Some aerodynamic and noise measurements on two centrifugal blowers

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SOME AERODYNAMIC AND NOISE MEASUREMENTS ON TWO CENTRIFUGAL BLOWERS

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SUMMARY

This report describes the aerodynamic performance and noise measurements conducted on two different commercial centrifugal blowers of 15 horsepower and 1/3 horsepower capacities. The main aim of these experiments was to relate the noise from these blowers to their aerodynamic performance.

The pressure versus flow characteristics of both the blowers were relatively flat with imperceptible to no stall and the noise characteristics were different. The prominent noise component for the 15 horsepower blower was the tone at the blade passing frequency and the 1/3 horsepower blower showed two less prominent tones which could be related to the struts supporting the blades and the Helmholtz resonance frequency of the casing. Low frequencies were prominent in the broadband noise levels.

Noise levels in each one third octave band varied with the flow coefficient and Strouhal number. The minimum noise levels for the 15 horsepower blower occurred at approximately the flow coefficient corresponding to the design point whereas the noise levels increased with the flow coefficient for the 1/3 horsepower blower with minimum levels at no flow and maximum pressure rise.

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1.0 INTRODUCTION

There is a clear need for controlling the noise from centrifugal fans and blowers, which are widely used from a fractional horsepower to several hundred horsepowers. Due to hearing safety and users' comfort it is becoming increasingly important to consider the acoustic performance of the fan in addition to its aerodynamic performance. Although the noisy units could be silenced by fitting silencers and mufflers, these would undoubtedly add to the cost and bulk of the air moving system and would probably affect the aerodynamic performance also. Therefore, there is a strong case for controlling the noise generation at the source itself by suitably designing the various elements of the fan without sacrificing the aerodynamic performance of the fan.

Research programs to identify and suggest means of controlling the noise in centrifugal blowers are in progress in the Engine Laboratory. A review on the existing literature on this subject has been published in Reference (1) to establish further areas of research. Noise measurements on a wide variety of blowers existing in the laboratory are also described in Reference (1) to identify the important frequency components. From these blowers, two were selected for more detailed acoustic and aerodynamic measurements, in order to study the factors relating the generated noise with the aerodynamic performance parameters. This report describes the results of these measurements made on a 15 horsepower and a 1/3 horsepower unit.

2.0 FAN NOISE SIMILARITY RELATIONS

The application of similarity principles for the aerodynamic performance of fans is well established. The advantages of this principle being that a series of dimensional curves for a similar group of machines collapses into a single curve on non-dimensional representation. For centrifugal blowers and fans of similar types the performance curves in the incompressible flow regime are represented by one single curve of pressure or head coefficient as a function of flow or capacity coefficient.

$$\psi = f(\phi)$$

ψ is the pressure coefficient, $\psi = \dfrac{\Delta p_T}{\rho_2 u_2^2}$

ϕ is the flow coefficient, $\phi = \dfrac{C_{m_2}}{u_2}$

mention this relation and deduce the dimensional relation for const speed

Δp_T is the total pressure difference across the fan

ρ_2 is the density of air at the impeller outlet

u_2 is the impeller tip speed

C_{m_2} is the meridonal velocity at the discharge side of the impeller

Flow coefficient is used as an independent variable and similarly the efficiency of the blowers is plotted as a function of flow coefficient. The dependence of Reynolds number should be considered in the performance analysis but the performance curves normally vary only slightly in a wide range of Reynolds number.

Various attempts have been made to establish fan noise laws with the aim of predicting the sound power produced by a group of similar machines based on the detailed measurements of one fan belonging to this group. Based on dimensional analysis, the dependence of sound power on aerodynamic parameters is given by References (2) and (3).

Sound Power Level, SWL = $F(M, R, S, \phi)$

where M is the Mach number given by $M = \frac{u_2}{a_o}$

a_o is the speed of sound

R is the Reynolds number given by $R = \frac{u_2 D}{\nu}$

ν is the kinematic viscosity

S is the Strouhal number given by $S = \frac{fD}{u_2}$

f is the frequency

D is the impeller diameter

For an impeller running at a constant speed, Reynolds number and Mach number remain constant and

$$SWL = F(\phi, S)$$

3.0 EXPERIMENTAL SET-UP, INSTRUMENTATION AND MEASUREMENT DETAILS

3.1 BRIEF DESCRIPTION OF TEST BLOWERS

15 horsepower blower:

rated capacity	1500 cfm at 0.9 psi
	2100 cfm at 0 psi
size	26.25 in. diameter impeller
	10 straight radial blades
drive	15 horsepower induction motor at 3500 rpm

The details of the impeller, casing and inlet are shown in Fig. 1. The original inlet to the blower had a butterfly valve to control the flow but this was replaced by a circular arc bellmouth for providing smooth flow. The flow from the impeller was collected by the volute and led on to a short diffuser duct.

1/3 horsepower blower:

rated capacity	400 cfm at 0.05 psi and 650 cfm at 0 psi
size	11.25 in. diameter impeller
	12 circular arc blades, backward curved
	2 in. length
	5.438 in. diameter inlet
	5.5 in. diameter outlet
drive	1/3 horsepower motor at 1700 rpm.

Details of this blower are shown in Fig. 2. Tests were again carried out by replacing the original sharp edged inlet with a circular section bellmouth.

3.2 AERODYNAMIC TEST SET-UP, INSTRUMENTATION AND METHOD OF MEASUREMENTS

The ducting arrangement and instrumentation for the aerodynamic performance measurements are shown in Fig. 3. Air flow through the fan was controlled by changing the back pressure at the downstream outlet with a simple throttle-plate, mounted sufficiently far downstream (20 diameters) from the measuring station.

Inlet temperatures and outlet temperatures in the 6 inch diameter pipe immediately after the blower casing exit were measured as shown. Static and total pressure readings were taken inside the 6 inch diameter pipe at 15 diameters downstream of the fan. The arrangement of total and static pressure tubes is shown in Fig. 3; the total pressure tubes were mounted at 3/4 radius from the pipe center to yield the average velocities in the pipe based on Reference (4) and the static taps were mounted circumferentially midway between the total head tubes.

Similar measurements were carried out for the 1/3 horsepower blower.

3.3 FLOW AND PRESSURE COMPUTATIONS

(i) Flow measurements

Air mass flow was calculated from measurements of T_{T3}, the total temperature at position 3, P_{T4} and P_{S4}, the total and static pressure at position 4 and the assumption that $T_{T3} = T_{T4}$

(ii) Pressure Measurements

Fan delivery pressures were calculated from measurements made at station 4 and an estimate of the pressure loss in the pipe based on a previously measured friction factor.

$$P_{T3} = P_{T4} + \text{pipe friction loss}$$

(iii) Pressure and Flow Coefficients

Pressure coefficients, $\psi = \dfrac{\Delta P_T}{\rho_2 u_2^2}$

where $\Delta P_T = P_{T_3} - P_{T1}$ and ρ_2 is assumed to be equal to ρ_3. u_2, as stated previously, is the rotor tip speed.

Flow coefficient, $\phi = \dfrac{C_{m_2}}{u_2}$

where C_{m_2}, the meridonal velocity, is estimated from continuity by assuming no change in density from rotor exit to station 3.

Hence $C_{m_2} = \dfrac{V_3 A_3}{A_2}$

3.4 TEST SET-UP FOR NOISE MEASUREMENTS AND INSTRUMENTION

Industrial fan and blower noise is usually measured by the in-duct method as this method is much simpler than either the free field or reverberant field methods and needs no expensive measurement facilities. There are existing ISO and British Standards for measuring sound power levels by the 'in-duct' method, the details of which are specified in LSO/TC43/SC1/WG3 and BS848.

The 'in-duct' method requires that the test duct should be terminated anechoically to prevent sound reflections from the sudden duct exit. The experimental set-up for the 15 horsepower blower is shown in Fig. 4. The exhaust duct was terminated with an exponential horn packed with loose fibre glass material with a center passage for the discharge flow. The noise measurements were made by a one-half inch microphone placed one diameter upstream of the horn and midway between the pipe axis and the wall. The microphone was provided with a nose cone to suppress the inherent flow noise. The same arrangement was used for measuring noise from the 1/3 horsepower blower.

4.0 RESULTS AND DISCUSSIONS

Measured mass flow and total and static pressure characteristics for the 15 horsepower blower are shown in Fig. 5. Power and efficiency measurements were not made during these tests. These results indicate that the pressure versus flow characteristics of the fan are relatively flat with almost imperceptible stall. These characteristics indicate an increase in horsepower consumed by the fan with the mass flow. The non-dimensional characteristic of the fan is shown in Fig. 6 at only one speed of 3500 rpm. Noise spectrum corresponding to point B on the blower performance characteristics in Figs 5 and 6 is shown in Fig. 7. The prominent noise component appeared to be the tone at the blade passing frequency in 630 Hz band with the noise levels remaining constant at 12-14 dB below this tone level in lower frequency bands. Noise levels at frequencies higher than this tone dropped off rapidly with increases in frequency. The spectrum at point A which correspondeds to low noise condition on the blower performance characteristics is shown in Fig. 8. The blade passing frequency tone was no longer as prominent in the spectrum and its level had almost reduced to the broadband noise levels in the neighbouring frequency bands. Even the levels of the broadband noise were reduced by 2-5 dB at this condition. Several

combine 7 and 8

spectra measured at other conditions showed a prominent tone at the blade passing frequency. The changes in noise levels in the various frequency bands with mass flow and capacity coefficient are shown in Figs 9 to 14. In Fig. 9 (frequency bands 100, 125, and 160 Hz) the noise levels changed by about 5-7 dB over the complete range of throttle conditions, with minimum noise level occurring at condition A. There were less significant variations in levels in frequency bands at 200, 250, 315 and 400 Hz which correspond to Strouhal numbers S = 1.08, 1.35, 1.70 and 2.16 respectively. Fig. 11 shows that the blade passing frequency tone varied significantly with mass flow. Of course, the minimum level occurred at condition A, with the level increasing by about 16 dB towards high and low mass flow conditions. The changes at 500 Hz band were not that drastic but nonetheless significant, an increase of 8 dB on either side of the minimum noise level. Figs 12 to 14 indicate that the changes were only slight in the middle regions, however at extreme mass flow conditions there were some noticeable increases in levels. mention this

Experiments of Deeprose and Brooks (5) have shown that the minimum noise condition occurs at a flow corresponding to the design point, the point of maximum efficiency. Based on this information Mellin (6) has discussed the design of blowers for a given pumping requirement for minimum noise condition. The flow processes inside a centrifugal impeller are very complex as the flow confined within the blade channels changes its direction in three dimensions and strong rotations are induced into the fluid flow. Even at the design point there would be a detached flow region at the suction wall trailing edge. The blade passing frequency tones have been established to be generated by the interaction of the exit flow from the impeller blades with the cut-off edge - Reference (7). The blade passing frequency tone level was minimum at condition A, which presumably was the design point with the best flow situation within the impeller. The flow within the impeller probably deteriorated at other flow rates increasing the tone level. The unstable flow region extends away from the design point which would explain the increase in broadband noise.

The measured air flow rate versus pressure rise and capacity coefficient versus head coefficient of the 1/3 horsepower blower are shown in Figs 15 and 16. The pressure rise increased continuously with reduction in the flow without any indication of stall. Noise spectra of this blower at flow conditions marked A, B, C, and D on the aerodynamic performance curves are shown in Fig. 17. The noise spectra at conditions A and B contained prominent tones at 160 and 200 Hz. But the levels of these tones reduced considerably at condition C and at condition D, which approximated to zero flow and maximum pressure rise, these tones disappeared completely. There was no evidence of the blade passing frequency tones, particularly of the fundamental (340 Hz) in the 315 Hz band. It may be recalled that in the case of the 15 horsepower blower, the level of the blade passing frequency tone reduced significantly at an intermediate flow condition which was expected to be closer to the design point. The 315 band, containing the blade passing frequency tone showed minimum level at condition C.

The spectra contained predominantly low frequency sound below 500 Hz, and the levels fell sharply with increase in frequency. The cause of the tone at 160 Hz could be attributed to the presence of six struts supporting the 12 blades as shown in Fig. 2. The tone in the 200 Hz band may be ascribed to the characteristic frequency associated with the casing geometry. Moreland (8) identified the presence of Helmholtz resonance peaks in casings whose dimensions were much smaller than the wavelength associated with the resonant peaks and presented a method to calculate the frequencies associated with these resonant peaks based on lumped impedance model. The calculated resonant frequency for the casing was 232 Hz. While the peaks at conditions A and B might be attributed to the Helmholtz resonance frequency of the casing, the total absence of this tone at condition D would suggest that this possibility is precluded. Perhaps the forcing mechanism provided by the fan at approximately zero flow condition was not sufficient to excite the casing.

Figs 18 to 23 demonstrate that the noise levels in almost all bands increased with the mass flow unlike the characteristics of the 15 horsepower blower, showing minimum noise levels at zero flow condition and maximum at full flow condition. This increase is quite steep in the frequency bands of 160 and 200 Hz, which contained tones, a rise of 16-18 dB from no flow to full flow conditions. In other bands these increases were of the order of 10-12 dB.

The two fans tested revealed different flow and noise characteristics. Based on the above results one may probably conclude that the minimum noise for the 15 horsepower blower was associated with the best efficiency point where the flow could be expected to be smooth and more stable compared to other conditions. There was no indication of the best efficiency point for the 1/3 horsepower blower as the noise was minimum at almost no flow. As the fan did not show any stall features, the power consumed at low flows was small and there was little work done. Therefore it would not be surprising that the noise levels in each band were low.

The spectral characteristics of the two blowers were different with the 15 horsepower blower showing a strong blade passing frequency tone and the 1/3 horsepower blower indicating tones corresponding to the supporting struts for the blades and the casing geometry with the absence of blade passing frequency tone.

5.0 CONCLUSIONS

(1) The aerodynamic performance characteristics of the 15 horsepower blower and 1/3 horsepower blower were relatively flat with the former showing almost imperceptible stall and no evidence of stall from the latter; with maximum pressure rise at zero flow condition.

(2) The prominent noise component for the 15 horsepower blower was the tone at the blade passing frequency with high levels of broadband noise at low frequencies. The 1/3 horsepower blower revealed the presence of two rather less prominent tones with relatively high broadband noise levels at low frequencies. These tones could be attributed to the presence of struts supporting the blades and the Helmholtz resonance frequency of the casing.

(3) The noise levels in general showed strong dependence on the mass flow or flow coefficient with the 15 horsepower blower showing minimum noise levels at a flow coefficient corresponding to the maximum efficiency point. The 1/3 horsepower blower noise levels increased with the flow coefficient over the entire flow range. Maximum noise levels for both fans were measured at free delivery conditions.

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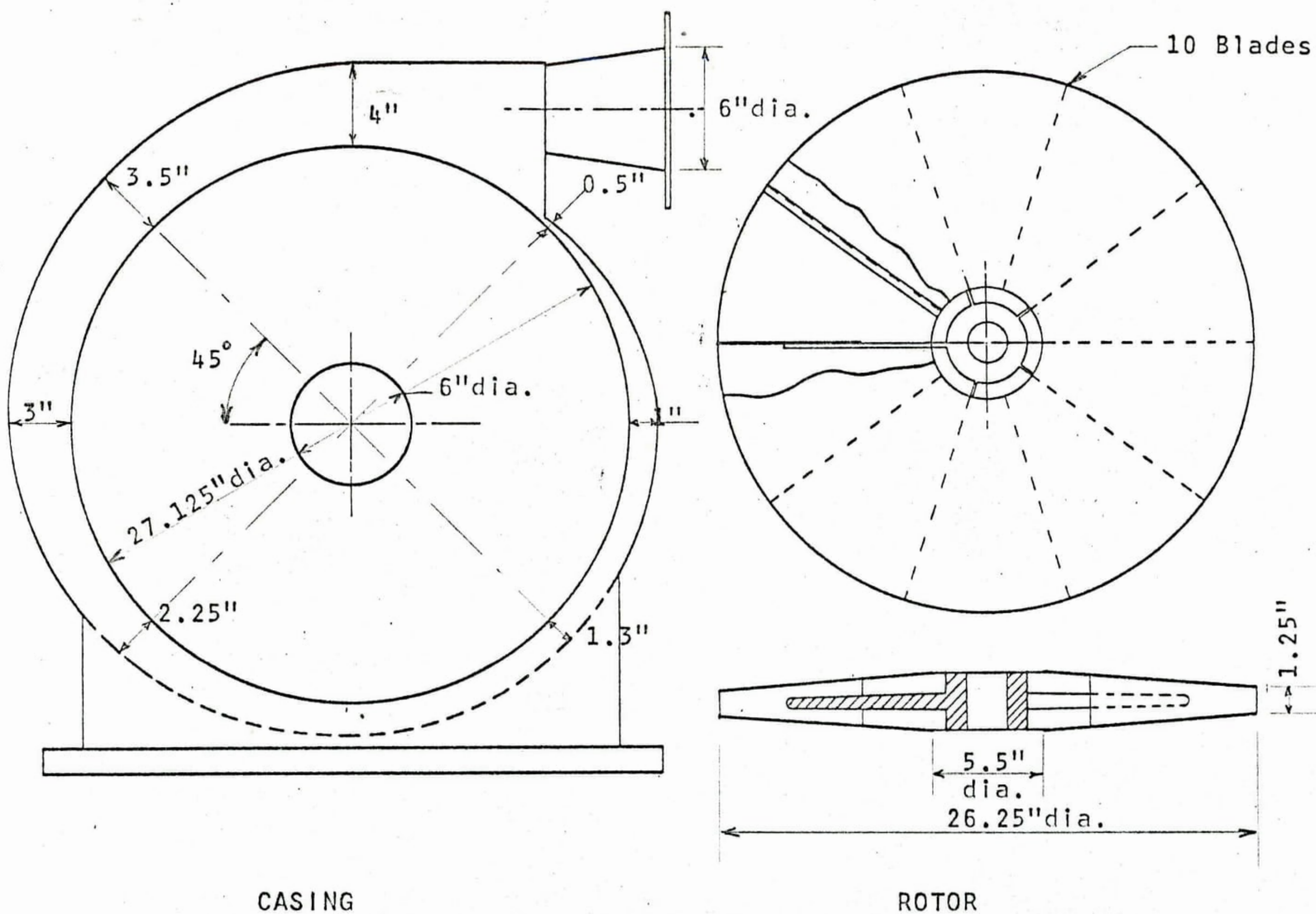

Figure 1. 15 H.P. Blower Details.

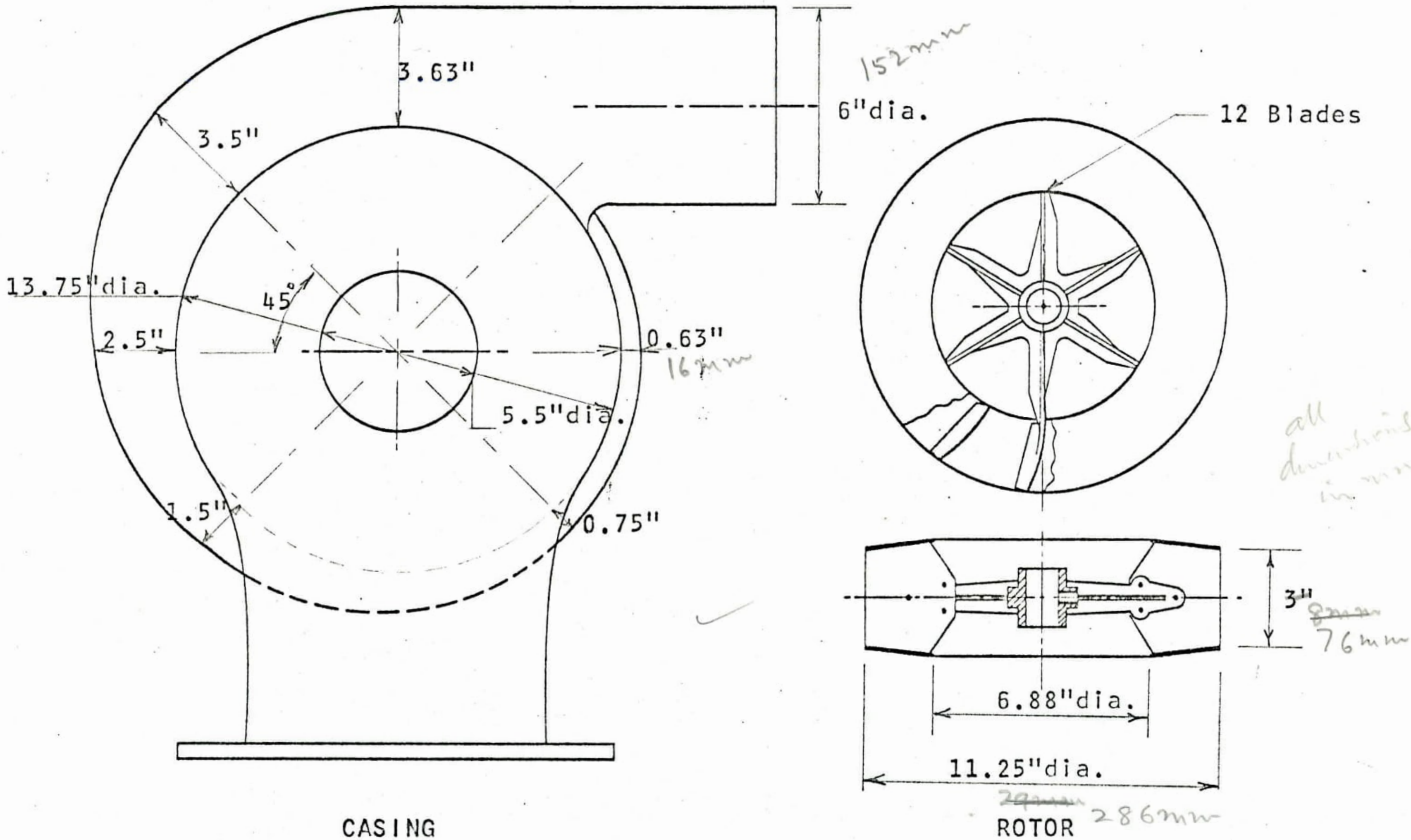

Figure 2. 1/3 H.P. Blower Details.

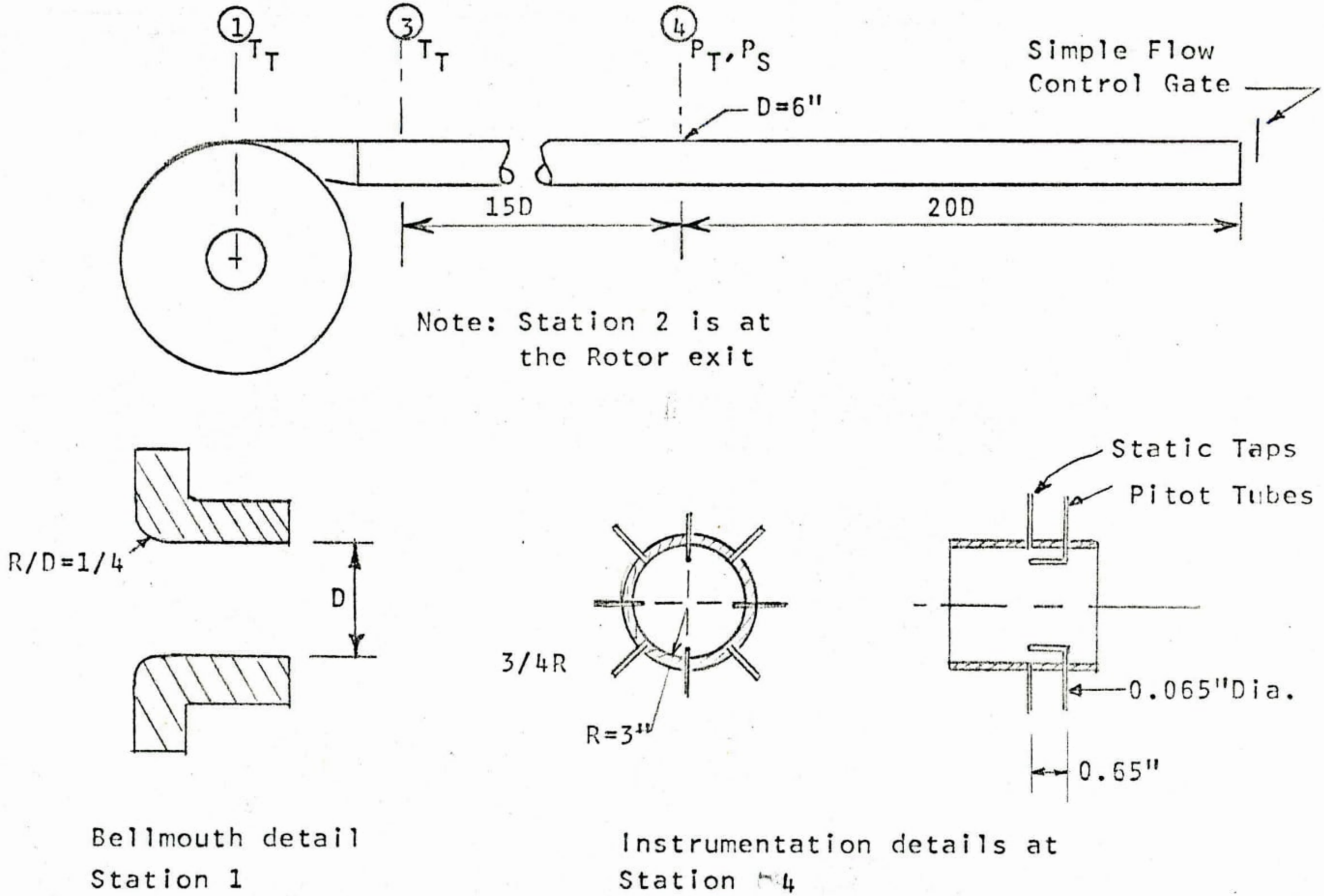

Figure 3. Ducting and Instrumentation Arrangements for Flow and Pressure Measurements.

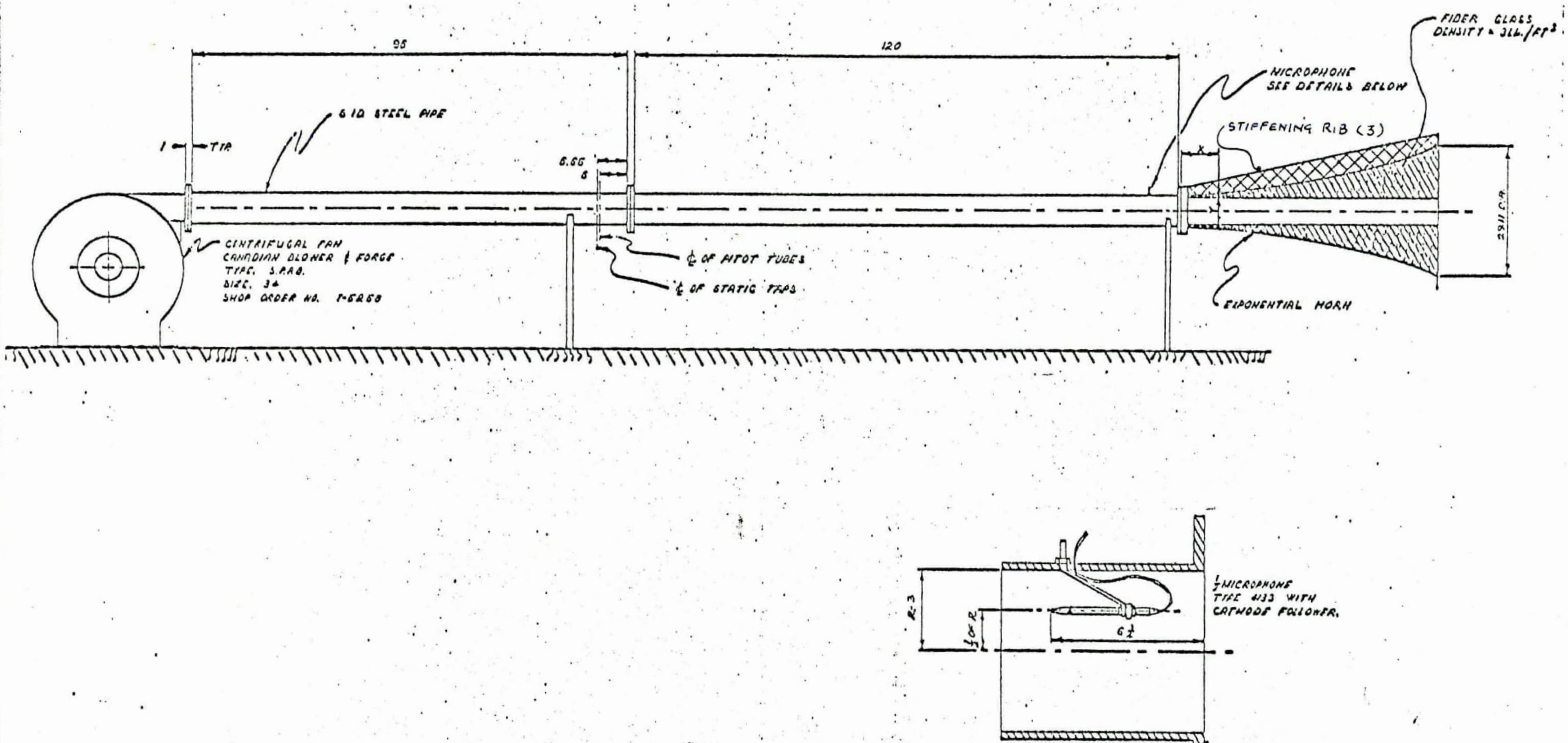

Figure 4. Ducting and Instrumentation Arrangements for Noise Measurements.

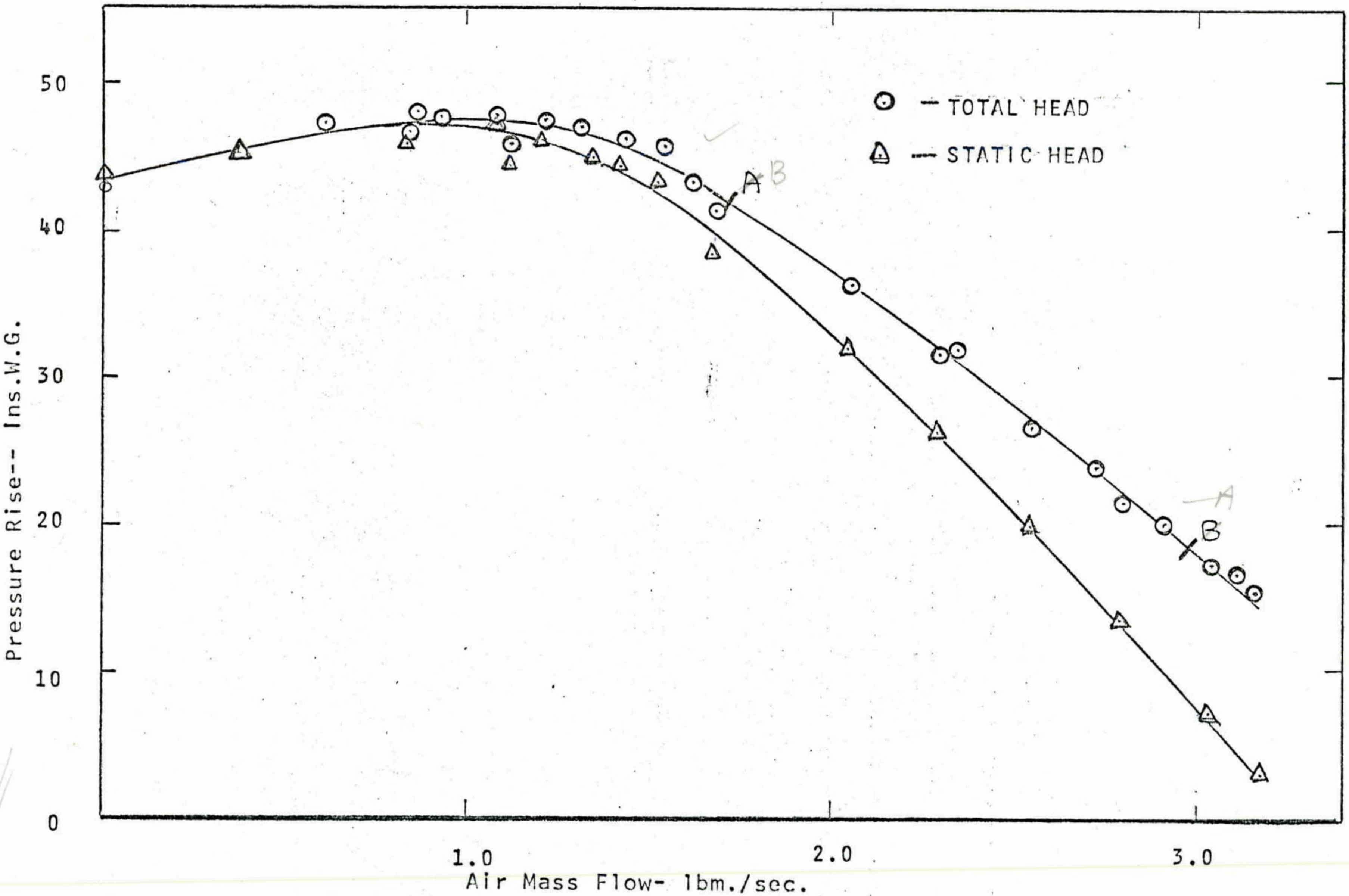

Figure 5. Mass Flow Versus Pressure Rise for 15 H.P. Blower.

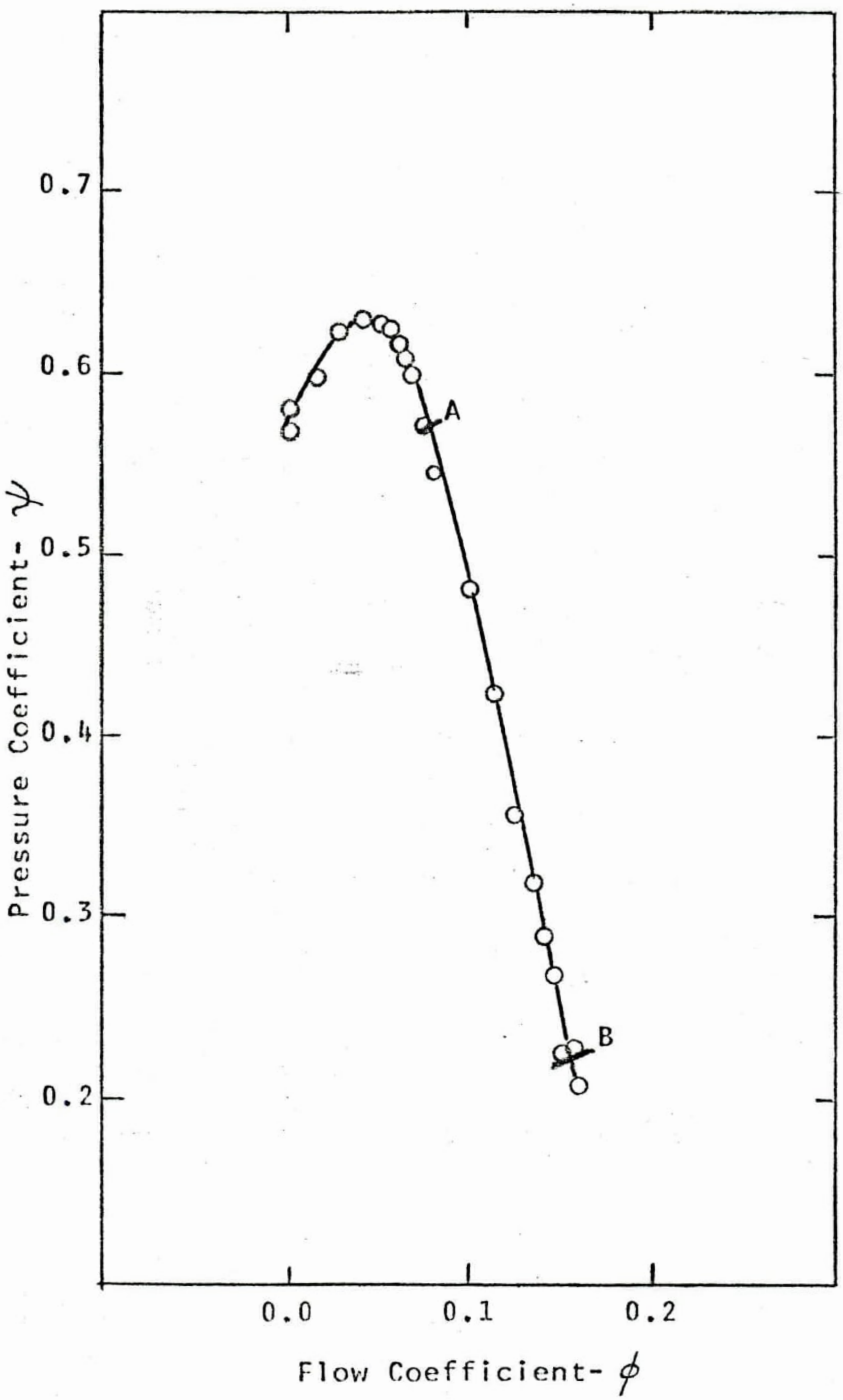

Figure 6. Pressure Versus Flow Coefficient for the 15H.P.Blower.

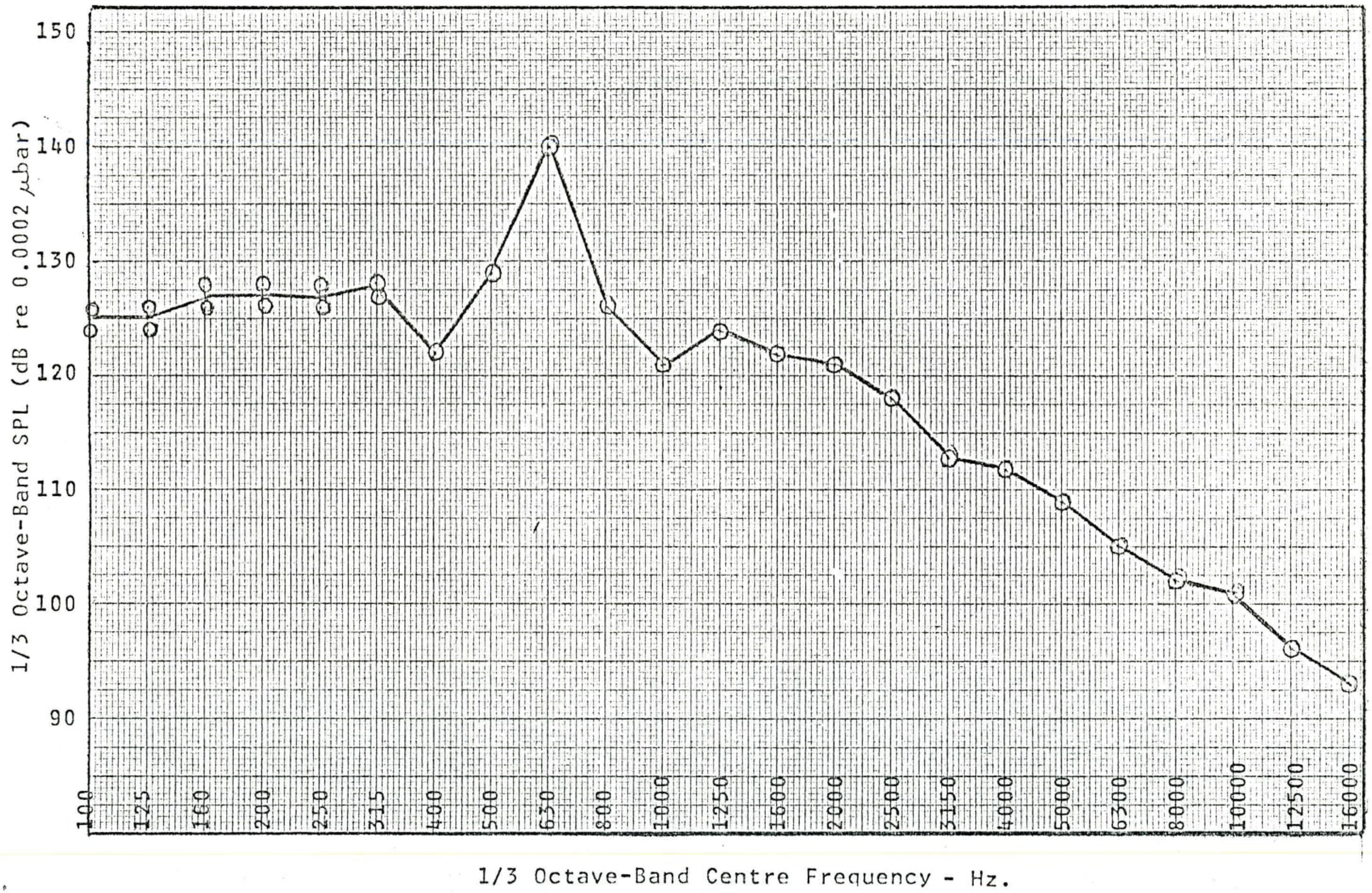

Figure 7. Noise Spectrum at Point 'B' (15H.P.Blower).

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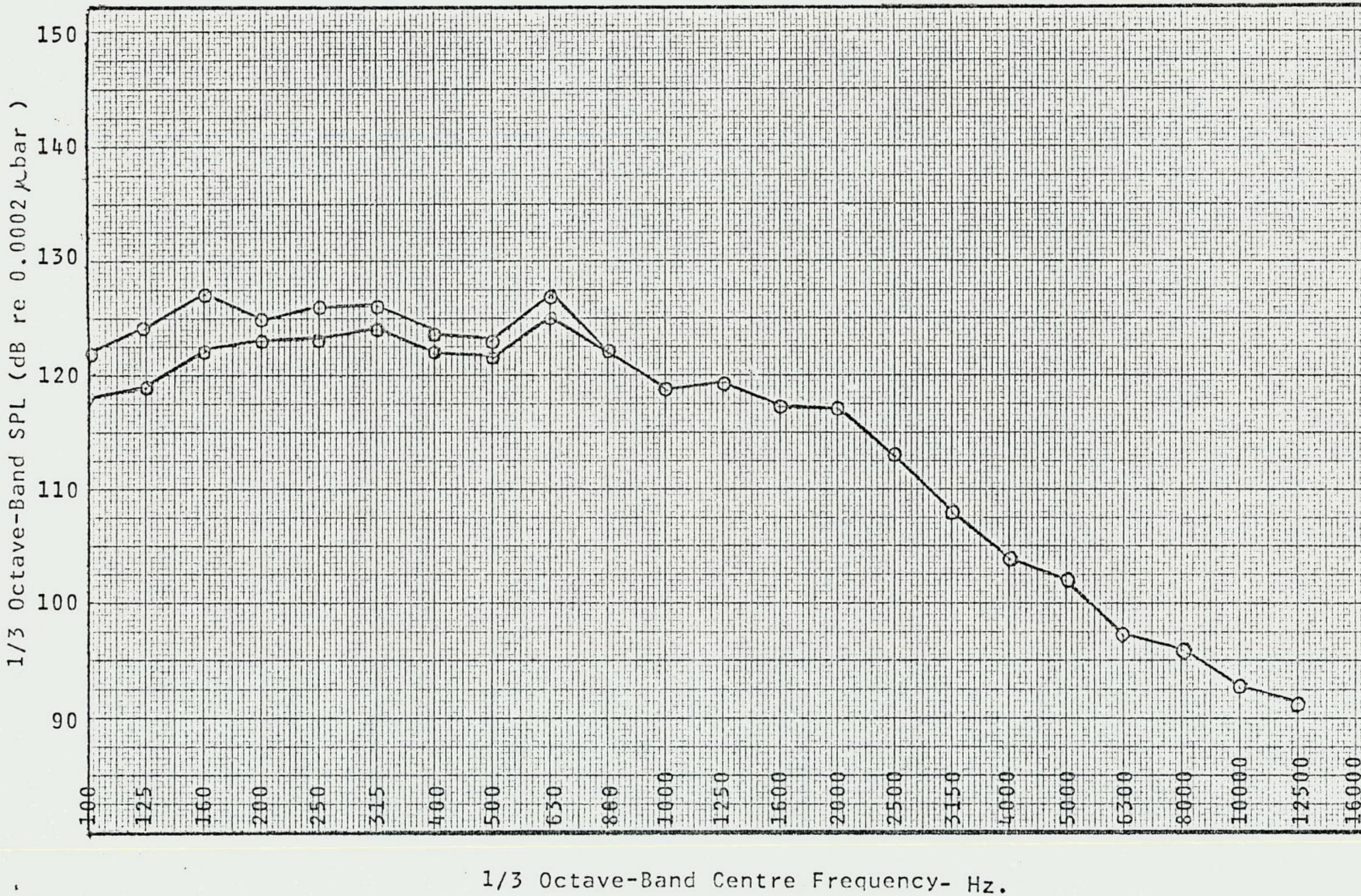

Figure 8. Noise Spectrum at Point 'A' (15 H.P.Blower).

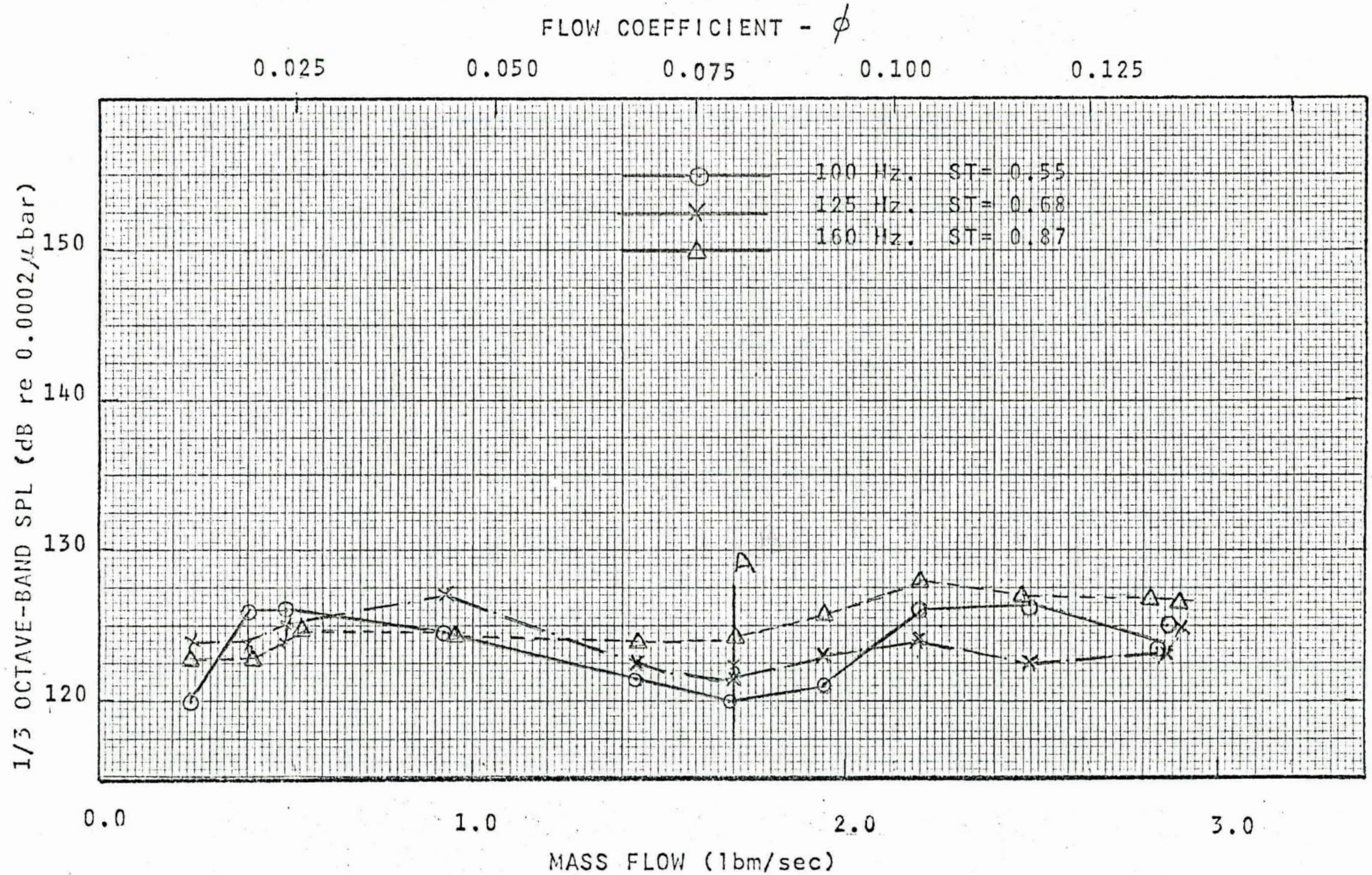

Figure 9. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 100, 125 and 160 Hz. (15 H.P.Blower).

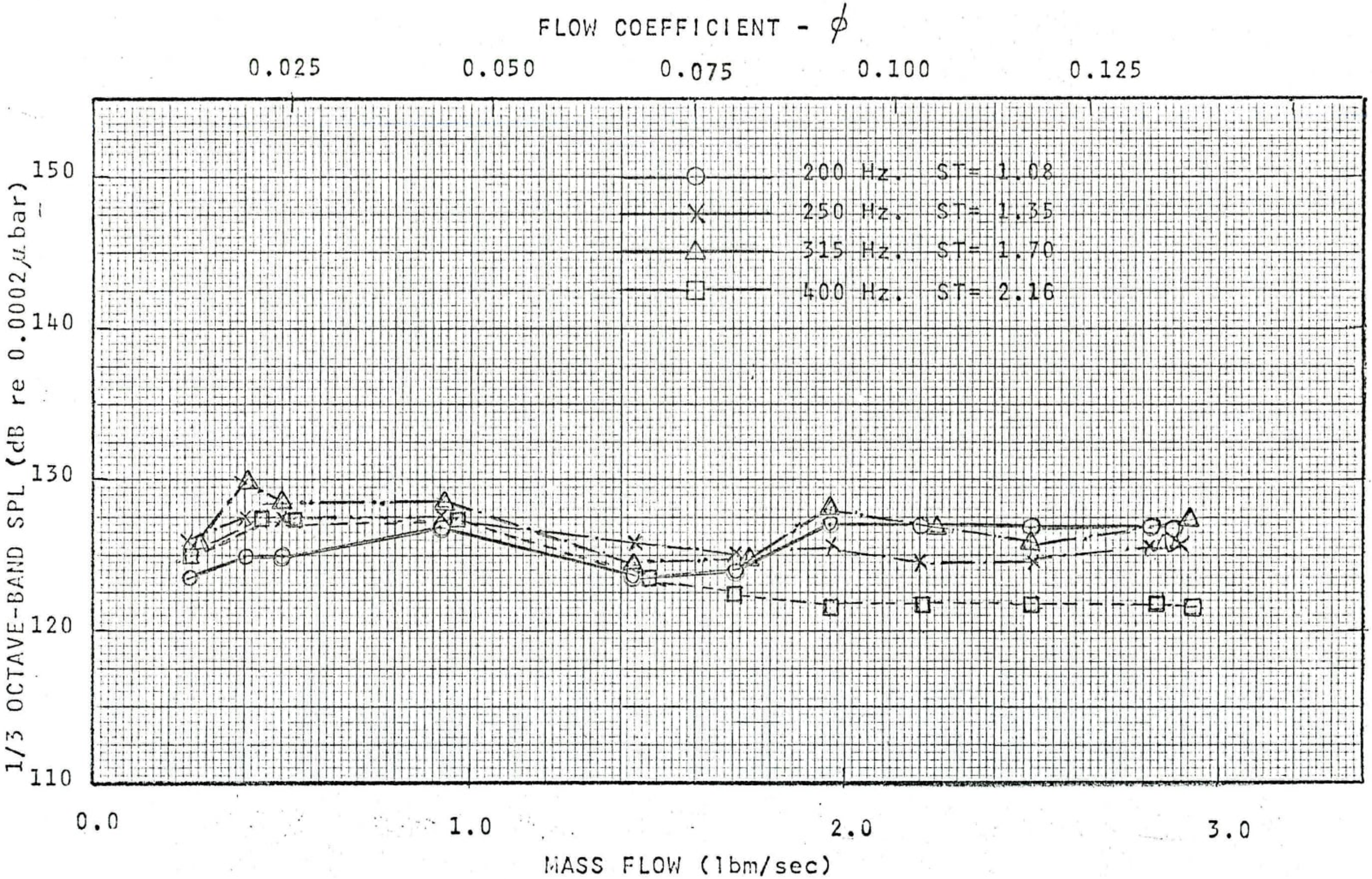

Figure 10. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 200, 250, 315 and 400 Hz. (15 H.P.Blower).

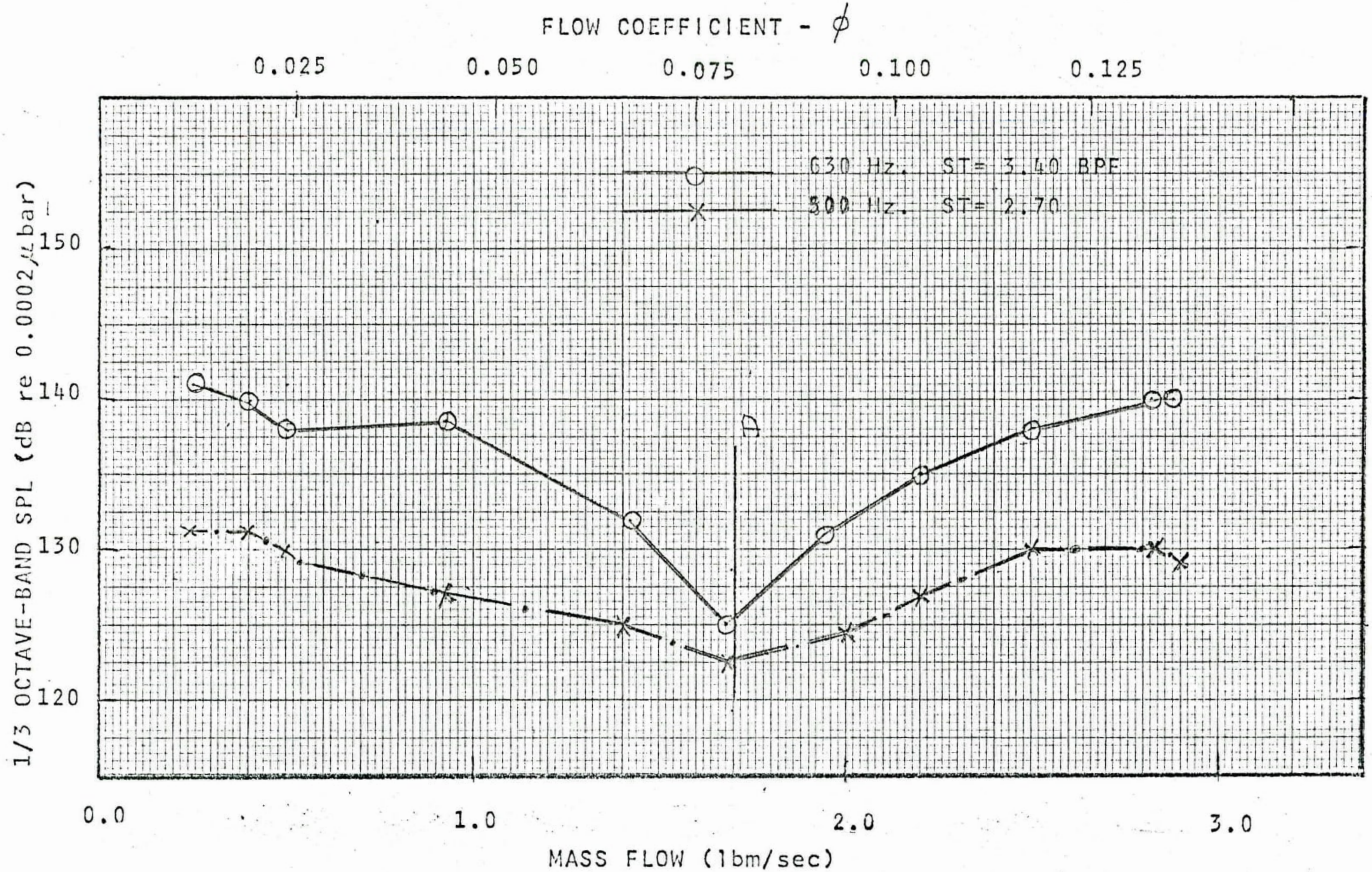

Figure 11. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 500 and 630 Hz. (15 H.P.Blower).

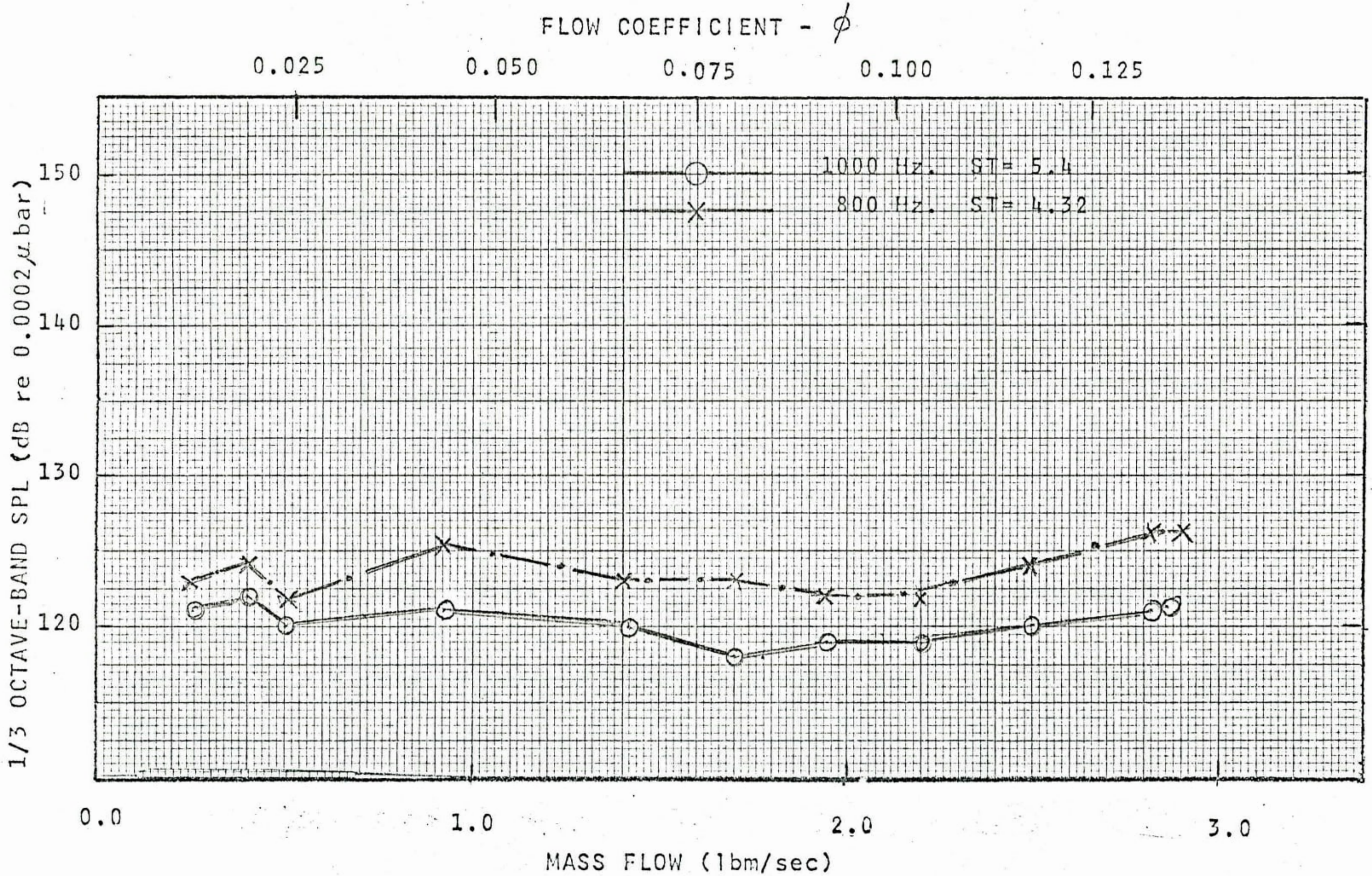

Figure 12. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 800 and 1000 Hz. (15 H.P.Blower).

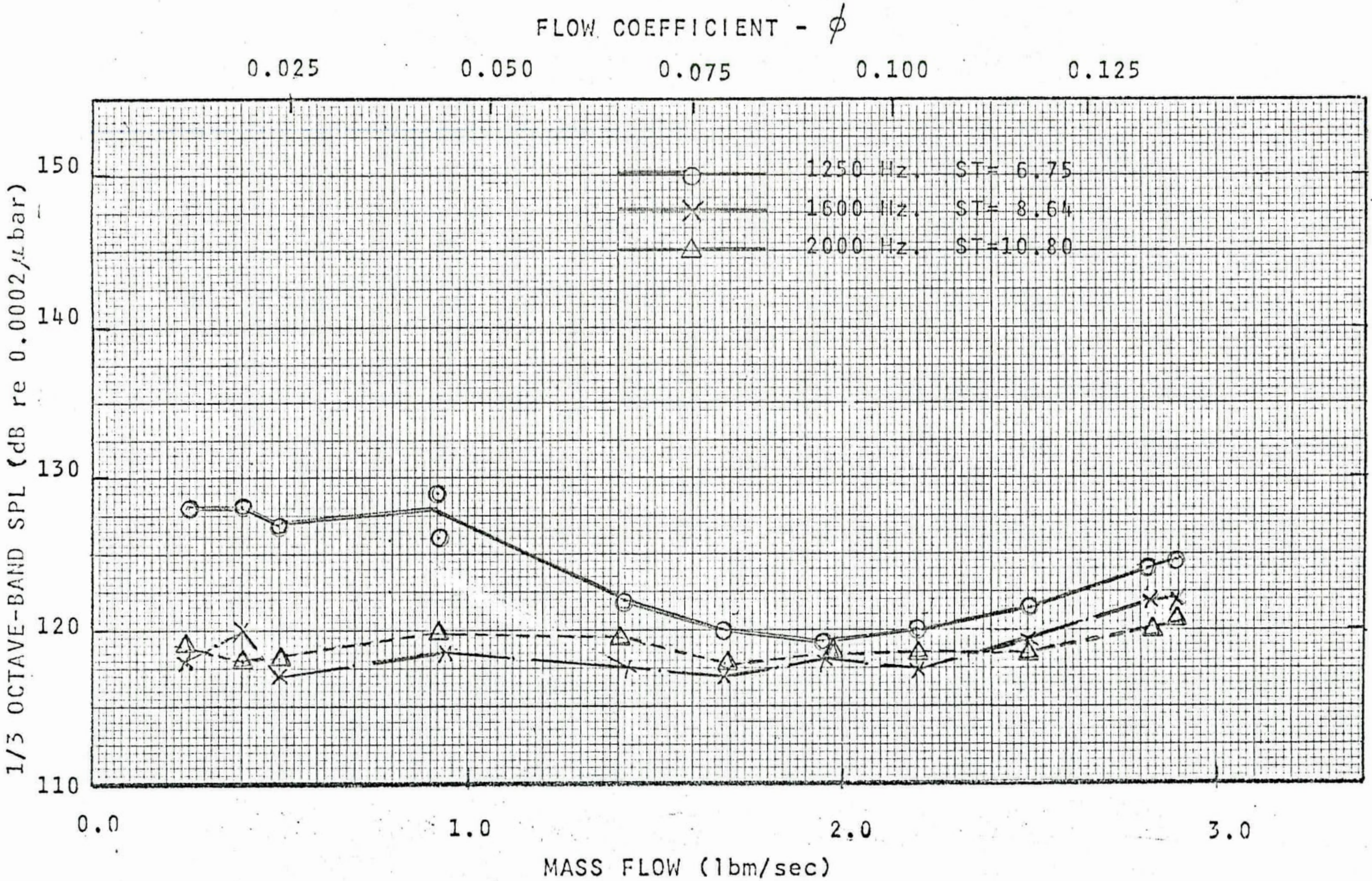

Figure 13. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 1250, 1600 and 2000 Hz. (15 H.P.Blower).

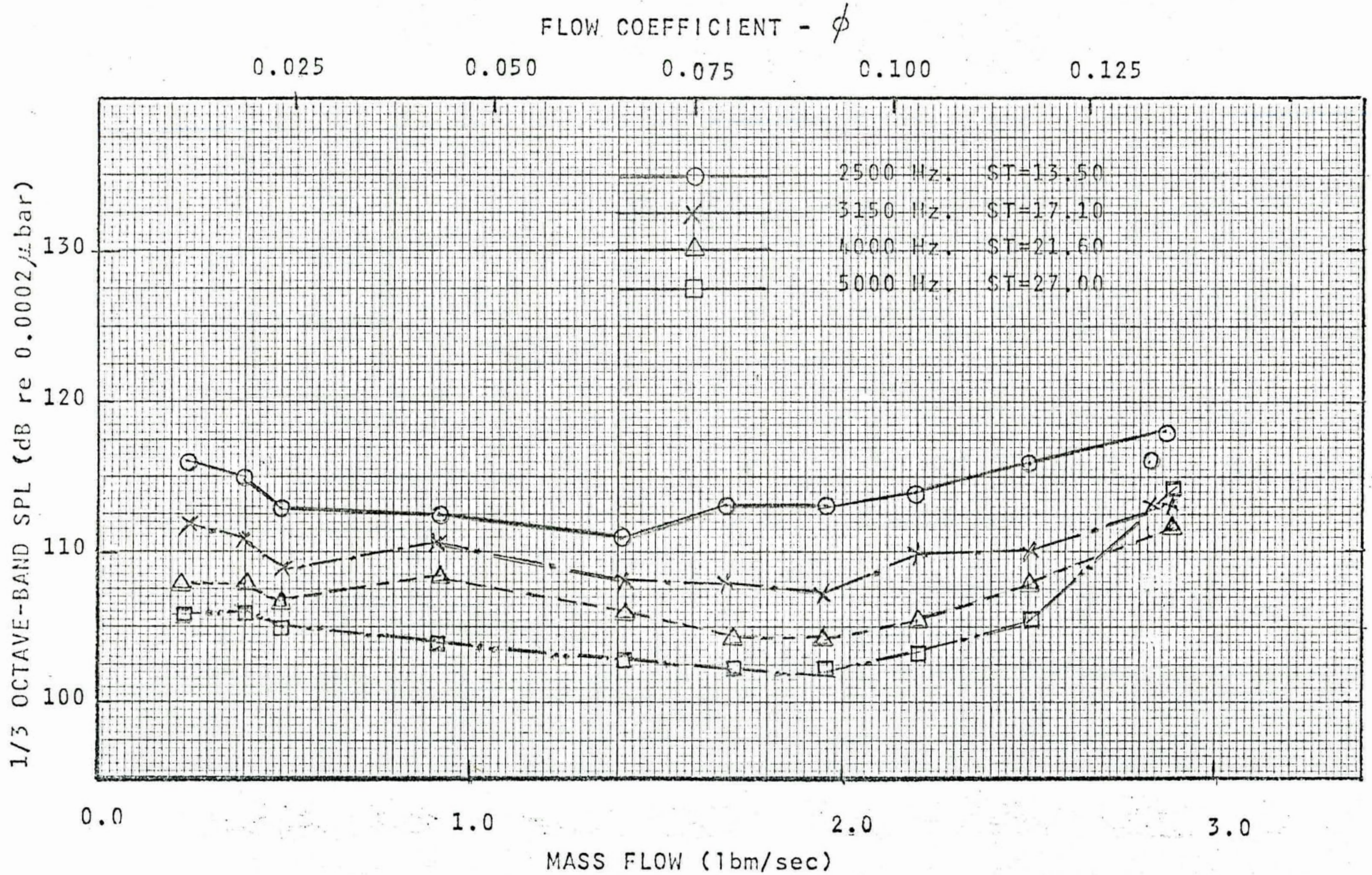

Figure 14. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 2500, 3150, 4000 and 5000 Hz. (15 H.P.Blower).

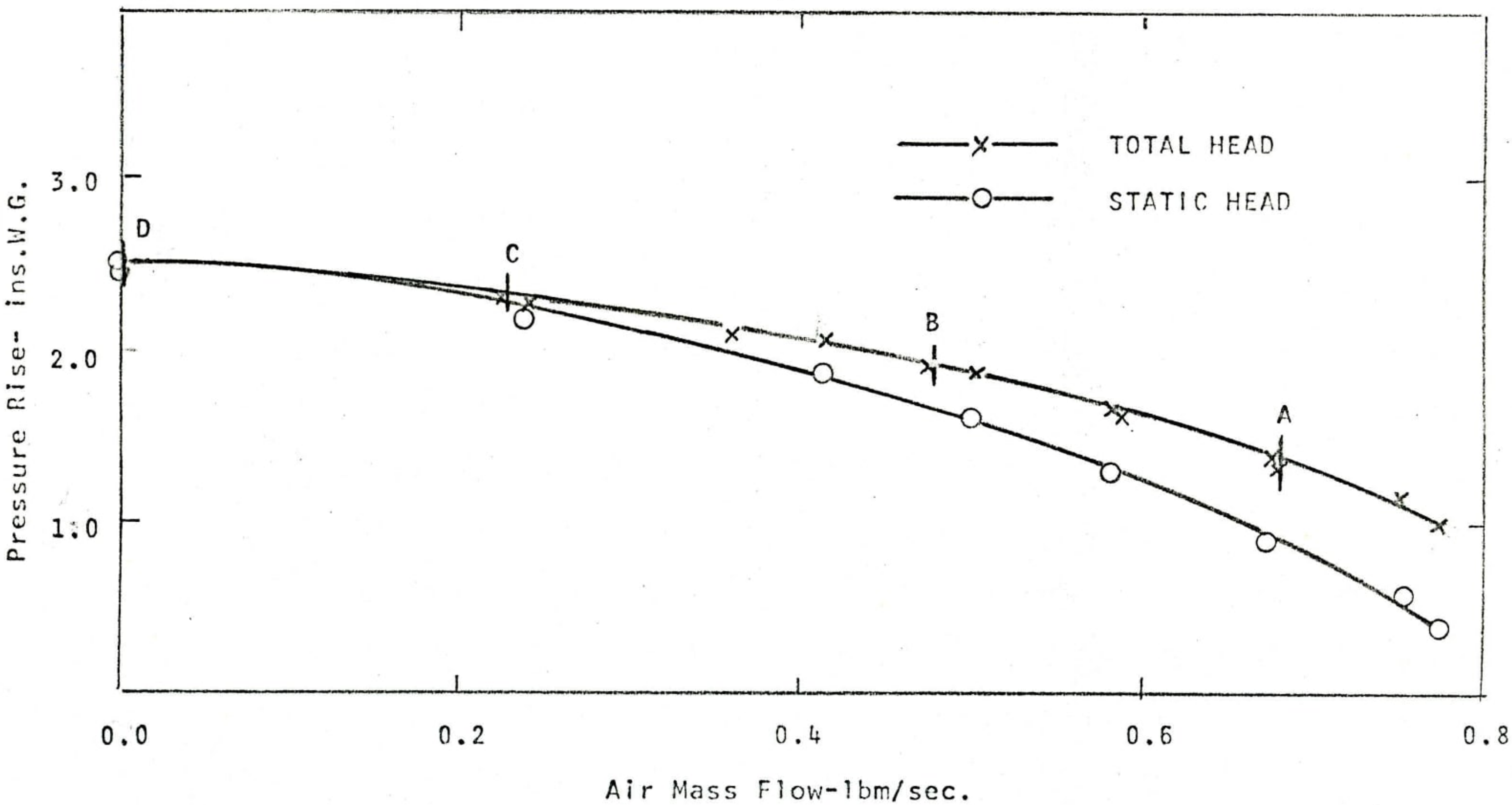

Figure 15. Mass Flow versus Pressure Rise for 1/3 H.P.Blower.

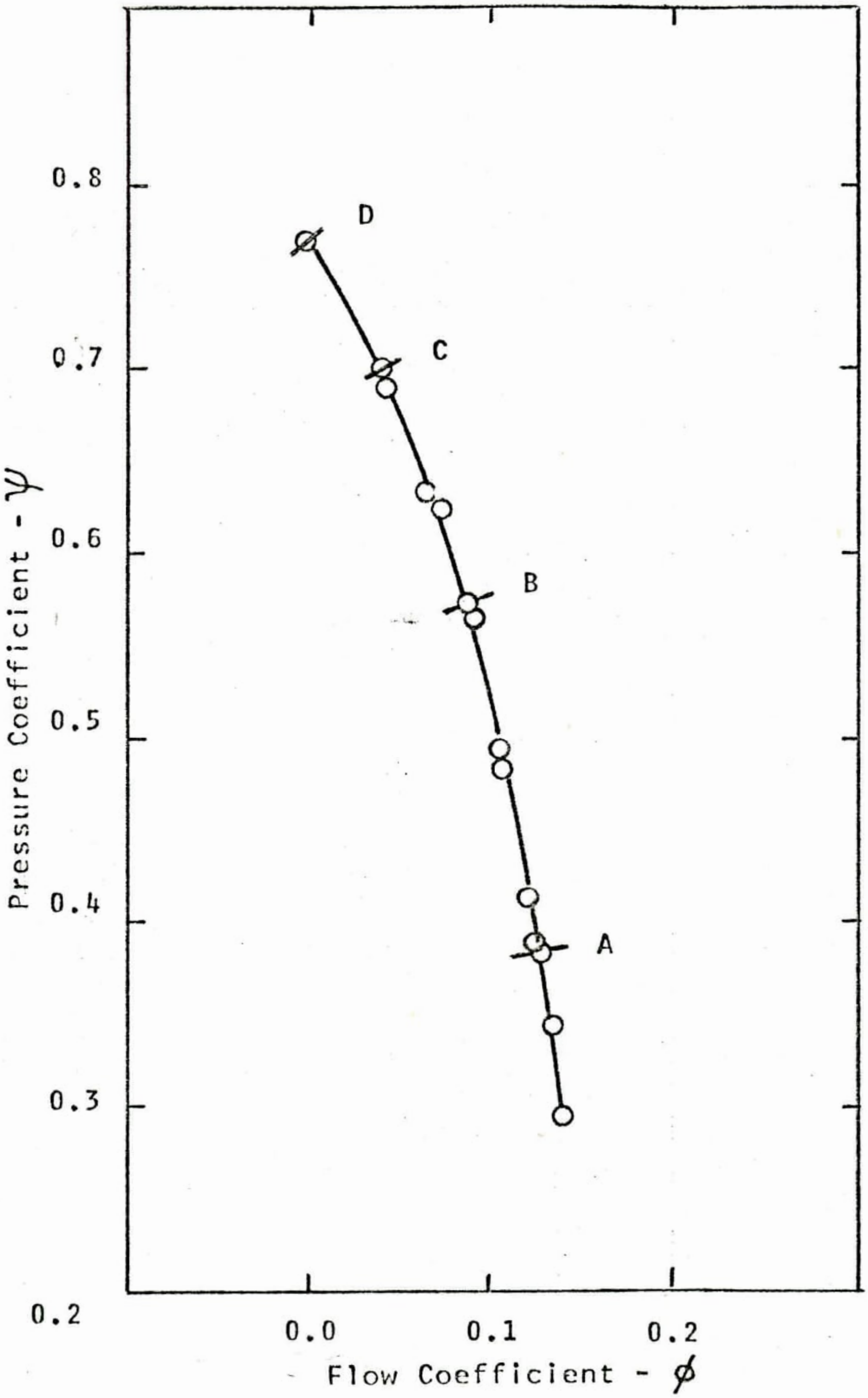

Figure 16. Head Versus Flow Coefficient for 1/3 H.P.Blower.

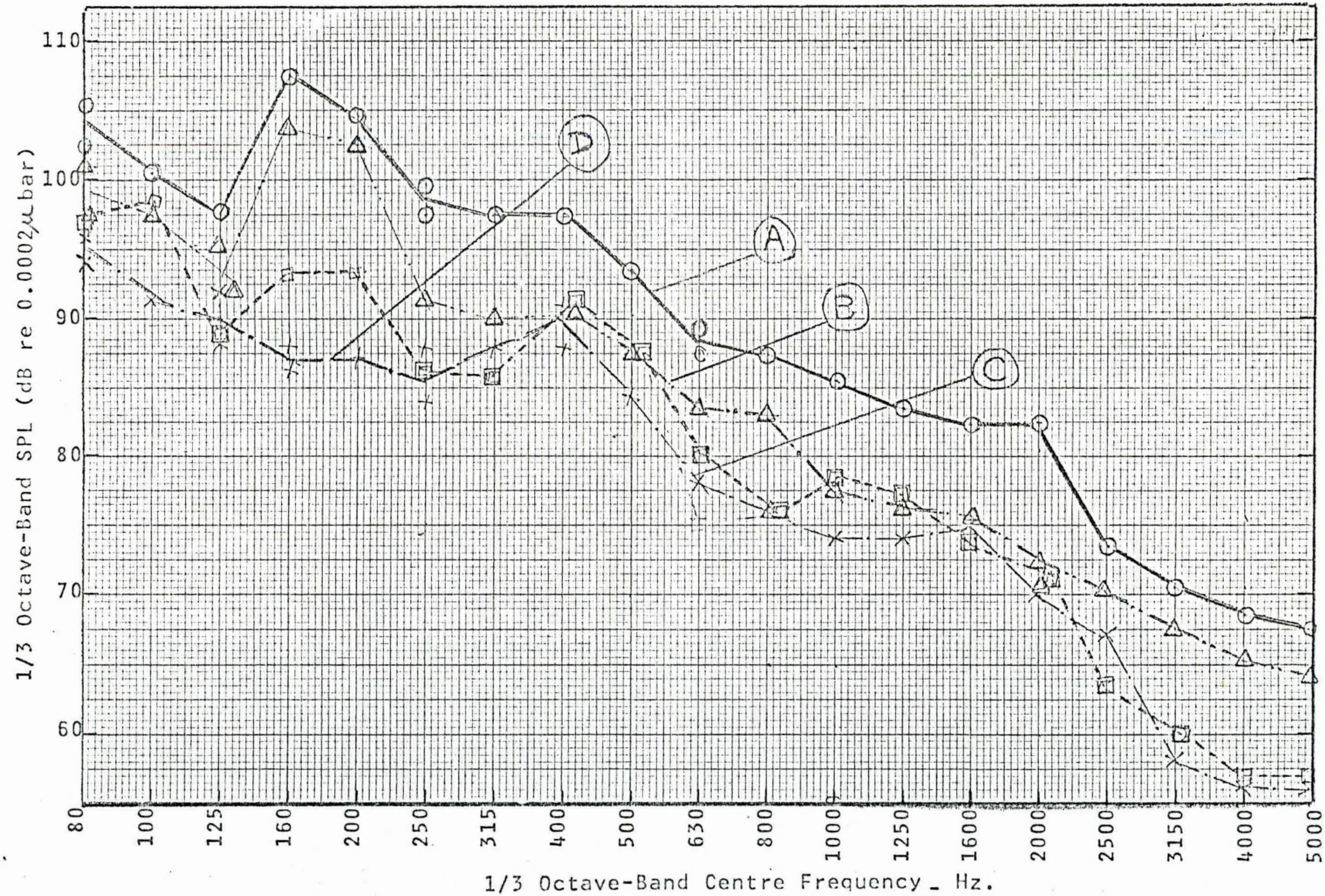

Figure 17. Noise Spectrum at Point 'A','B','C' and'D'.(1/3 H.P.Blower).

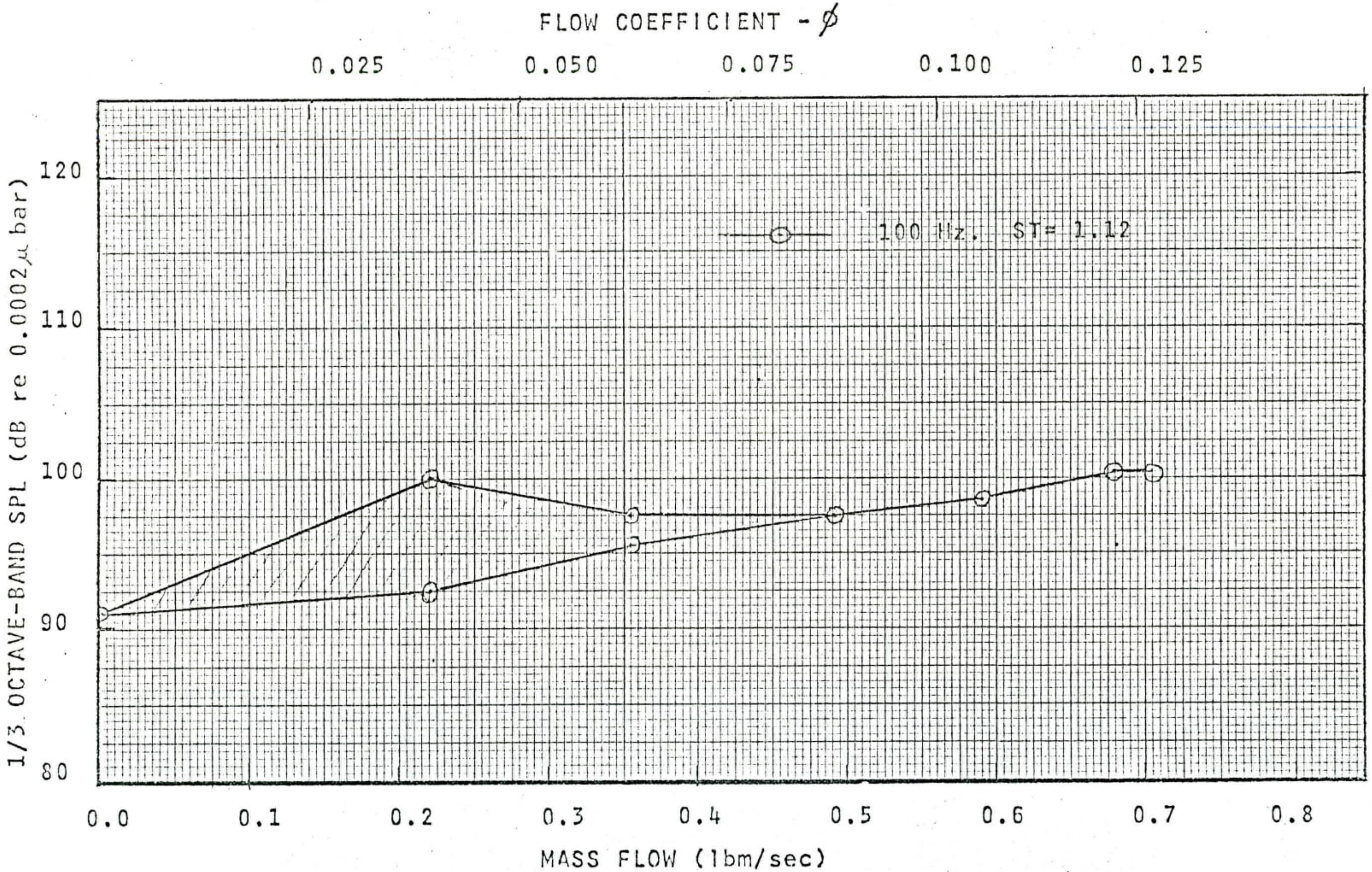

Figure 18. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 100 Hz. (1/3 H.P.Blower).

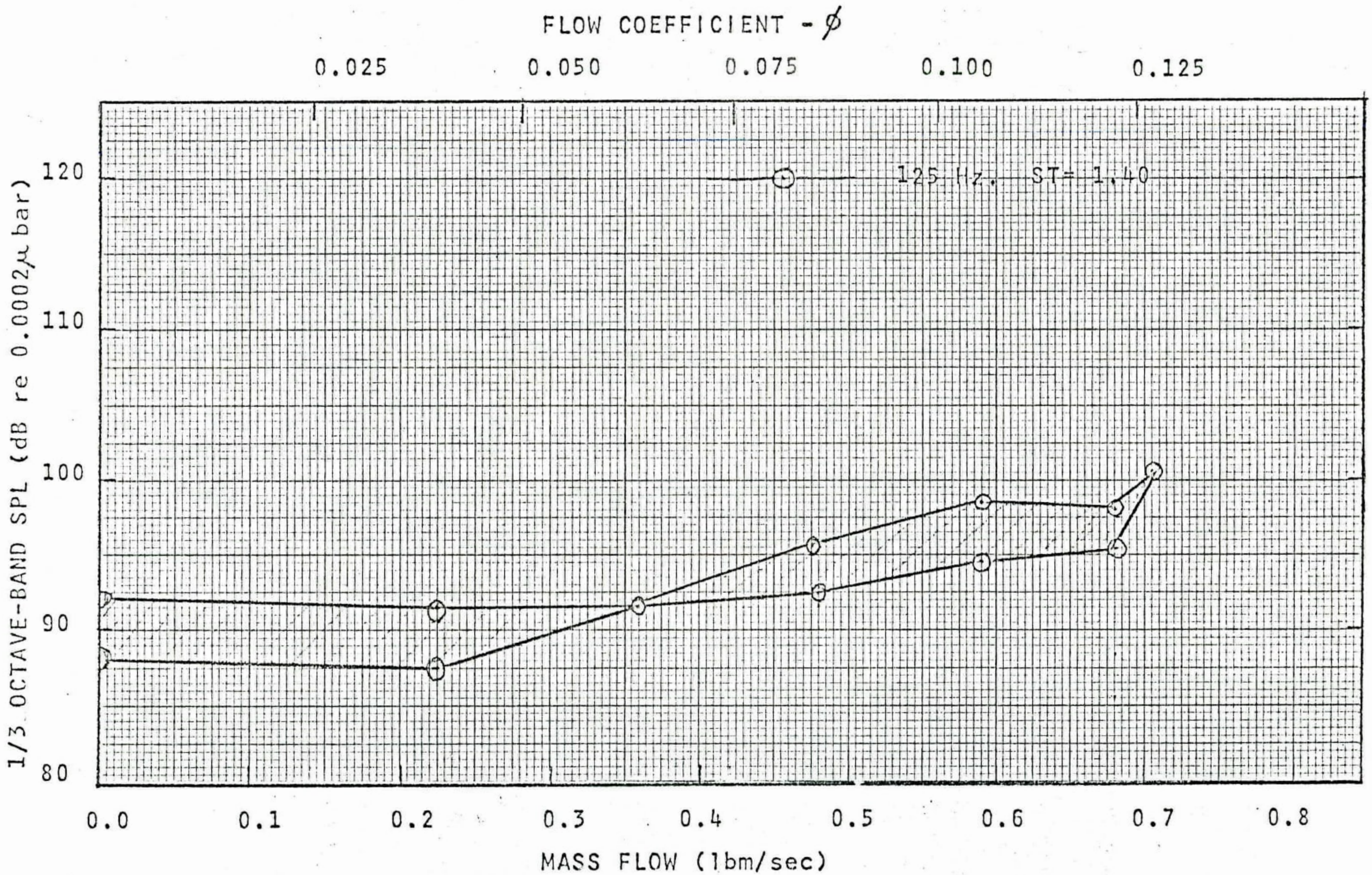

Figure 19. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 125 Hz. (1/3 H.P.Blower).

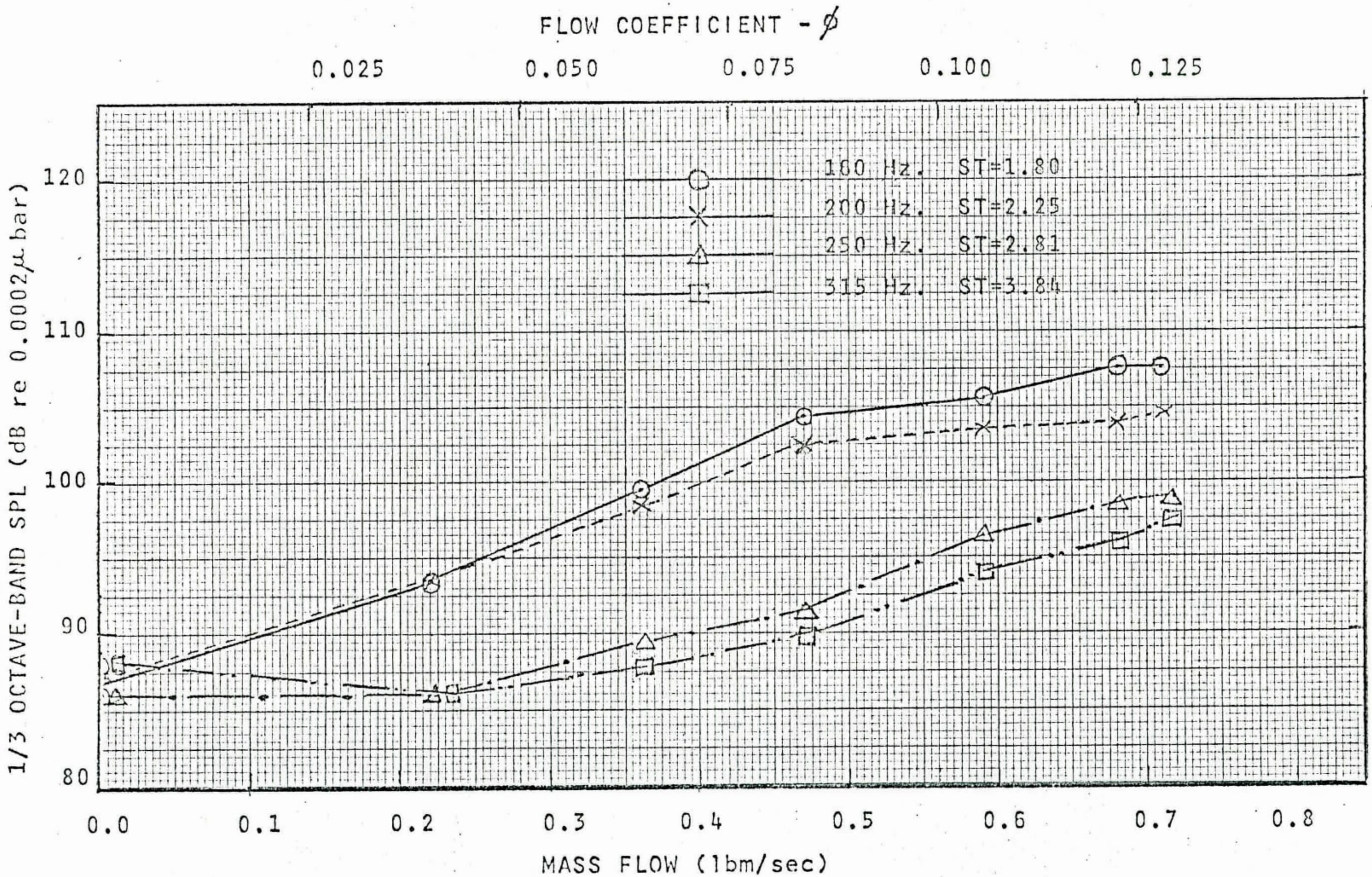

Figure 20. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 160,200,250 and 315 Hz. (1/3 H.P.Blower).

Fig 8

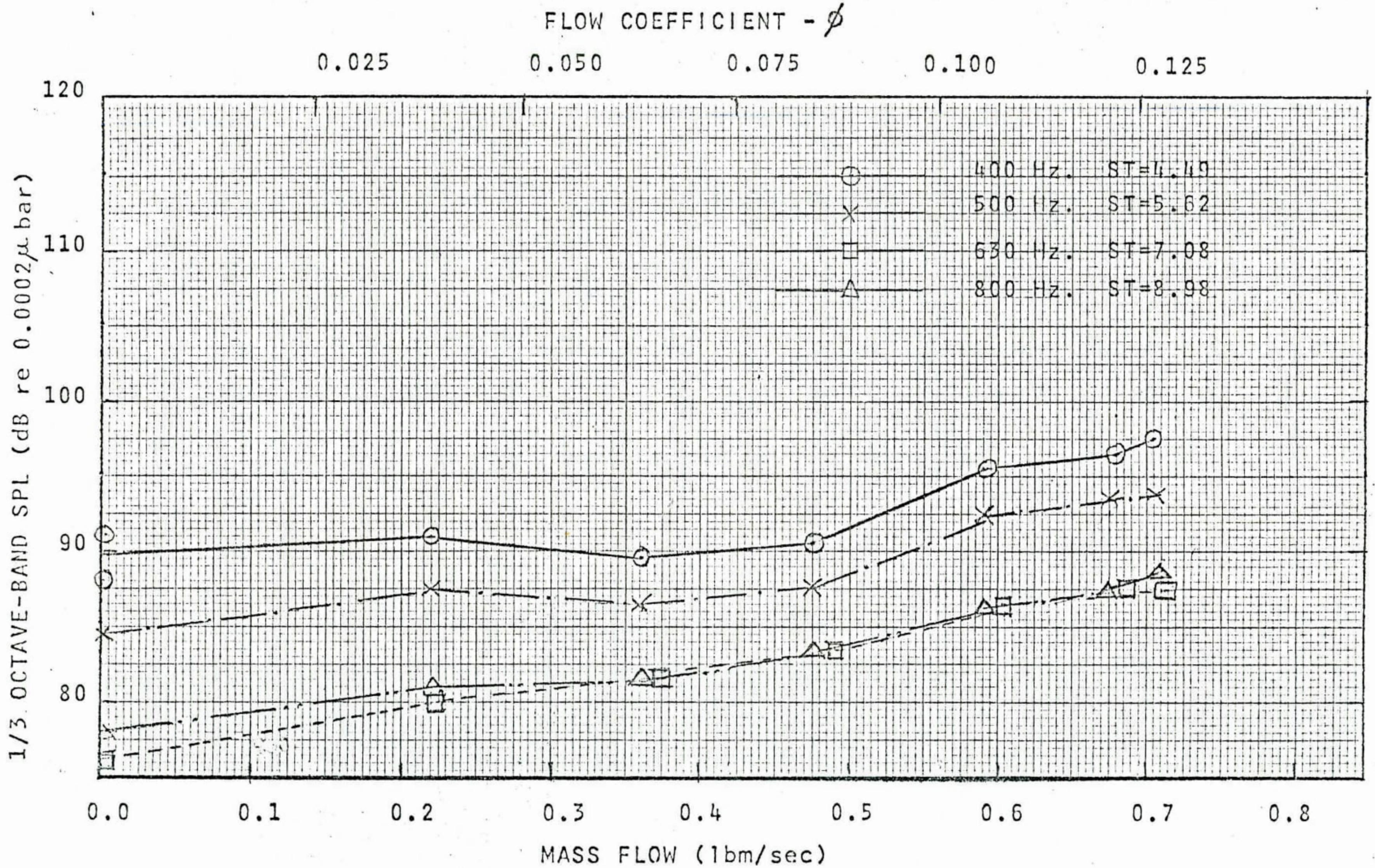

Figure 21. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 400,500,630 and 800 Hz. (1/3 H.P.Blower).

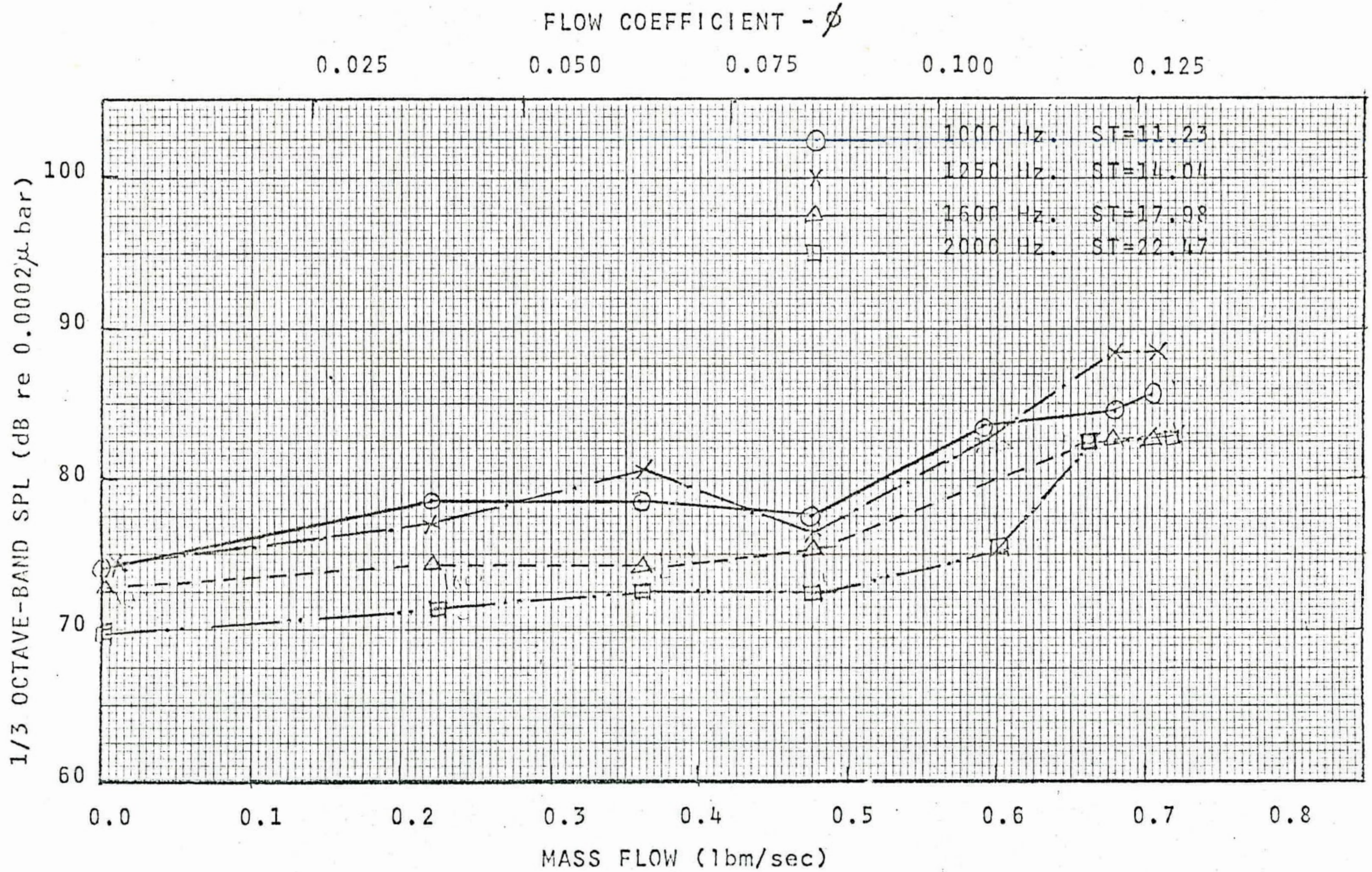

Figure 22. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 1000,1250,1600 and2000 Hz. (1/3 H.P.Blower).

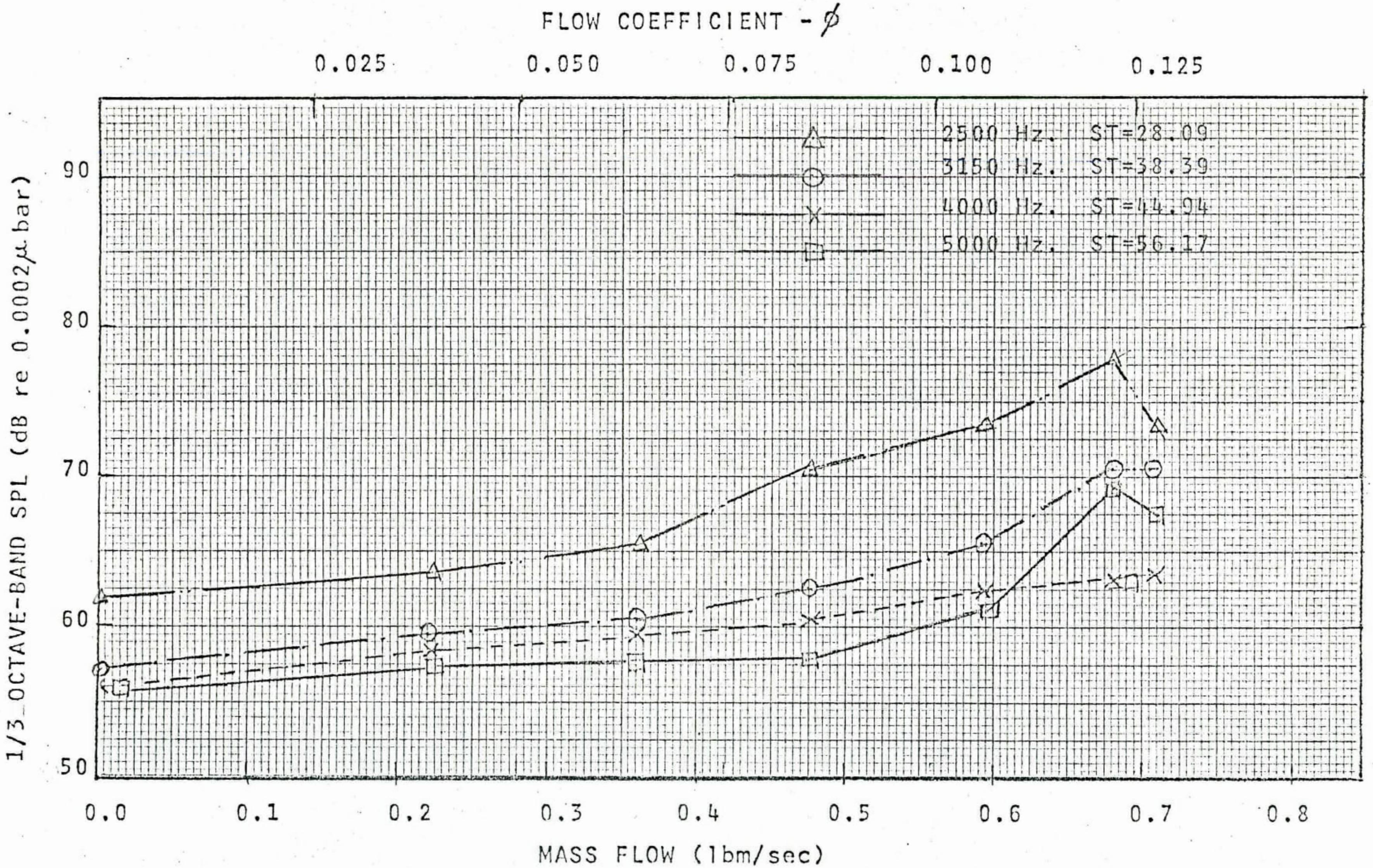

Figure 23. 1/3 Octave-Band Sound Pressure Level Variation with Flow at 2500,3150,4000 and 5000 Hz.(1/3 H.P.Blower).