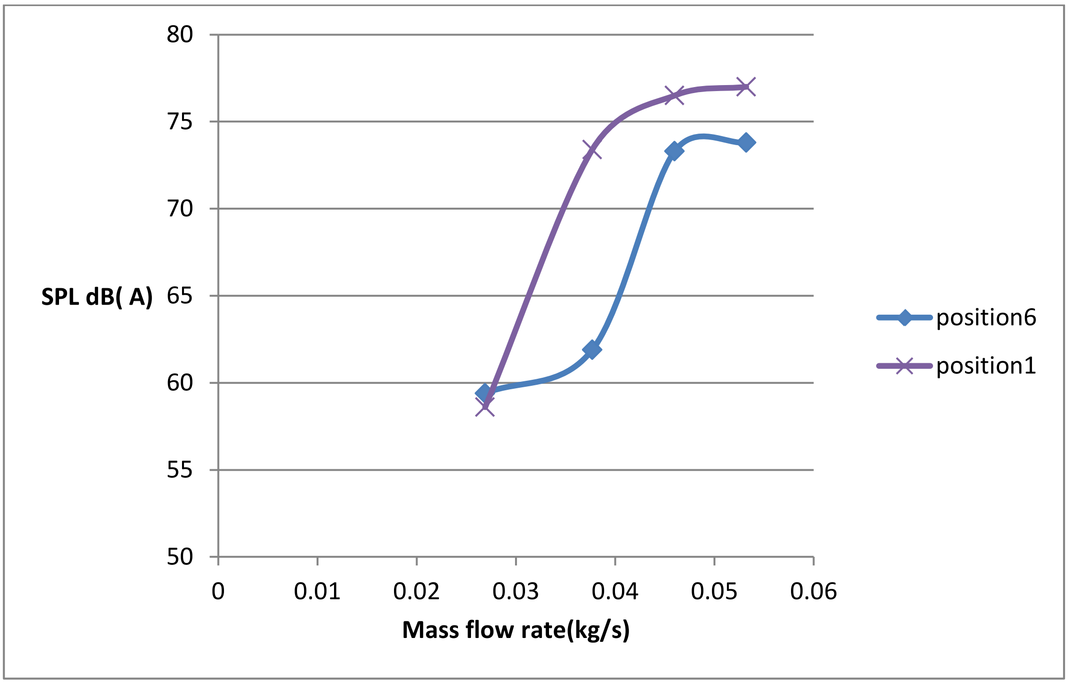


Figure 18. Variation of SPL with mass flow rate for unshroud impeller at velocity 5200 r.p.m.

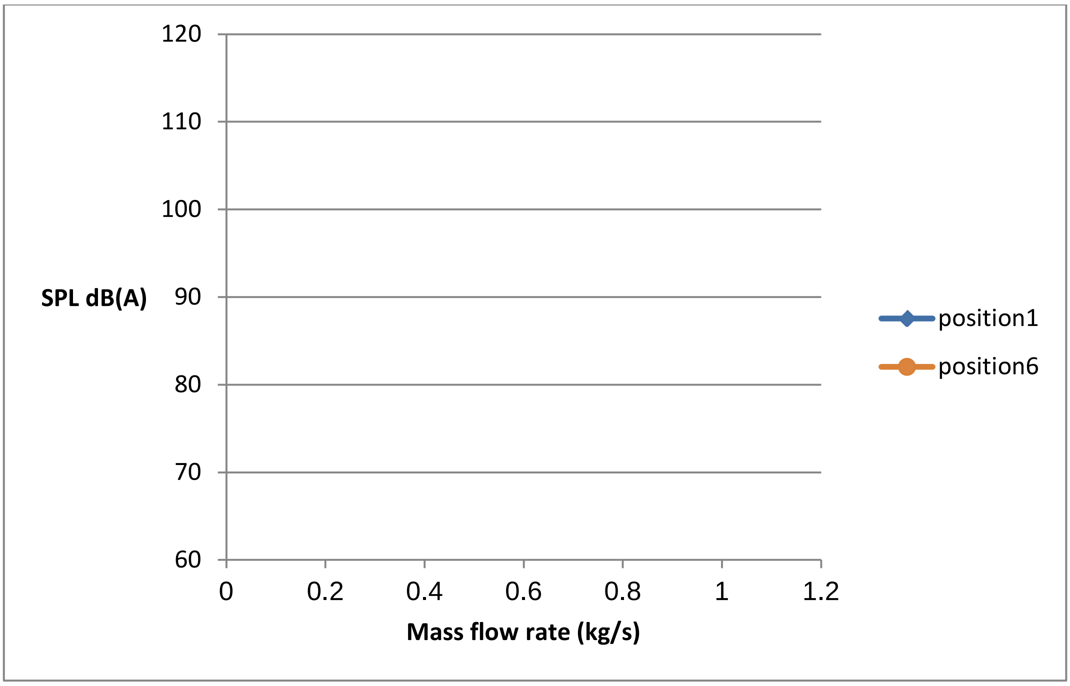


Figure 19. Variation of SPL with mass flow rate for shroud impeller at velocity 5200 r.p.m.

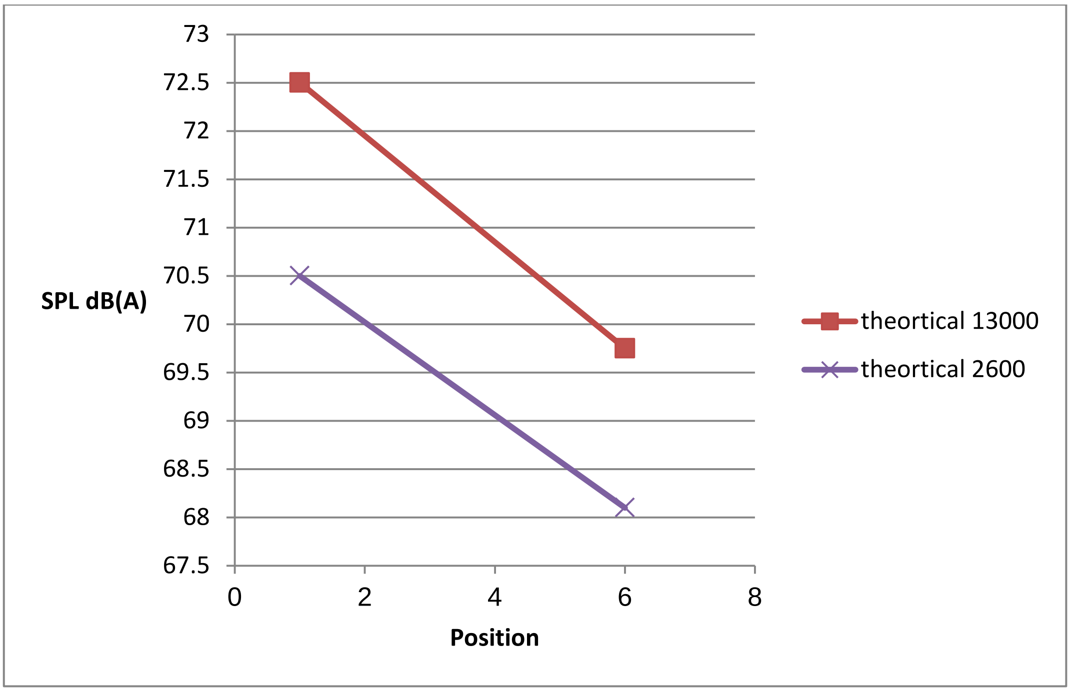


Figure 20. Variation of SPL with distance from the source in shroud impeller.

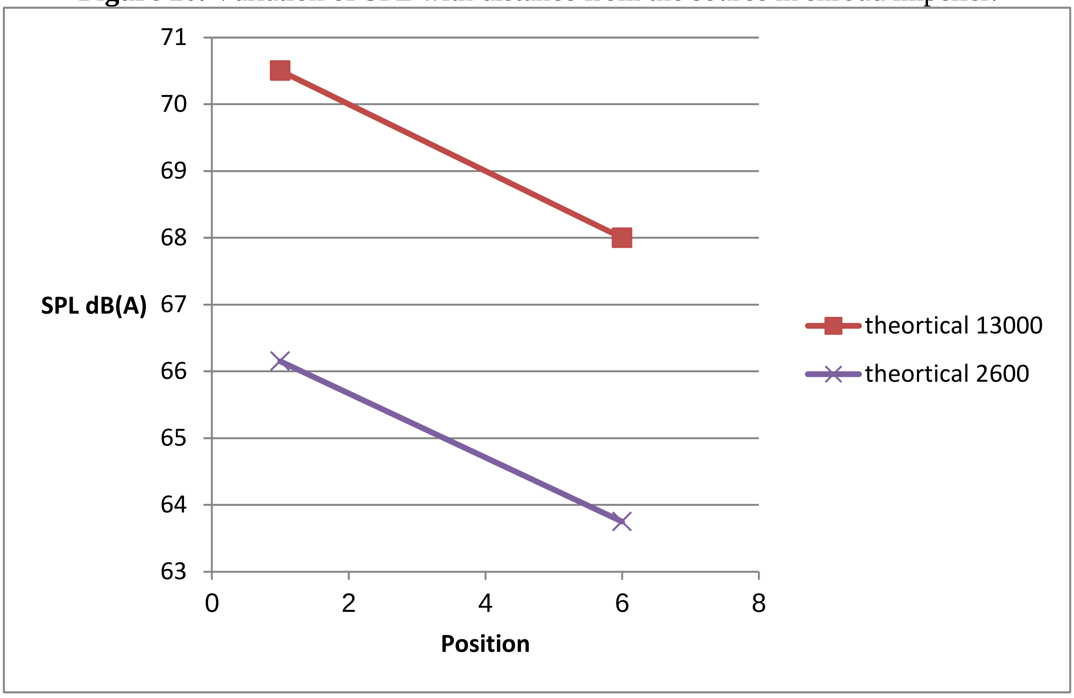


Figure 21. Variation of SPL with distance from the source in unshroud impeller.