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# TOTAL-PRESSURE AVERAGING IN PULSATING FLOWS

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## ABSTRACT

A number of total-pressure tubes were tested in a non-steady flow generator in which the fraction of period that pressure is a maximum is approximately 0.8, thereby simulating turbomachine-type flow conditions. Most of the tubes indicated a pressure which was higher ($P_{ind}/P_{true}$ up to 1.06) than the true average. Organ-pipe resonance which further increased the indicated pressure was encountered within the tubes at discrete frequencies. There was no obvious combination of tube diameter, length, and/or geometry variation used in the tests which resulted in negligible averaging error. A pneumatic-type probe was found to measure true average pressure, and is suggested as a comparison instrument to determine whether nonlinear averaging effects are serious in unknown pulsation profiles. The experiments were performed at a pressure level of 1 bar, for Mach number up to near 1, and frequencies up to 3 kHz.

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## SUMMARY

A number of total-pressure tubes were tested in a non-steady flow generator in which the fraction of period that pressure is a maximum is approximately 0.8, thereby simulating turbomachine-type flow conditions. Most of the tubes indicated a pressure which was higher ($P_{ind}/P_{true}$ up to 1.06) than the true average. Organ-pipe resonance which further increased the indicated pressure was encountered within the tubes at discrete frequencies. There was no obvious combination of tube diameter, length, and/or geometry variation used in the tests which resulted in negligible averaging error. A pneumatic-type probe was found to measure true average pressure, and is suggested as a comparison instrument to determine whether nonlinear averaging effects are serious in unknown pulsation profiles. The experiments were performed at a pressure level of 1 bar, for Mach numbers up to near 1, and frequencies up to 3 kHz.

## NOMENCLATURE

| | |
|---|---|
| a | acoustic velocity |
| $A_1$ | area of first orifice of pneumatic probe |
| $A_2$ | area of second orifice of pneumatic probe |
| $C_1$ | discharge coefficient of first orifice of pneumatic probe |
| $C_2$ | discharge coefficient of second orifice of pneumatic probe |
| D | tube inside diameter |

| | |
|---|---|
| $f$ | frequency |
| $L$ | tube length |
| $P_1$ | total pressure upstream of pneumatic probe |
| $P_2$ | total pressure at second orifice of pneumatic probe |
| $P_{c1}$ | total pressure at inlet of compressor stage |
| $P_{c2}$ | total pressure at outlet of compressor stage |
| $P_{ind}$ | average pressure indicated by total-pressure tube |
| $P_{max}$ | maximum applied total pressure |
| $P_{min}$ | minimum applied total pressure |
| $P_{true}$ | time-weighted average of applied pressure |
| $T_1$ | total temperature upstream of pneumatic probe |
| $T_2$ | total temperature at second orifice of pneumatic probe |
| $\beta$ | fraction of period that pressure is a maximum |
| $\eta_c$ | compressor stage efficiency |
| $\psi$ | efficiency error function |

## INTRODUCTION

The steady-state indications of total-pressure tubes placed behind a row of rotating blades are used to determine the performance of turbomachinery components. These tubes indicate a pressure which is assumed to be the true time-weighted average of the pulsating total pressure. However, when a total-pressure tube, connected to a pressure transducer through a length of tubing, is subjected to a pulsating pressure, the transducer indication has to be such that the net mass flow of gas in the tube is zero over a complete pulsation profile. In general, the equations governing the mass flow are not linear with respect to the applied pressure, and thus, the indicated pressure will not be the time-weighted average of the applied pressure pulsation.

The importance of being able to obtain an accurate time-weighted average pressure is shown in figure 1. Here the sensitivity of compressor efficiency error to accuracy of pressure ratio measurement is given as a function of compressor pressure ratio. At a pressure ratio of 1.4, a 1-percent error in outlet pressure (assuming no error in inlet pressure), results in a 3 percent error in efficiency. For the case of a fan with a pressure ratio of 1.15, the efficiency error would be greater than 7 percent for a 1 percent error in pressure. It is evident that average pressures are required with errors much less than 1 percent if compressor efficiency is to be determined to 1 percent.

In reference 1 both the analytical and experimental averaging problems were treated. The principal nonlinearities considered were the change in gas density between flow into and flow out of long tubes, as well as the square root of the pressure drop function for short tubes. However, this study was restricted to frequencies below 50 Hz, so gas inertial effects were negligible. Reference 2 reported experimental averaging studies at higher frequencies (over 6 kHz) downstream of a wheel consisting of small rotating rods. These experiments indicated that measurements with total-pressure tubes can either underestimate or overestimate the level of losses in stages of turbomachines. Gas velocities in reference 2 were generally below 100 meters per second. Reference 3 deals with techniques for generating known pressure pulsations. In experiments of this type, the most difficult task is obtaining a pulsation wave shape in which the time-averaged pressure is accurately known.

In the experimental tests reported herein, a stationary total-pressure tube is subjected to pressure pulsations that are generated by 20 small nozzles located in a rotating wheel. This simulates flow conditions at the exit of a turbomachine stage. Tubes of inside diameters up to 2 mm and lengths up to 63 cm were tested. The experiments were performed at a pressure level of 1 bar, for Mach numbers approaching the tube inlet of up to near 1, and frequencies up to 3 kHz.

A pneumatic probe (ref. 4) was also tested as a possible means of obtaining the true average pressure in an unknown pressure profile.

The main purpose of these experiments was to point out the types of errors that a user is likely to encounter, the sign and magnitude of these errors, and a suggested means of dealing with the problem.

## APPARATUS AND TESTS

### Total-Pressure Tubes

The inside diameter of the tubes tested were 0.42, 1.0, and 2.0 mm; inside-to-outside diameter ratios were 0.59, 0.67, and 0.80, respectively. Lengths varied from 1.6 to 63 cm. The diameters chosen span the sizes normally encountered in turbomachine testing. All of the tubes tested had squared-off ends. A few had 30-degree-half-angle internal bevels for use in studying effect of variation in end geometry. The end of the total-pressure tube closest to the pressure transducer extended into a larger tube whose cross-sectional area was at least 8 times larger than the tube area. The larger tube was about 1 meter long.

## Pneumatic Probe

The pneumatic probe consists of two sonic flow orifices in series and is shown schematically in figure 2. The probe is traditionally used for the measurement of total temperature (ref. 4).The total temperature $T_1$ in front of the probe is calculated by equating the mass flow rate through the two orifices and measuring the total pressures $P_1$ and $P_2$, and the temperature $T_2$. However, if the two temperatures are equal, or measurable, the probe can then be used to obtain the total pressure $P_1$ in front of the first orifice. In this latter mode, the probe can be used to obtain the time average of pulsating stream total pressure. For frequencies of pulsations of interest in this report (in the kHz range), the flow through the second orifice is essentially steady. Only the first orifice has an alternating component superimposed on its main flow. Before the probe can be used, it must be calibrated (for its value of the ratio $P_1/P_2$) in a steady gas stream of known total pressure and temperature, preferably over the range of Reynolds number of intended use.

In order to achieve sonic flow through both orifices, for the case where the pressure drop between the orifices is small, the diameter of the second orifice should be about 1.5 times the diameter of the first orifice. For the probe used in this report, the throat diameter of the first orifice was 0.5 mm, and the second was 0.8 mm. The pressure ratio across the entire probe should be greater than 4 to insure sonic flow through both orifices.

## Non-Steady Flow Generator

A cross-sectional view of the pulsating-flow generator is shown in figure 3. The generator consists of a rotating wheel containing 20 nozzles of 0.82 cm exit diameter placed around a 7.0 cm diameter circle. The space between nozzles is 0.27 cm. The stationary total-pressure tube is exposed to a high total pressure when it is in the jet flow, and close to room pressure when between jets. The nozzle size and spacing were chosen so that the pressure profile produced by the generator would be similar to that developed in turbomachines. The inlets of the total-pressure tubes were placed between 1 and 2 tube diameters from the face of the wheel.

The wheel is driven through belts and pulleys by a variable-speed motor. Wheel speeds range up to 9000 rpm (3000 jets/sec.). The airflow supplying the jets flows through a 8.9 cm diameter duct.

A small amount of leakage occurs between the end of the duct and the upstream face of the wheel due to a wheel clearance of about 0.5 mm. This leakage does not influence the performance of the wheel.

The 8.9 cm duct was also used as a free jet, with the wheel removed. The pneumatic probe was calibrated in the isentropic core of this jet at the exit of the duct.

## Tests

Low wheel speed. - For each test point of each total-pressure tube tested, a total-pressure survey was made at the exit of the nozzles at low wheel speed (approx 1 jet/min). This provided an accurate determination of the total-pressure profile to which the tube was subjected.

Integration of this profile provided the true time-weighted average of applied pressure. Tests showed that the true average remained essentially constant as wheel speed was increased.

High wheel speed. - Without moving the tube under test, the wheel speed was then increased while continuously recording the tube indication. This indication, over a range of frequencies, was compared with the average obtained by integration of the low-speed profile.

The upper limit of frequency for high values of $(P_{max} - P_{min})/P_{min}$ was limited by the mechanical drive system of the wheel to about 3 kHz. For lower values of $(P_{max} - P_{min})/P_{min}$, the upper limit of frequency was restricted below 3 kHz to insure that the tangential component of the flow velocity remained small (less than 20 percent) compared to the axial component.

Pneumatic probe. - The pneumatic probe was tested in the same manner as the total-pressure tubes. However, the probe indication had to be multiplied by the calibration factor $P_1/P_2$ in order to obtain the indicated average pressure, $P_{ind}$.

## Accuracy

Most of the results presented herein are in the form of $(P_{ind} - P_{true})/P_{true}$ vs frequency. Frequency was determined with negligible error. The true average pressure was calculated from the following expression.

$$P_{true} = P_{min} + \beta(P_{max} - P_{min}) \qquad (1)$$

The inaccuracy of each of the three terms in the expression contribute about 0.2 percent limit of error in $P_{true}$. $P_{ind}$ was determined to within

0.1 percent limit of error. Combining these errors according to the root-sum square formula results in an overall probable error in the ratio $(P_{ind} - P_{true})/P_{true}$ of about 0.002.

## RESULTS AND DISCUSSION

### Pressure Pulsation Profile

A typical example of the pressure profile at the exit of the total pressure pulsation generator is shown in figure 4. Although the profiles of only 3 jets are shown, the shapes of the other 17 are identical with those shown to within the accuracy of the experiment. The steepness of the pressure profile, for a given value of $(P_{max} - P_{min})/P_{min}$, depends on two factors: the axial location of the inlet of the total pressure tube and its diameter. The closer the tube to the face of the wheel and the smaller the tube, the steeper the profile. Generally, the inlet of the tube was maintained between 1 to 2 tube diameters from the wheel face.

The fraction of period that the pressure is a maximum, $\beta$, for the wheel used, is approximately 0.8. This value is about the same as would be encountered behind the rotating blades of an axial flow compressor or turbine. The valleys in the profile simulate wakes downstream of a blade row. It should be emphasized that results presented in the following sections apply only for a value of $\beta$ near 0.8. The indication of a total pressure tube in relation to the true time averaged pressure is a function of $\beta$, and results for small values of $\beta$ can actually be of reversed sign from those of high values of $\beta$ (ref. 1).

The value of $(P_{max} - P_{min})/P_{min}$ was varied by varying the Mach number of the flow through the nozzles. Mach numbers ranged from 0.28

for $(P_{max} - P_{min})/P_{min}$ of 0.05, to 0.98 for $(P_{max} - P_{min})/P_{min}$ of 0.85. $P_{min}$ was approximately room pressure because the flow from the nozzles issued into the room.

The pressure profile used for these tests may be more severe than that encountered in turbomachines, because in the pulsation generator the flow varies from its full value at $P_{max}$ to zero at $P_{min}$. In a turbomachine, there normally would be a flow associated with $P_{min}$.

## Laminar Example

Figure 5 shows the averaging error of a small diameter (0.42 mm) total pressure tube in which the flow remained laminar over the range of $(P_{max} - P_{min})/P_{min}$ and frequencies tested. Reynolds numbers of the flow within the tube varied from a few hundred to a few thousand. The ordinate is expressed as the ratio of the difference between the tube and the true averaged pressure divided by the difference between the maximum and minimum pressures of the imposed profile. Several features of the figure are of interest. First, the incompressible flow case is represented by the curve for $(P_{max} - P_{min})/P_{min}$ equal to 0.05. The averaging error for this case is about 2 percent at the low frequencies and decreases to zero at 600 Hz. This is the pattern which is generally assumed for total pressure tubes used in turbomachines. At the low frequencies, the density of the flow into the tube is higher than the density of the flow out of the tube resulting in an indicated pressure which is higher than the true average. At some higher frequency, the gas column within the tube acts like a solid mass, resulting in slug or piston-like flow. The tube then indicates the true average for small values of $(P_{max} - P_{min})/P_{min}$. The curves also have peaks which are associated with standing waves or resonance, resulting in an additional error being superimposed on the basic curves. For the example shown in figure 5, the calculated fundamental frequency for an open tube is 1700 Hz compared with 1500 Hz obtained experimentally. As the value of $(P_{max} - P_{min})/P_{min}$ increases, the error no longer becomes zero at the higher frequencies, but remains positive due to nonlinear averaging.

In figure 5 the averaging error is expressed in terms of $(P_{max} - P_{min})$. Expressing the error in this manner tends to collapse the data for different values of $(P_{max} - P_{min})/P_{min}$. However, in testing turbomachines, quantities such as pressure ratio and efficiency are of prime importance, so it is more practical to express the averaging error in terms of the pressure ratio, $P_{ind}/P_{true}$. The results of figure 5, expressed in terms of $(P_{ind} - P_{true})/P_{true}$, are shown in figure 6. In this form the averaging error is negligible for the incompressible case. However, at moderate values of $(P_{max} - P_{min})/P_{min}$ the error can be significant. The value of $(P_{max} - P_{min})/P_{min}$ of 0.42 in the figure corresponds to a Mach number of about 0.7 in the region of flow at $P_{max}$. At the highest value of $(P_{max} - P_{min})/P_{min}$, the averaging error reached a value of 6 percent during resonance.

In general, the values of averaging error at $f \approx 0$, for the case of laminar flow, agree with those predicted by reference 1. For the tube of figure 6, the agreement is good for the three lower values of $(P_{max} - P_{min})/P_{min}$, but reference 1 underestimates the experimental value by $1\frac{1}{2}$ percent for $(P_{max} - P_{min})/P_{min} = 0.85$.

The tube resonance phenomenon for small diameter tubing is more clearly illustrated in figure 7 in which the averaging error is plotted against the parameter $2fL/a$ for three different lengths of tubes. An abscissa value of 1 represents the fundamental frequency for an open tube. This resonance is most clearly shown for the 10-cm-long tube where the averaging error has increased from 0.01 to 0.04 at resonance. The smaller peak at half the fundamental frequency probably represents the second harmonic of the forcing function. The longer tube (40 cm) exhibits considerable damping while still peaking at an abscissa value near 1, and traces of the higher harmonics are barely discernible. In general, the longer tubes have a more positive error at the higher frequencies with less severe resonant peaks, while the shorter tubes (e.g., 2.5 cm) may even indicate a pressure actually lower than the true average during nonresonance.

## Turbulent Example

The averaging error for two different lengths of 1.0 mm diameter tubes is shown in figure 8. Although the Reynolds number for this case implies that the flow is turbulent, the curves have the same trend as the laminar case. That is, the shorter tube has a higher initial averaging error, but produces a lower error at the higher frequencies than does the longer tube. A main feature of the results for larger diameter tubes is the persistence of the resonance phenomenon. In figure 8 even the 9th harmonic is discernible for the 63 cm tube. This figure also demonstrates that the resonance is more pronounced for the shorter tubes presumably because damping is lower. Based on this result, the indication of a short total-pressure tube resonating at its fundamental frequency in a turbomachine would be unreliable.

## Effect of Variation in Tube Geometry

Several variations in tube-entrance or tube-exit geometry were tested to determine the effect on the averaging error. Figure 9 shows the effect of three combinations of beveled inlet tubes compared to the simple square ended tube. Beveling is desirable when there is uncertainty in the flow direction approaching the tube inlet. A 30-degree-half-angle bevel (the angle used in figure 9), increases the flow direction insensitivity from 10 degrees for the simple tube to about 20 degrees. The results for all 4 tubes fall within the shaded band in figure 9. The width of the band is less than 0.5 percent in regions of no resonance, and 1 percent or less in peaking regions of resonance.

Figure 10 presents the effects of three alterations to a 2.0 mm diameter tube, 63 cm long, compared with its original averaging error. The original tube showed numerous sharp resonances - ten are visible in figure 10. Terminating the tube with either a porous block or an orifice reduced the resonance phenomenon but the averaging error at the high frequencies remained about 2 percent high. When the tube is filled with a bundle of fine wires, it behaves in the same manner as a long tube with laminar flow, and exhibits only a small peak at the fundamental resonance. Its averaging curve would be about the same as the 40 cm curve of figure 7, if plotted with the same abscissa.

It is evident from the laminar, turbulent, and geometry-variation examples, that there is no obvious combination of tube diameter and length which will have negligible averaging error; at least for the imposed pressure profiles of these tests.

Pneumatic Probe

A measurement made with the pneumatic probe is presented as a suggested method of determining whether there actually is a total-pressure averaging problem in any given turbomachine experiment. By locating the pneumatic probe at the same radial position as the total-pressure tube of interest, a direct comparison can be made in the actual pressure profile of intended use.

The initial calibration factor $(P_1/P_2)$ of a pneumatic probe is first obtained in a steady-flow facility by comparing the pressure $P_2$ (fig. 2) with the pressure $P_1$ measured by a total-head tube (same as pressure $P_2$ with zero flow through the probe). Such a calibration is shown in figure 11. Thereafter, to obtain the stream total pressure, the pressure $P_2$ in front of the second orifice is measured. This measurement, multiplied by the predetermined ratio $P_1/P_2$, yields the total pressure in front of the probe. If there is a variation in gas temperature between the two orifices, this must be taken into account in the calculation of total pressure. For the case of constant ratio of specific heats, the following formula may be used.

$$P_1 \doteq P_2\left[\frac{C_2A_2}{C_1A_1}\right]\sqrt{\frac{T_1}{T_2}} \qquad (2)$$

$\left[\frac{C_2A_2}{C_1A_1}\right]$ represents the initial calibration factor.

The value of $P_1/P_2$ in the steady-state calibration in figure 11 remained constant over the range of pressures tested. Depending on the shape of the orifices in the pneumatic probe, there could be a small change in $P_1/P_2$ with pressure. It is difficult to control the shape of such small orifices (less than 1 mm throat diameter); consequently, the two orifices could have different coefficients of discharge at the same Reynolds number. This would result in a small variation in the steady-stage calibration factor $(P_1/P_2)$ with pressure (eq. (2)).

It is desirable for the first orifice to have a bell-mouthed inlet. Such a shape will make the probe insensitive to flow direction over ±20 degrees.

The pneumatic probe used in the present tests had a first orifice diameter of 0.5 mm and was installed in a tube of 1.5 mm outside diameter. The calculated total pressure obtained from the probe, when subjected to the flow-generator pressure pulsations, was the true average pressure to within the accuracy of the experiment. In addition, no resonance was observed in the recorded output of the pressure in front of the second orifice over the range of frequencies tested.

The pneumatic probe is more complicated to use than a total-pressure tube, so it is not proposed as a substitute for the simpler device. Rather, it can be used as a comparison instrument during initial testing in situations where an averaging problem is suspected.

## CONCLUDING REMARKS

A number of total-pressure tubes were tested in a nonsteady flow generator in which the fraction of period that pressure is a maximum is approximately 0.8. Most of the tubes indicated a pressure which was higher than the true average. Values of $P_{ind}/P_{true}$ ranged up to 1.06. Organ-pipe resonance was encountered within the tubes. During resonance, nonlinear averaging increased the indicated pressure by as much as 3 percent. The resonance amplitude could be reduced by inserting restrictions within or at the exit of the tubes; however, the restrictions changed the shape as well as the level of the curve of indicated pressure versus frequency. There was no obvious combination of tube diameter, length, and/or geometry variation, which had negligible averaging error.

The true average pressure was determined with a pneumatic probe using two sonic-flow orifices. Such a probe, although impractical for routine testing, is useful to determine whether nonlinear averaging effects are serious in a given experimental situation.

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1. Johnson, R. C., "Averaging of Periodic Pressure Pulsations by a Total-Pressure Probe," TN 3568, 1955, NACA.
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4. Warshawsky, I. and Kuhns, P. W.: Review of the Pneumatic-Probe Thermometer. Temperature - Its Measurement and Control in Science and Industry. Charles M. Herzfeld, ed., Reinhold Publ. Corp., vol. 3, part 2, 1962, pp. 573-586.

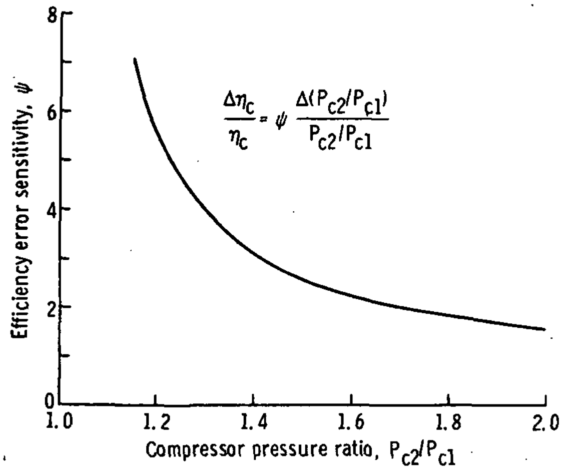


Figure 1. - Variation of compressor efficiency error sensitivity with pressure ratio.

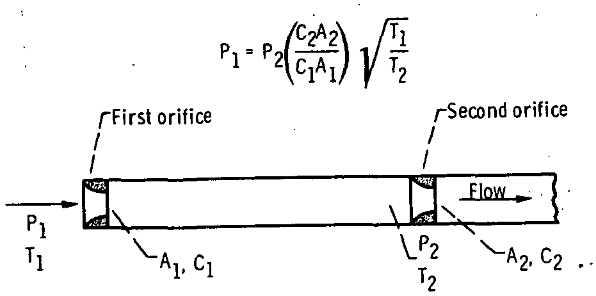


Figure 2. - Schematic of pneumatic probe.

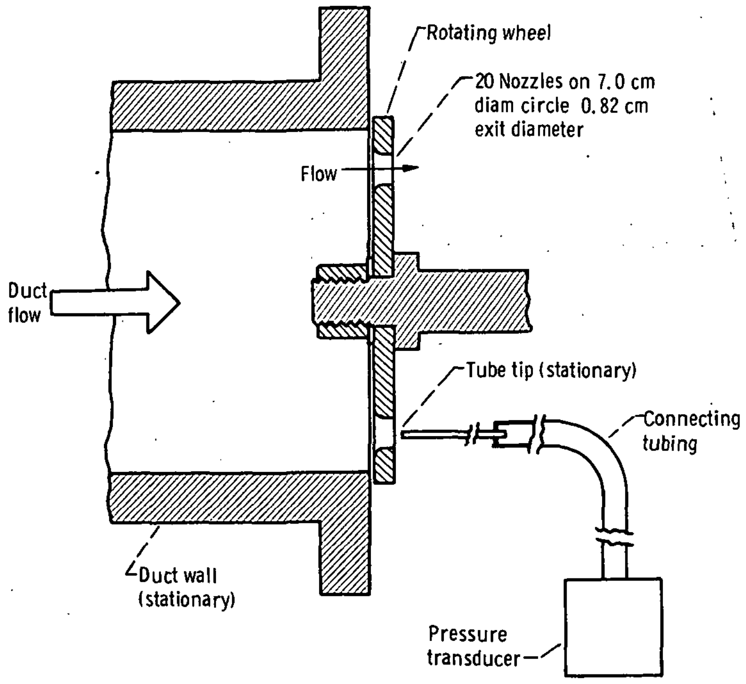


Figure 3. - Detail of total-pressure pulsation generator.

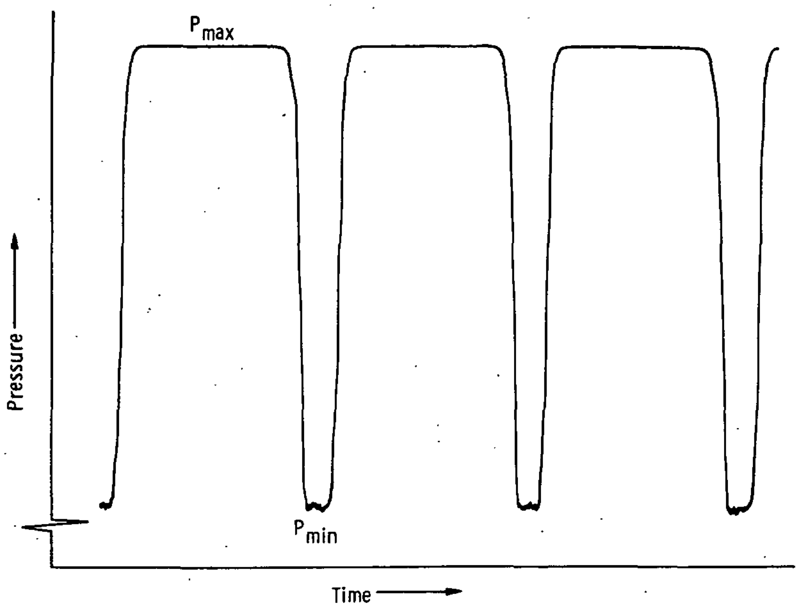


Figure 4. - Actual wave form of applied pressure. Tube diameter, 1.0 mm; $(P_{max} - P_{min})/P_{min}$, 0.26; β, 0.79.

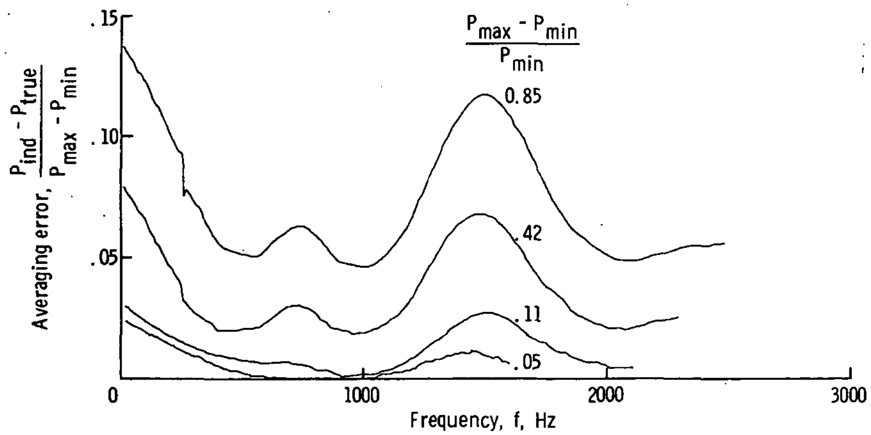


Figure 5. - Variation of total-pressure tube averaging error with frequency for a range of $(P_{max} - P_{min})/P_{min}$. Tube diameter, 0.42 mm; length, 10 cm; β, 0.75.

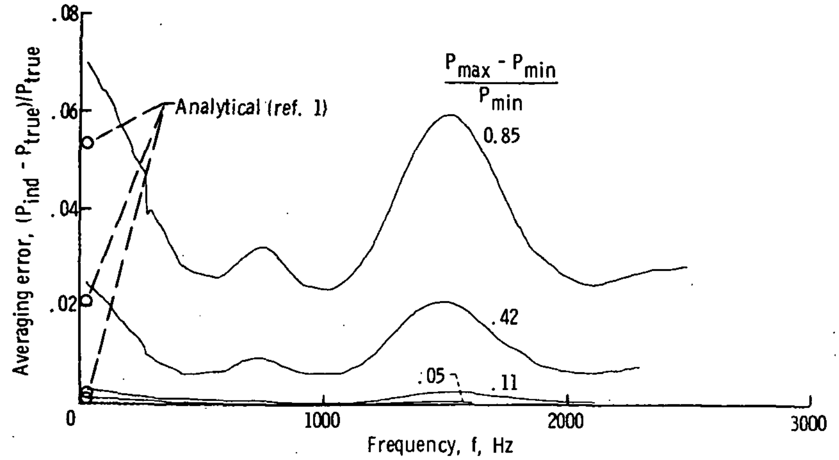


Figure 6. - Variation of total-pressure tube averaging error with frequency for a range of $(P_{max} - P_{min})/P_{min}$. Tube diameter, 0.42 mm; length, 10 cm; $\beta$, 0.75.

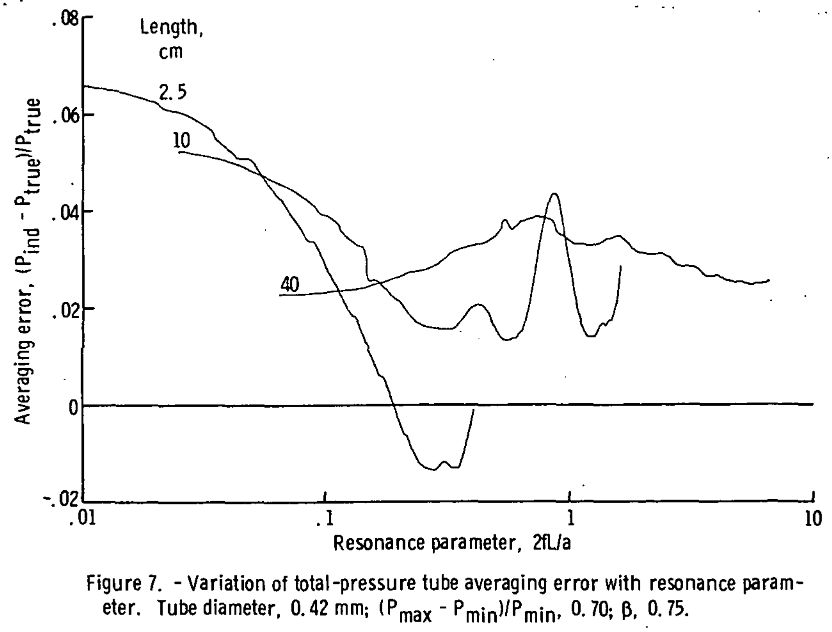


Figure 7. - Variation of total-pressure tube averaging error with resonance parameter. Tube diameter, 0.42 mm; $(P_{max} - P_{min})/P_{min}$, 0.70; $\beta$, 0.75.

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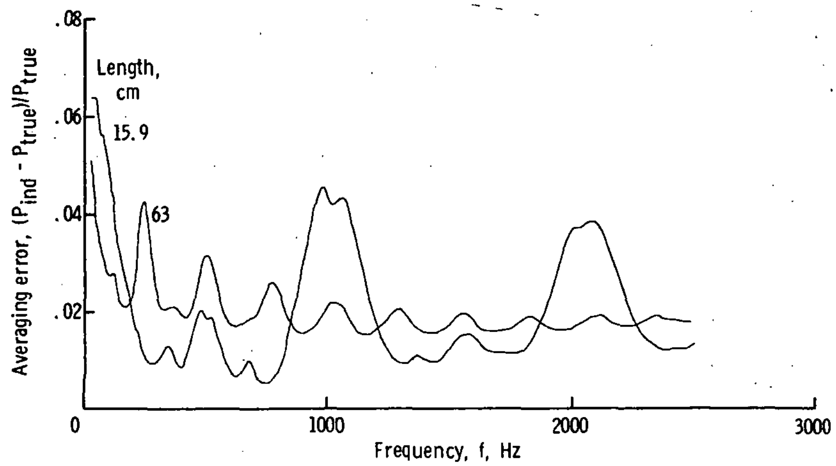


Figure 8. - Variation of total-pressure tube averaging error with frequency. Tube diameter, 1.0 mm; $(P_{max} - P_{min})/P_{min}$, 0.70; β, 0.79.

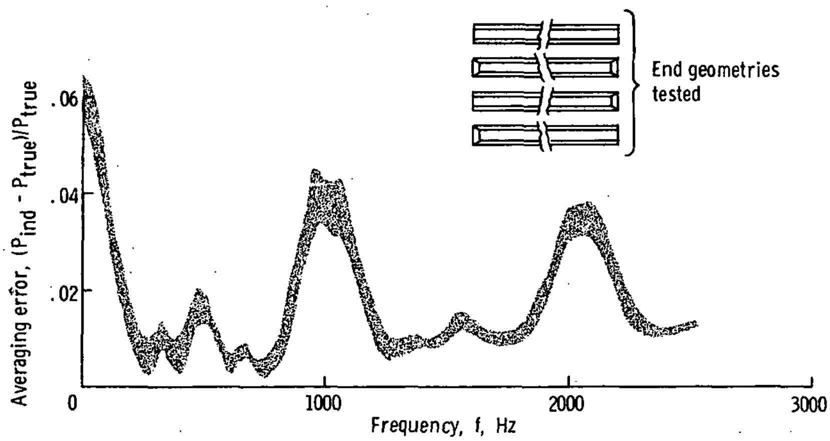


Figure 9. - Range of variation of total-pressure tube averaging error with frequency for 4 different end geometries. Tube diameter, 1.0 mm; length, 15.9 cm; $(P_{max} - P_{min})/P_{min}$, 0.70; β, 0.80.

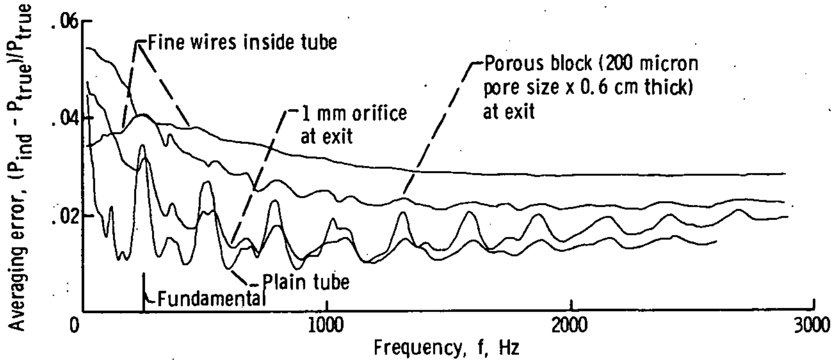


Figure 10. - Variation of total-pressure tube averaging error with frequency for changes in tube termination and internal geometry. Tube diameter, 2.0 mm; length, 63 cm; $(P_{max} - P_{min})/P_{min}$, 0.68; β, 0.83.

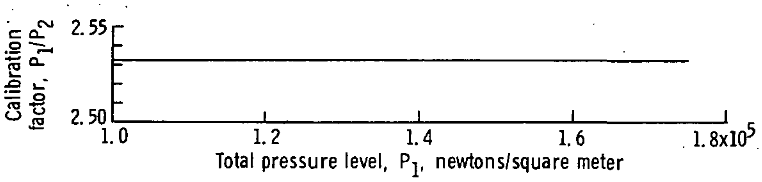


Figure 11. - Pneumatic probe calibration.

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