THE UNIVERSITY OF OKLAHOMA

GRADUATE COLLEGE

THE EFFECT OF WALL TEMPERATURE ON THE OPERATION OF THE RIJKE TUBE

AF 33(608) 1096

A THESIS

SUBMITTED TO THE GRADUATE FACULTY

in partial fulfillment of the requirements for the

degree of

MASTER OF ENGINEERING

BY

QUITMAN W. LOTT

Norman, Oklahoma

1965

THE EFFECT OF WALL TEMPERATURE ON THE
OPERATION OF THE RIJKE TUBE

A THESIS
APPROVED FOR THE SCHOOL OF
AEROSPACE AND MECHANICAL ENGINEERING

BY

____

____

____

ACKNOWLEDGMENTS

The author wishes to express his sincere appreciation to Dr. Darrel G. Harden, of the School of Aerospace and Mechanical Engineering, for his constant encouragement and assistance in his direction of this study.

In addition the author wishes to express his appreciation to his friend, John R. Bode, who introduced the author to this subject and provided many helpful ideas, his wife, Betty, for her encouragement and understanding, Dewayne Nelson for his many ideas and technical assistance in the construction of the apparatus, and the United States Air Force for making this work possible.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF ILLUSTRATIONS

THE EFFECT OF WALL TEMPERATURE ON THE OPERATION OF THE RIJKE TUBE

CHAPTER I

INTRODUCTION

In 1859, Rijke(8) obtained a heat-driven sound from a round tube held in a vertical position by placing a grid one-fourth of the way from the bottom of a 0.8 meter long tube and heating the grid to a red heat with a gas flame. When the flame was removed, an intense note sounded. The frequency of the note was observed to be near the fundamental frequency of the tube. Later it was observed that the tube would sound in a horizontal position if air were forced through the tube at the proper velocity. These heat driven oscillations in the simple tube are now recognized to be caused by the same mechanism which results in various types of combustion instabilities. In their mild form these instabilities are undesirable and can become quite destructive in their more severe forms. Better known examples of these instabilities occur as rumble and afterburner screech in jet engines and in rocket engine instabilities.

The purpose of this study was to construct an experimental Rijke tube apparatus and investigate the effect of the parameters, velocity, heater power, and wall temperature, with primary emphasis on wall temperature. In earlier exploratory experiments the wall temperature seemed to be an important parameter. In addition, Carrier(1) indicated that heat conduction near the wall is an important source of energy loss, and Maling(3) said that heat conduction losses near the wall must be accounted for.

Literature Survey

Extensive surveys of the literature which include the early work on the heat-driven sound tube have been completed by Neuringer and Hudson(5) and by Putnam and Dennis(6).

Neuringer and Hudson assumed that the heat-driven oscillations could occur only under turbulent flow conditions. In addition they hypothesized that the turbulence in the flow is of such a nature that it is coupled to the acoustic vibrations in the tube. An example calculation was included to show a Reynolds number of 3700 based on a flow velocity of 50 cm/sec. It shall be shown that sound is possible with gas velocities as low as 5 cm/sec. The Reynolds number would thus be much less than the 2300 necessary for the onset of turbulent flow. Velocity, pressure, and density were written in terms of a uniform component plus a small perturbation due to oscillations. The energy, momentum, and mass conservation equations were solved assuming a step variable change across the heater interface. From the solution it was shown that the tube would sound only if the heater were in the upstream half of the tube. No numerical calculations were made.

Carrier(1) treated the sound wave upstream of the heater and the wave downstream of the heater separately. The wave upstream of the heater was assumed to be essentially isentropic. The wave downstream of the heater was broken up into isentropic and non-isentropic components. Calculations were included to show that the sound wave reflects from 0.613R outside the end of the tube. The effective length of the tube is thus the length plus 0.613D, where D is the diameter of the tube. The heater response was estimated by assuming a semi-infinite plate. A numerical calculation was included showing that for a velocity of 1.7 ft/sec and a heater temperature of about 800°F the tube should sound for a heater ribbon width of 0.08 cm. No information was included about the required length of the ribbon. The occurrence of harmonics of the fundamental frequency was verified theoretically.

Putnam and Dennis(6) proposed a theory for heat-driven oscillations based upon a phase lag between changes in the flow rate and changes in the heat transfer rate. Part of their theory is justified by inference and more work needs to be done before it can be proved or disproved. Acoustic oscillations produced when a flame burns from a grid in an eight-inch diameter chamber were studied. Results are shown for different positions and angles of the grid.

Trilling(9) examined the sound fields produced in a real gas by boundary temperature variations to determine how the pressure, temperature, and vorticity modes of motion interact.

Merk(4) used the laws of conservation of momentum and energy to match the acoustic fluctuations across the heat source and the law of conservation of mass to determine the non-isentropic fluctuations. The characteristic equations governing the excitation of sound in the tube are derived by introducing transfer functions of the heat source and acoustic admittances and impedances. The transfer functions were derived for a heating element composed of thin spirally-wound wire. Neutral curves were calculated for the fundamental and first harmonic frequency. A numerical calculation was included to show that the minimum velocity for oscillations to occur is 2 cm/sec. The author concluded that this was too low as Neuringer and Hudson reported a minimum velocity of 50 cm/sec. It shall be shown that the 2 cm/sec can be approximated experimentally.

Maling(3) started his analysis with a wave equation for a medium containing a heat source. The dependence of the onset of oscillations on the mean flow and heater power is derived. Experiments were conducted to check the theoretical work. For heater powers of 930, 1100, and 1290 watts the calculated maximum velocities were 0.71, 0.83, and 0.92 m/sec, respectively with corresponding measured velocities of 0.84, 0.87 and 0.91 m/sec. Maling reported that as the velocity was reduced below 0.4 m/sec the oscillations abruptly ceased. At this velocity the mean flow was reported to be unstable. Maling surmised that the instability was possibly due to his experimental apparatus. According to Maling, Kerwin(2) reported mean flow instabilities, but the instabilities occurred at smaller velocities.

A problem encountered when reviewing the literature on the Rijke tube was that the equations were presented in such a manner that verification of the equations or independent calculations were possible in only very restricted instances.

CHAPTER II

DESCRIPTION OF EXPERIMENTAL APPARATUS

The experimental apparatus was constructed as shown in figures 1A and 1B. The principal parts of the apparatus are the sound tube, heating element with the power supply, air flow measuring system, and the air supply system.

The tube chosen for this experiment was a copper tube with a nominal inside diameter of 3.906 inches with a 3/16 inch wall thickness and a length of four feet. That particular size tubing was chosen because similar size tubing was used by previous investigators of the Rijke tube phenomenon. Also, heating elements can be more easily constructed for a large diameter tubing.

Iron constantan thermocouples were attached to the tube walls in the locations shown in figure 2. The thermocouples were arbitrarily placed axially to prevent biasing of the temperature measurement. As the inside wall temperature of the tube was desired, 1/16 inch holes were drilled almost through the tube wall, and the ends of the thermocouple were placed in the holes and held in place with liquid solder. Two additional thermocouples were provided to measure the air temperature inside the tube. One thermocouple was positioned

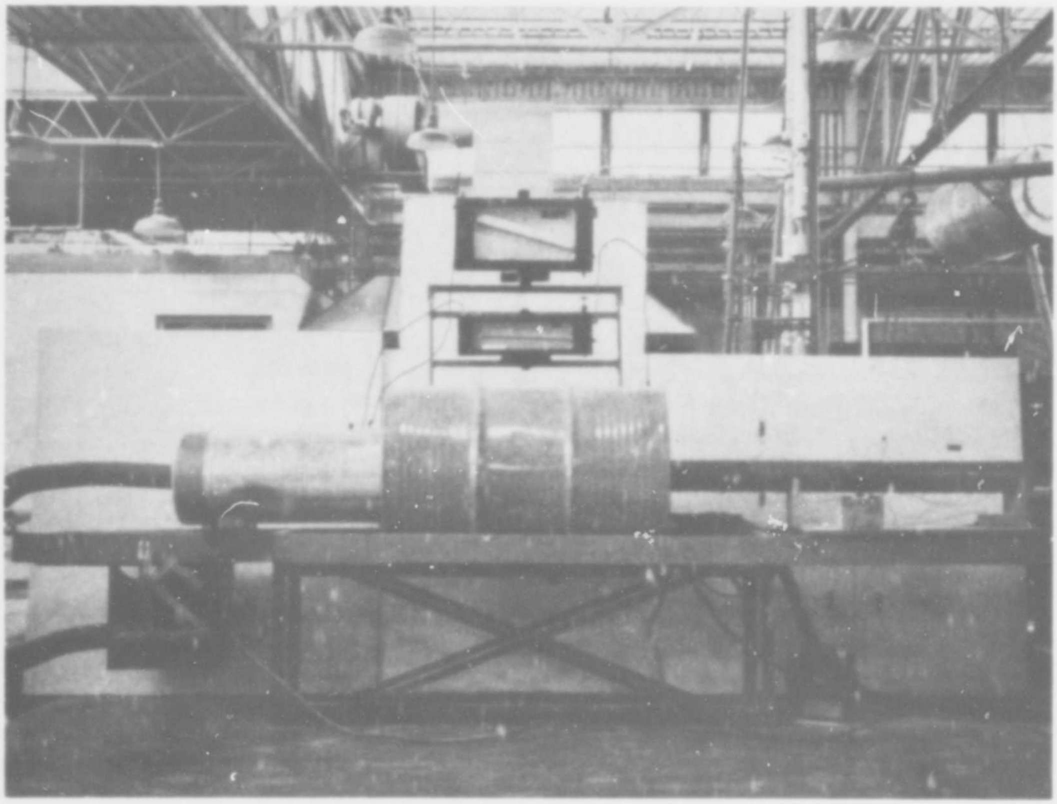


Figure 1A. Photograph of Test Apparatus

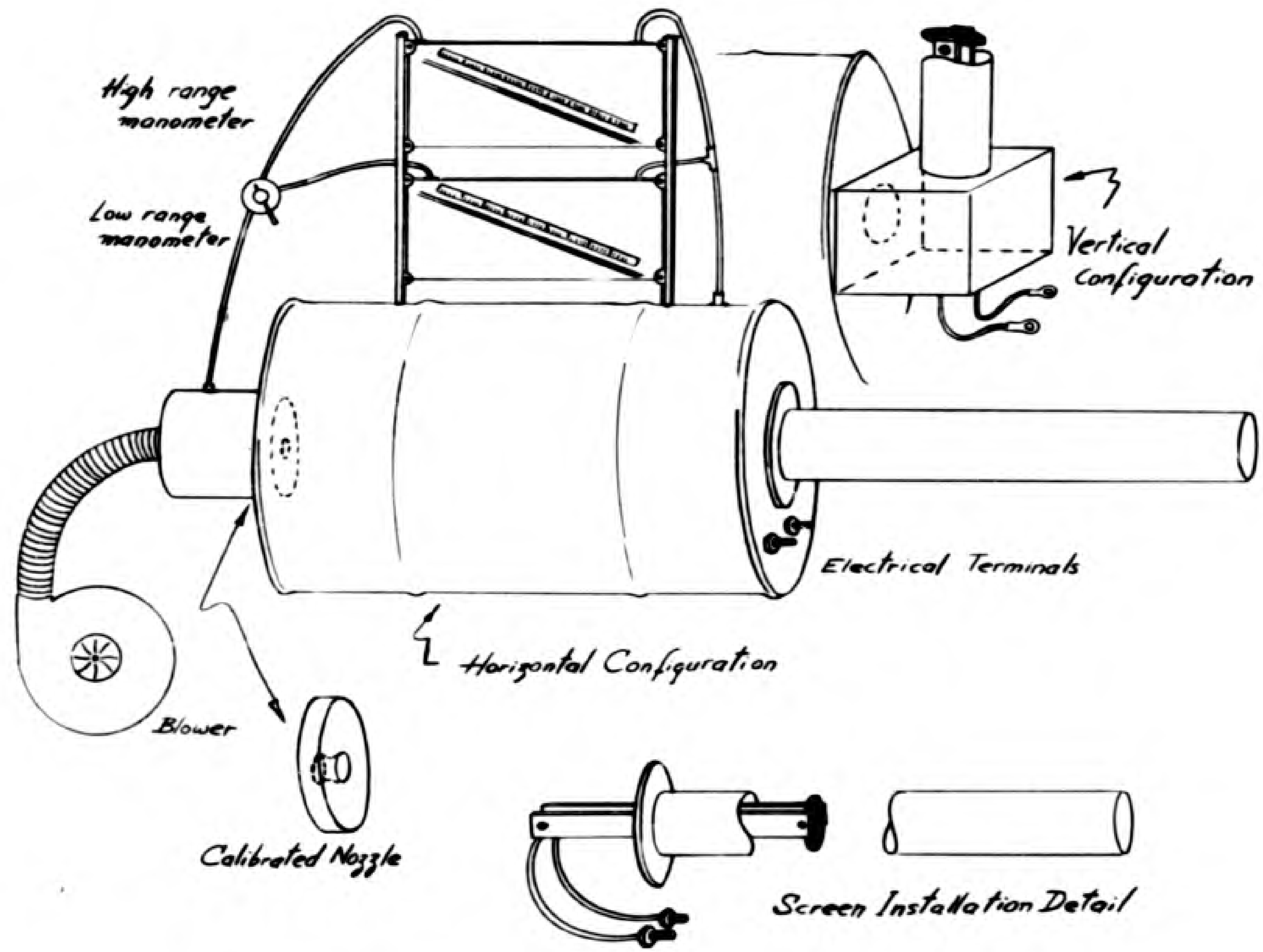


Figure 1B. Schematic of Test Apparatus

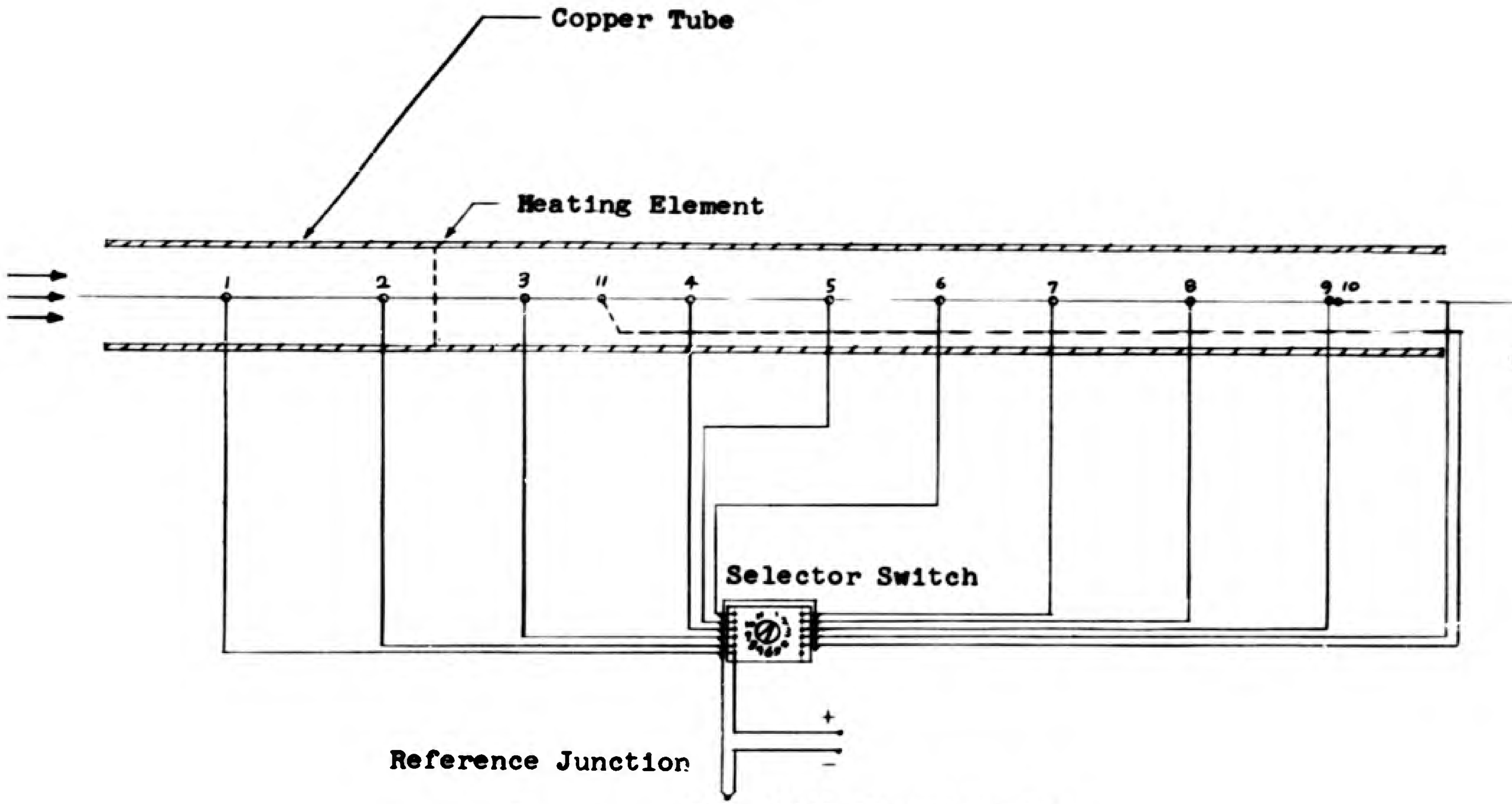


Figure 2. Tube With Thermocouples

to measure the temperature in the center of the tube six inches downstream of the heating element. The other thermocouple was positioned to measure the air temperature in the center of the tube four inches inside the exit end of the tube. A potentiometer was used in conjunction with an eleven-position junction box and a 32°F reference bath to read the thermocouple outputs.

The intensity of the sound was measured with a type 1551-A sound-level meter manufactured by General Radio Company. The calibration of the meter was not known. Therefore the sound intensity measurements were only relative. The attenuator was placed in the C-Flat position for all measurements to make the results comparable.

As it was desirable that the tube be easily disconnected from the plenum chamber to allow access to the heating element, an adapter ring was attached to the plenum chamber and a rubber seal provided to prevent air leakage around the tube. The tube could be easily removed by sliding the tube straight out of the opening and off the heating element. An adapter was constructed from plywood to allow the air measurement system to be used with the tube installed in a vertical position. Fiberglas insulation was installed on the tubing when it was desired to operate the tube at high wall temperatures. For the experiments which were to be conducted with the tube wall at a constant temperature, an air distribution system was constructed from copper tubing to direct high pressure air along the length of the tube wall. The compressed air provided satisfactory cooling for all except very high heater power settings.

Experience indicates that the construction of the heating element is critical. Initially it was desired to use a 30 ampere, 220 volt variac for a power supply, so many attempts were made to construct a suitable heating element which would draw a maximum of 30 amperes. First an annular-shaped frame was moulded from clay. Holes were drilled, and ten feet of 0.025 inch heating element wire was strung in the frame in the manner indicated in figure 3. No sound could be obtained with the element although a wide range of air velocities was tried, and heater powers up to 1000 watts were used. Next a ribbon type heating element was constructed as illustrated in figure 4. The two rings were made from aluminum. Holes for the rods were drilled each 1/4 inch. The rods which support the ribbon had to be nonconducting and resistant to high temperatures, so solid silicon glass rod was used. Many failures were experienced in which different lengths of ribbon and different methods of lacing the ribbon were tried. The ribbon available was 1/32 inch wide. In order to get more ribbon in the element 25 feet of the ribbon was laced double so that the effective width of the element was 1/16 inch. The ribbon was laced in three planes 1/4 inch apart. This element was tried with heater powers of up to 2000 watts, and on only one trial was any sound obtained. The trial could not be duplicated.

It is felt that both the wire and the ribbon heating

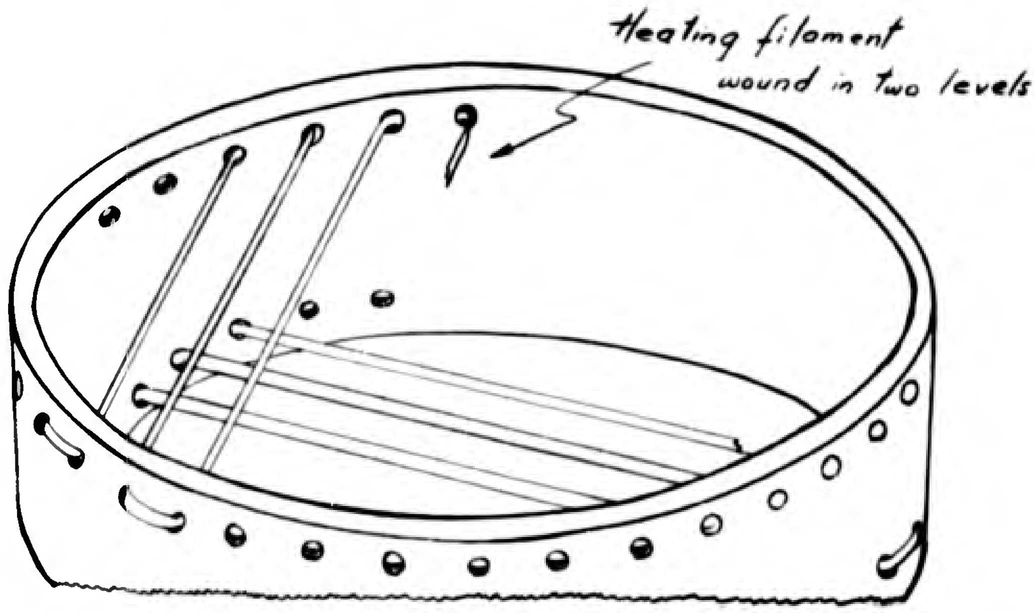


Figure 3. Wire Heating Element

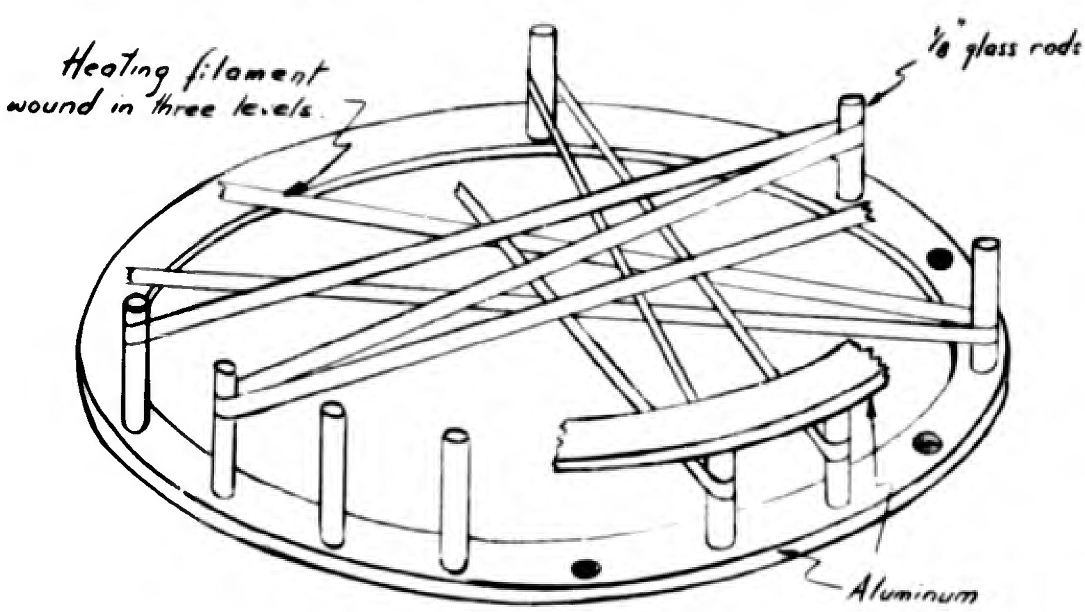


Figure 4. Ribbon Heating Element

elements could not be wound densely enough to obtain the heat transfer per unit mass of air necessary for the instability to occur. The problem was a mechanical one in that the wire and the ribbon even when tightly wound would expand when heated and move laterally 1/8 inch or more. If the element were made with the wire too close together, the wire would touch and melt. In an attempt to get more heat transfer the wire was laced in layers. The elements then were 1/2 or 3/4 inch in thickness. Possibly these elements did not closely enough approximate a plane for the instability to occur.

The element finally used was made from chromel wire gauze squares. The screen was 16 mesh, made from 24 gage wire, and is listed in the W. H. Curtin and Company Catalog. This element proved to be suitable over a wide range of air velocities and heater powers. The details of the heating element are illustrated in figure 5. The frame was made of steel, but a stainless steel frame would be more satisfactory. The composition material used to separate the ends of the rings was the plug-in pin support from a Universal Automatic coffee pot. The double mesh portion of the screen is not absolutely necessary but contributes to more uniform screen temperatures and a longer screen life. The function of the double mesh is to decrease the difference in the resistance between the center and the edge of the screen. The frame of the element was insulated from the wall of the tubing by a four-inch wide strip of sheet asbestos. As the resistance of the screen was very low, a very high current source was

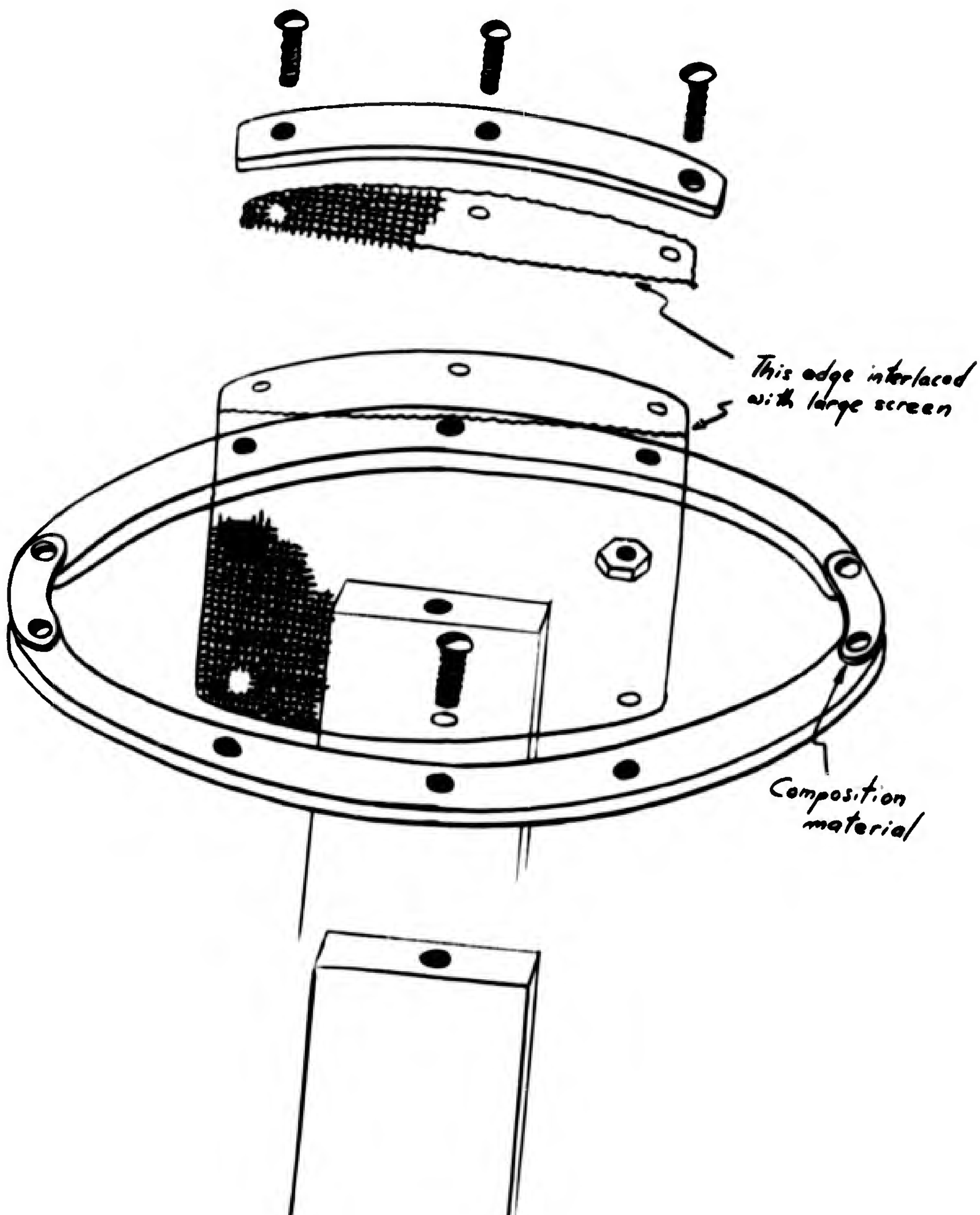


Figure 5. Screen Heating Element

needed. An AC-1000 Lincoln arc welder in which the current can be continuously adjusted from 125 to 1000 amperes was available. To prevent overheating from the high current flow, the lead-in wires were made from A. W. G. 000 cable. The conductors from the wire to the element were made from 1/4 by one inch copper bus bar material. The bus bars also supported the heating element. All experiments were conducted with the element 12 inches from the tube inlet. A transformer with a ratio of 80 to 1 was used with a 0-5 ampere precision ammeter to determine current flow. The voltage drop across the heater was measured with a 0-10 volt capacity precision voltmeter.

The apparatus for air flow measurement consisted of two plenum chambers, a set of calibrated nozzles, and differential pressure gages. The plenum chambers were required because the nozzles were calibrated in terms of static pressure drop across the nozzle. The nozzle screwed into an adapter ring between the plenum chambers. One of two differential pressure gages was used. For low mass flow rates, an inclined manometer with a capacity of 0 to 0.5 inches of water was used. For higher mass flow rates, an inclined manometer with a capacity of 0 to 3 inches of water was used.

The operation of the flow measuring system is as follows. For each nozzle a chart was available listing mass flow rate as a function of static pressure difference across the nozzle referred to 60°F and 29.92 inches of mercury. A correction factor, $F = (T_2 P_2' / T_2' P_2)^{\frac{1}{2}}$, was used to correct for the existing conditions where

$T_2 = 520°R$

$T_2' =$ temperature in the plenum chamber

$P_2 =$ corrected dry air pressure - DP

$P_2' = 29.92 - DP$

DP = differential pressure between the plenum chamber which connected to the sound tube and the atmosphere

corrected dry air pressure = barometric pressure - barometric pressure correction for temperature-vapor pressure.

DP was taken to be zero as the highest value attained was of the order of 10^{-4} inches of mercury, and the other measurements were not of sufficient accuracy to justify its use. The mass flow rate was obtained by multiplying the mass flow rate read from the chart by the correction factor calculated.

The air supply available for most of the experimental work was a fan with provision for two-speed operation. A baffle was positioned in the plenum chamber in front of the fan to reduce the turbulence in the plenum chamber and to change the kinetic energy of the air to static pressure. The pressure was varied simply by moving the fan toward or away from the entrance to the plenum chamber. The maximum static pressure attainable with the fan was 0.4 inches of water. In the final stages of the experimental work a variable-speed centrifugal blower became available. The speed control for the blower provided step-type control. In order to allow precise control of air flow, the speed control was placed at its highest setting and a variac was used to vary the voltage input. With the variac the blower could be precisely controlled to give a differential pressure of 0 to 3 inches of water across the nozzle.

CHAPTER III

EXPERIMENTAL PROCEDURE

Before each run the inclined manometers were both leveled and adjusted to read zero. The thermocouples were checked to ascertain that the readings were uniform. The variation was less than 0.04 millivolt for all runs. The microphone for the sound intensity meter was placed in the same location each time, about four inches from the end of the tube.

Because of the exploratory nature of this experiment the different test configurations will be discussed under separate headings.

The Effect of Wall Temperature on Sound Intensity

In this test the tube was horizontal and insulated. The heating element power was held constant at 361 watts, and the air velocity was held constant at 22 cm/sec. In this study velocities are expressed in cm/sec to facilitate comparison with the literature, most of which use metric units. An initial temperature profile along the tube was taken, and all temperatures and the sound intensity were recorded at ten-minute intervals.

Minimum Air Velocity As A Function of Power

In this test the tube was horizontal and was not insulated. A spray bar was installed to direct high pressure air on the tube for the purpose of holding the tube wall temperature constant. The variation in tube wall temperature with time was held to within 30°F. Because of the low air velocities, heater power had to be low to prevent overheating of the heating element. A small arc welder with a maximum capacity of 180 amperes was used as the power source. With a given heater power the air velocity was reduced until the heater stopped sounding. The air velocity was slowly increased until the tube sounded, then decreased in increments of 0.005 inches of water until the tube stopped sounding. The lowest differential pressure at which the tube sounded was recorded. The procedure was repeated for five different power settings.

Maximum Air Velocity As A Function of Heater Power

In this test the tube was horizontal, not insulated, and cooled with high pressure air. The higher capacity manometer was used. With a given power setting the air flow was increased in increments of 0.01 inches of water until the tube stopped sounding. The last reading at which the tube sustained sound was recorded. This was actually a trial-and-error procedure, and the flow was adjusted up and down to be sure of the data point. Seven data points were taken at power settings from 690 to 2342 watts.

Sound Intensity As A Function of Air Velocity

In this test the tube was horizontal, not insulated, and cooled with high pressure air. The initial plan was to take six data points at each of two different power settings, but the data appeared inconclusive so a run was made at a power setting of 1510 watts from near minimum air flow to maximum air flow, that is until the tube stopped sounding. Thirty-one data points were recorded.

Sound Intensity As A Function of Heater Power

In this test the tube was horizontal, not insulated, and cooled with high pressure air. The air velocity was held constant and the sound intensity was recorded for power settings from 406 to 1300 watts.

Tube Vertical

An adapter was installed to allow flow measurements to be made with the tube in a vertical position. All of the runs with the tube vertical were made with the insulation installed. Two methods were used to control the air flow rate. In the first the fan was used with small nozzles. With this method the flow could be closely controlled and measured. In the second method, the flow rate was determined by convection alone. Large nozzles were used in order not to unduly restrict the flow. The very low differential pressures which resulted were below the calibration curves for the nozzles, so the results were only relative.

An attempt was made to determine whether the tube would sound with the tube wall temperature higher than the inside air temperature. It was determined experimentally that the tube would heat more quickly with low air velocities as the heat transfer to the air per unit mass is higher and the air temperature is much higher. Air temperatures as high as 1000°F were obtained while heating the tube. With a low air flow so that the tube barely sounded and 734 watts heater power the tube was heated until the tube wall at thermocouple number four was approximately 500°F. The heater power was increased to 1720 watts and the air flow was allowed to increase to near the maximum velocity. This proved to produce the lowest air temperature in the center of the tube. The tube continued to sound with the tube wall temperature at thermocouple number four slowly decreasing. The power and velocity were reduced; the tube was heated until a quasi-steady state was reached, and all thermocouple readings were recorded. The power was increased to 1315 watts, and the velocity was allowed to increase to near maximum. The tube continued to sound, but the temperature of the tube wall at thermocouple number four decreased about four °F in 15 minutes of running. Thermocouple number four was monitored because it was not possible to monitor all of the thermocouples, and number four was the hottest thermocouple which was not adjacent to the heating element.

CHAPTER IV

EXPERIMENTAL OBSERVATIONS AND RESULTS

The effect of wall temperature on sound intensity was checked first. The numerical results of the test run were included in Table 1. The tube wall temperature changed from a uniform temperature of 78°F to the temperature profile illustrated in figure 6. The temperature of the hottest place on the tube was about 370°F. No significant change in sound intensity was observed. It was observed that the air temperature readings taken from both thermocouples 10 and 11 fluctuated. The range of fluctuations was from about 0.1 to 1.0 millivolts. These fluctuations seemed to be coupled with fluctuations in the air flow rate. Also worth noting was that the air temperature measured at thermocouple 11 reached a maximum of about 625°F with the inlet air temperature at 74°F. This was a much larger increase in air temperature than any previously reported in the literature.

These results with the tube horizontal seemed to conflict with the exploratory experiments which were conducted with the tube vertical. An adapter was made and the tube was installed in a vertical position. It was observed that as the tube wall heated the convection flow increased and the

TABLE 1

SOUND INTENSITY VS. TUBE WALL TEMPERATURE WITH TUBE HEATING

Thermocouple Number	Temperature (°F) 0 min	10 min	20 min	30 min	40 min	50 min	60 min	70 min
1	78	121	163	202	225	244	258	281
2	78	148	195	235	259	279	294	318
3	78	188	240	277	303	304	342	368
4	78	175	225	267	294	317	335	364
5	78	171	221	262	290	314	331	361
6	78	157	205	245	273	299	317	342
7	78	142	184	222	249	275	292	319
8	78	127	166	202	227	251	269	295
9	78	122	158	190	214	238	252	279
10	78	U - 151	U - 178	U - 190	U - 170	U - 214	U - 218	U - 238
11	78	U - 490	U - 532	U - 540	U - 568	U - 588	U - 605	U - 625
Sound Intensity (DB)	91.2	91.2	91.2	91.2	91.2	91.2	91.1	91.1

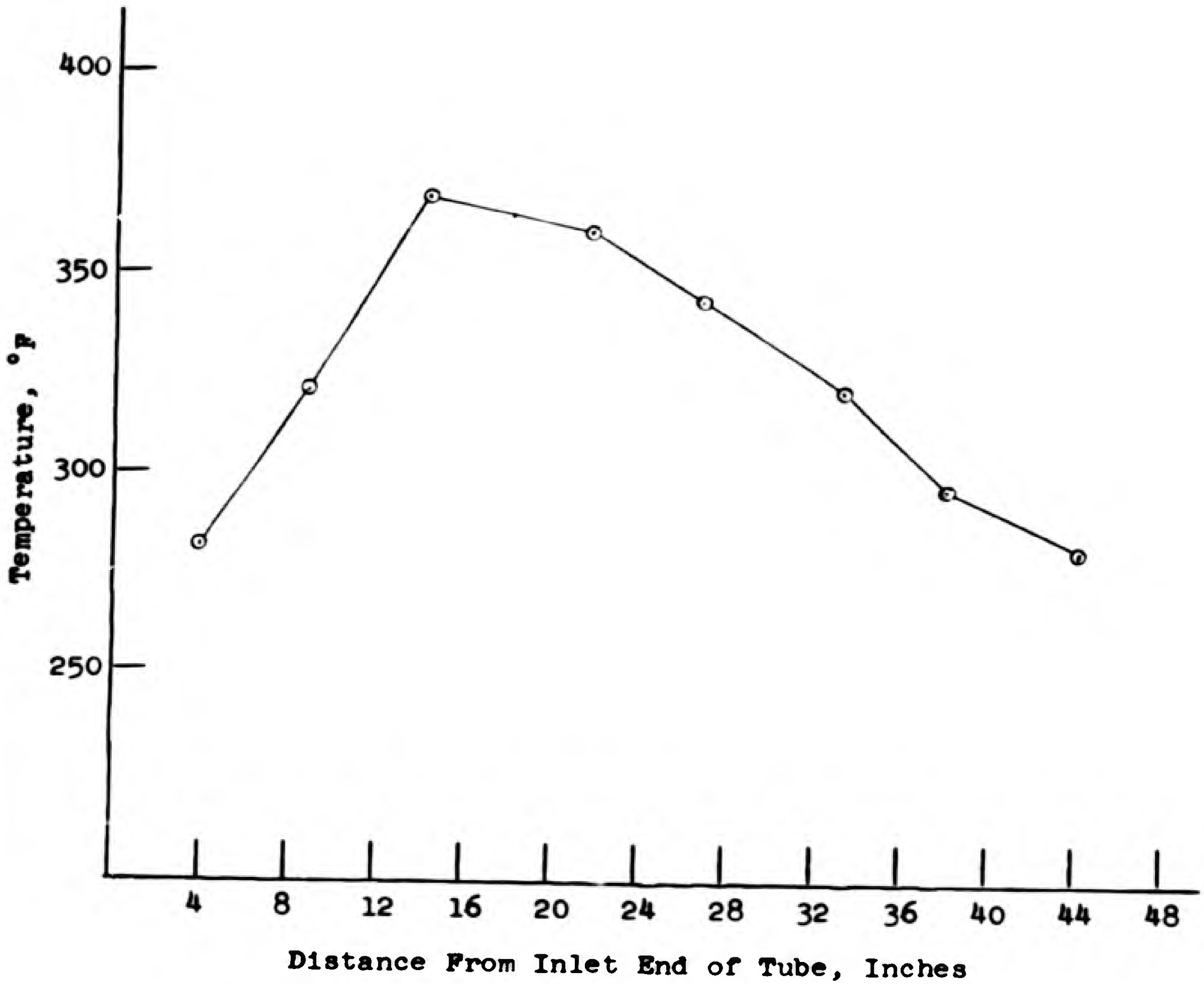


Figure 6. Typical Tube Temperature Profile

sound intensity increased. When the power and flow were held constant, no significant change in sound intensity was observed as the tube wall temperature changed. In several test runs the tube wall was heated and the power and flow were adjusted so that the tube continued to sound with the tube wall cooling, indicating that the air temperature next to the wall was lower than the wall temperature. In the most extreme instance the wall temperature at thermocouple four was 756°F, and the air temperature in the center of the tube adjacent to thermocouple number four was fluctuating about 920°F. The tube wall sounded for fifteen minutes with the tube wall temperature constantly decreasing. The total decrease was about four degrees. The previously noted fluctuations of air temperature were observed on all test runs, including the runs where the air flow was from convection alone. It was concluded that the fluctuations were inherent to the heat-driven oscillations and were not caused by the air source.

The strong dependence of sound intensity on air velocity and heater power became apparent in the first runs. A plot of sound intensity as a function of air velocity is included in figure 7. The graph indicates that near the minimum velocity, sound intensity increases rapidly with velocity, but quickly reaches a maximum sound intensity which is relatively constant over a wide range of air velocities. As the maximum air velocity is neared, the sound intensity decreases slightly. A further increase in air velocity causes the sound to abruptly cease.

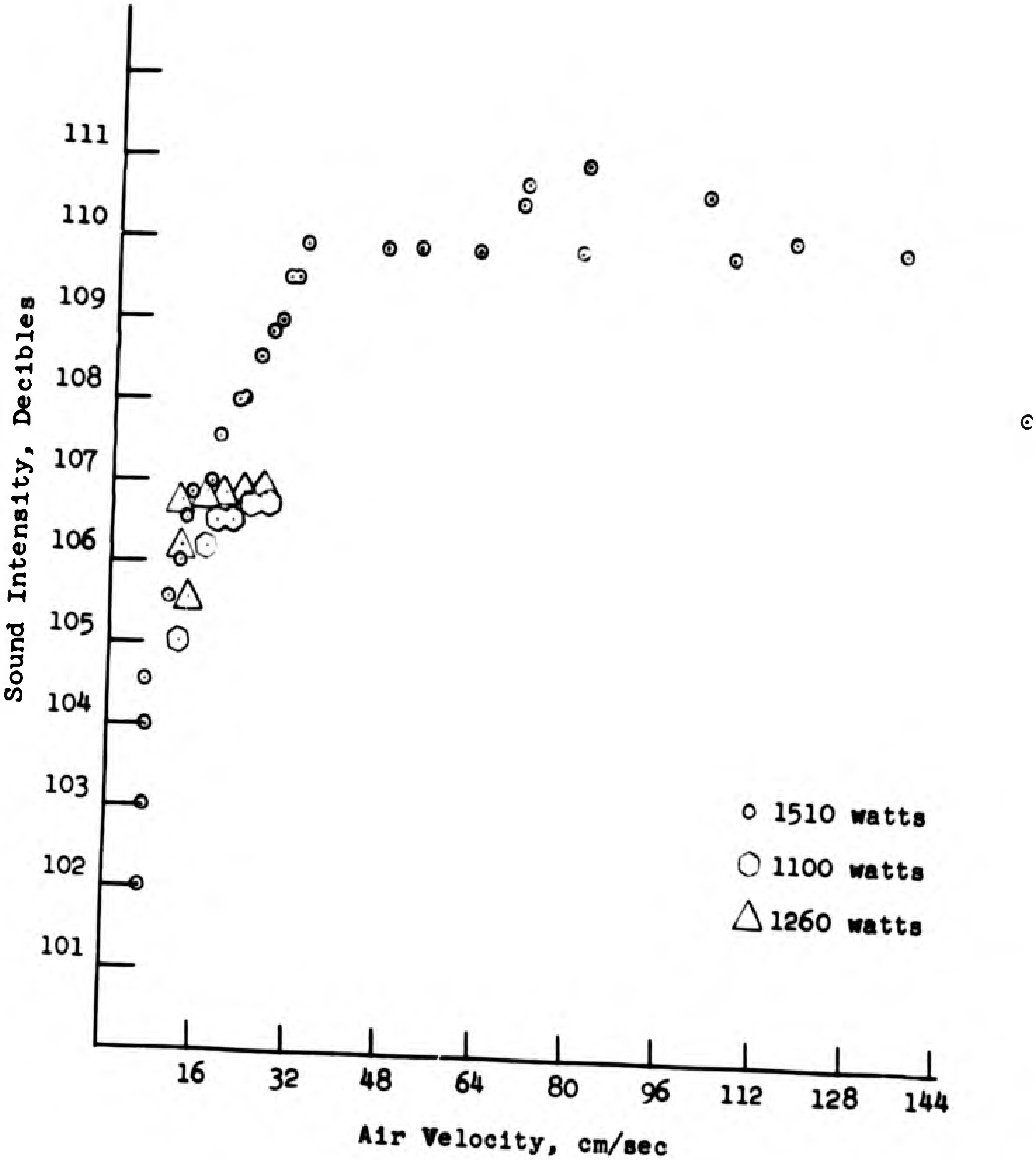


Figure 7. Sound Intensity As A Function of Air Velocity

Maling(3) reported a dependence of maximum air velocity on heater power. A plot of these results in this study was included in figure 8. From the graph a measured maximum air velocity of about 140 cm/sec compares with 91 cm/sec reported by Maling for a heater power of 1290 watts.

Sound intensity is plotted as a function of heater power in figure 9. From the graph it can be seen that sound intensity is approximately a linear function of heater power over the range tested.

The minimum air velocity as a function of power is not reported quantitatively as all except one of the differential pressure measurements were below the calibration curve for the smallest nozzle. Qualitatively the indications were that the minimum velocity, at least for the five values checked, decreased as heater power was increased. For a heater power of 423 watts the minimum air velocity was about eight cm/sec. As the heater power was increased the minimum air velocity decreased to less than five cm/sec. Maling(3) reported a minimum air velocity of 40 cm/sec.

It is felt that the maximum and minimum air velocities at which sound can be attained are strongly influenced by the type of heating element used. The analyses reported in the literature assume a ribbon or a coiled wire heating element. A heating element made from screen much more closely approximates a plane and is probably more efficient in transferring heat to the air. In this study a new screen seemed to be more efficient than a screen which had been used and was

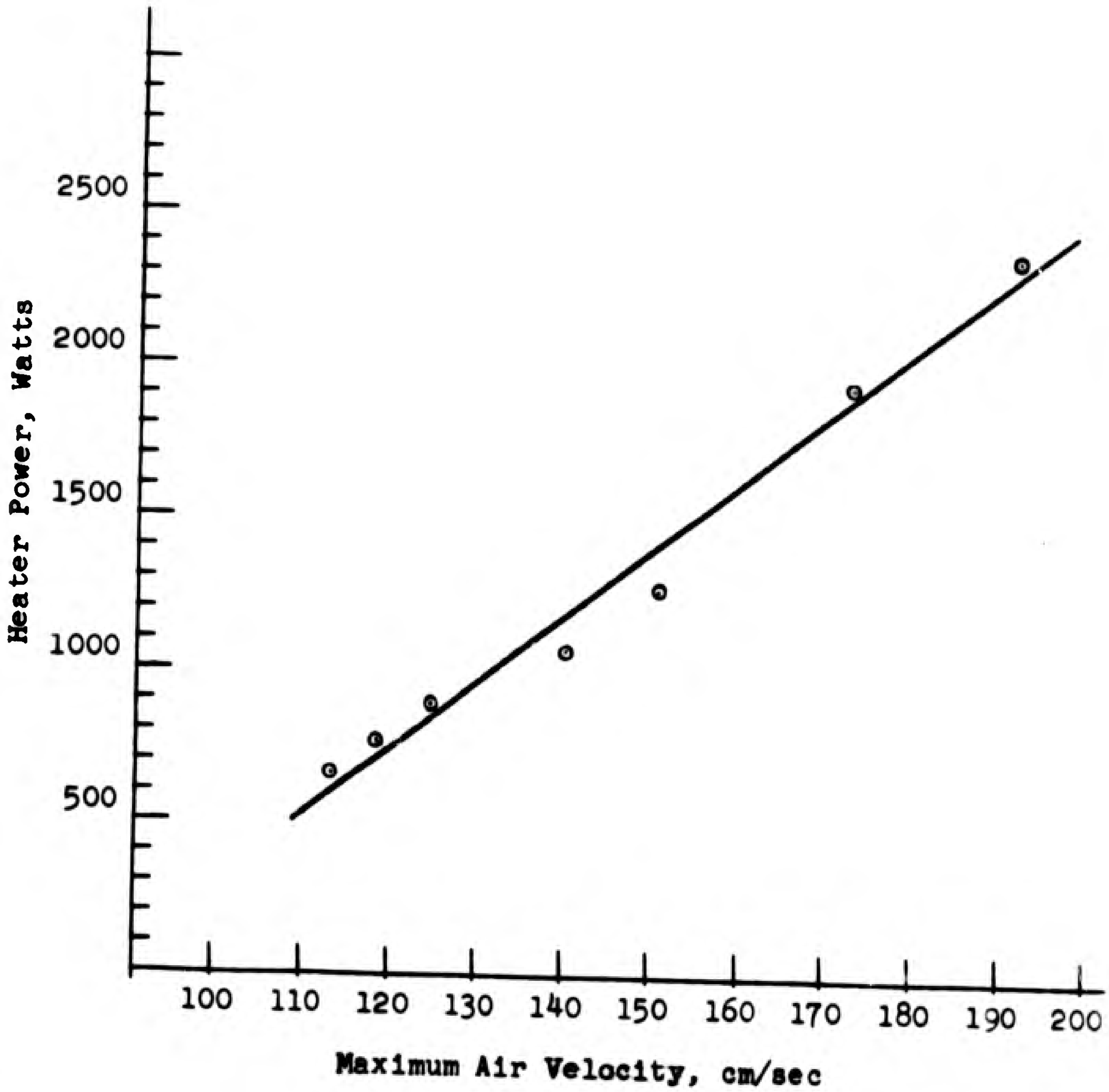


Figure 8. Maximum Air Velocity As A Function of Heater Power

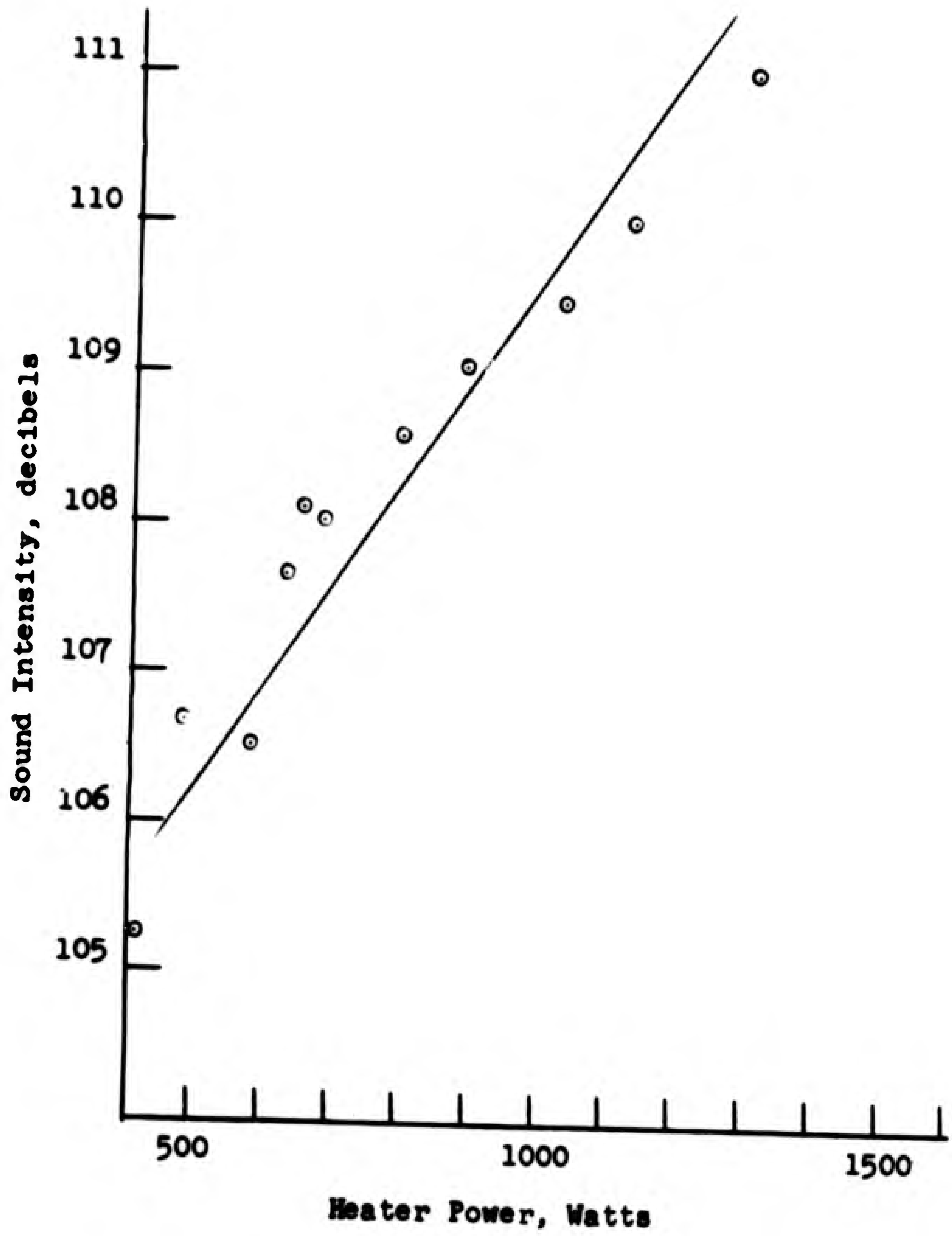


Figure 9. Sound Intensity As A Function of Heater Power

oxidized, but no studies were conducted to substantiate this.

Merk(4) calculated a minimum velocity of two cm/sec for a sample sound tube. He concluded that his velocity was too low based on a previously reported experimental value of 50 cm/sec. Using the equations developed by Merk, a minimum velocity of 2.42 cm/sec was calculated for the sound tube used in this study. The calculation is included in Appendix A. Based upon an extrapolation of the calibration curve, included in Appendix B, [illegible] minimum velocity attained in this study was between 3.33 and 3.65 cm/sec.

CHAPTER V

CONCLUSIONS

The general conclusions which can be made as a result of this study are:

1. Sound intensity is not a function of tube wall temperature.

2. Heat transfer to the tube wall is not an important parameter in the mechanism of heat-driven oscillations.

3. The heat-driven oscillations occur at very low air velocities.

4. Air velocity is an important parameter only near the minimum and maximum air velocities.

5. Maximum air velocity is approximately a linear function of heater power.

6. Sound intensity is approximately a linear function of heater power.

In the review of the literature it was noted that the equations developed in the various analyses have only limited applicability. It is felt that the salient features of the parametric relationships noted above are not discernible in any of the analyses reviewed. Indeed, it appears that the analyses were made without a knowledge of most of these important relationships.

CHAPTER VI

RECOMMENDATIONS

A better system for measuring the air velocity is needed. In particular accurate measurement of very low air velocities muut be possible.

Further study of the air temperature fluctuations which were observed could lead to a better understanding of the basic mechanism of heat-driven oscillations. A time trace of the temperatures both upstream and downstream of the heating element would be most helpful.

As the heat-driven oscillations occur at very low air velocities, it should be possible to observe and photograph the air flow at the heating element by using a pyrex tube and injecting smoke into the air stream.

As the construction of the heating element was so critical, studies with the following element configurations would be beneficial:

1. Vary the mesh size and the wire size from which the mesh was constructed.

2. Vary the area of the screen to determine the minimum screen area for heat-driven oscillations to occur.

3. Study the characteristics of a screen which is

uniformly heated over the entire cross section of the tube and which has the frame and conductors outside the tube such that the only interference to the air flow is from the screen.

4. Stack the screens so heat is not added in a plane.

BIBLIOGRAPHY

1. Carrier, G. F., "The Mechanics of The Rijke Tube," Quarterly of Applied Math, 12, 1955, p. 383.

2. Kerwin, E. M., Jr., "A Study of a Class of Thermoacoustical Oscillations," Sc. D. Thesis, E. E. Dept., MIT, 1954.

3. Maling, G. C., Jr., "Simplified Analysis of The Rijke Phenomenon," J. A. S. A., 35, 1963, p. 1058.

4. Merk, H. J., "Analysis of Heat Driven Oscillations of Gas Flow," Appl. Sci. Res., A6, 1957, pp. 317-336, 402-420.

5. Neuringer, J. L., and Hudson, G. E., "Investigations of Sound Vibrations in a Tube Containing a Heat Source," J. A. S. A., 24, November 1952, pp. 667-674.

6. Putnam, A. A., and Dennis, W. R., "A Study of Burner Oscillations of The Organ Pipe Type," J. A. S. A., 26, 1954, pp. 716-725.

7. Rayleigh, Lord, Theory of Sound, Vol. 2, Dover Publications, 1945.

8. Rijke, P. L., Phil. Mag. 17, 1859, p. 419.

9. Trilling, Leon, "On Thermally Induced Sound Fields," J. A. S. A., 27, May 1955, pp. 425-431.

APPENDIX A

MINIMUM VELOCITY CALCULATION

From Merk(4), $U_{cr} = C \frac{3.55}{8.15} a_1 \frac{d}{\ell_e}$

where $C = 0.2$

$a_1 \approx 1144$ ft/sec (speed of sound at 75°F)

$\ell_e = L + 0.613\ D$

$= [48 + 0.613(3.906)]/12$

$= 4.199$ ft

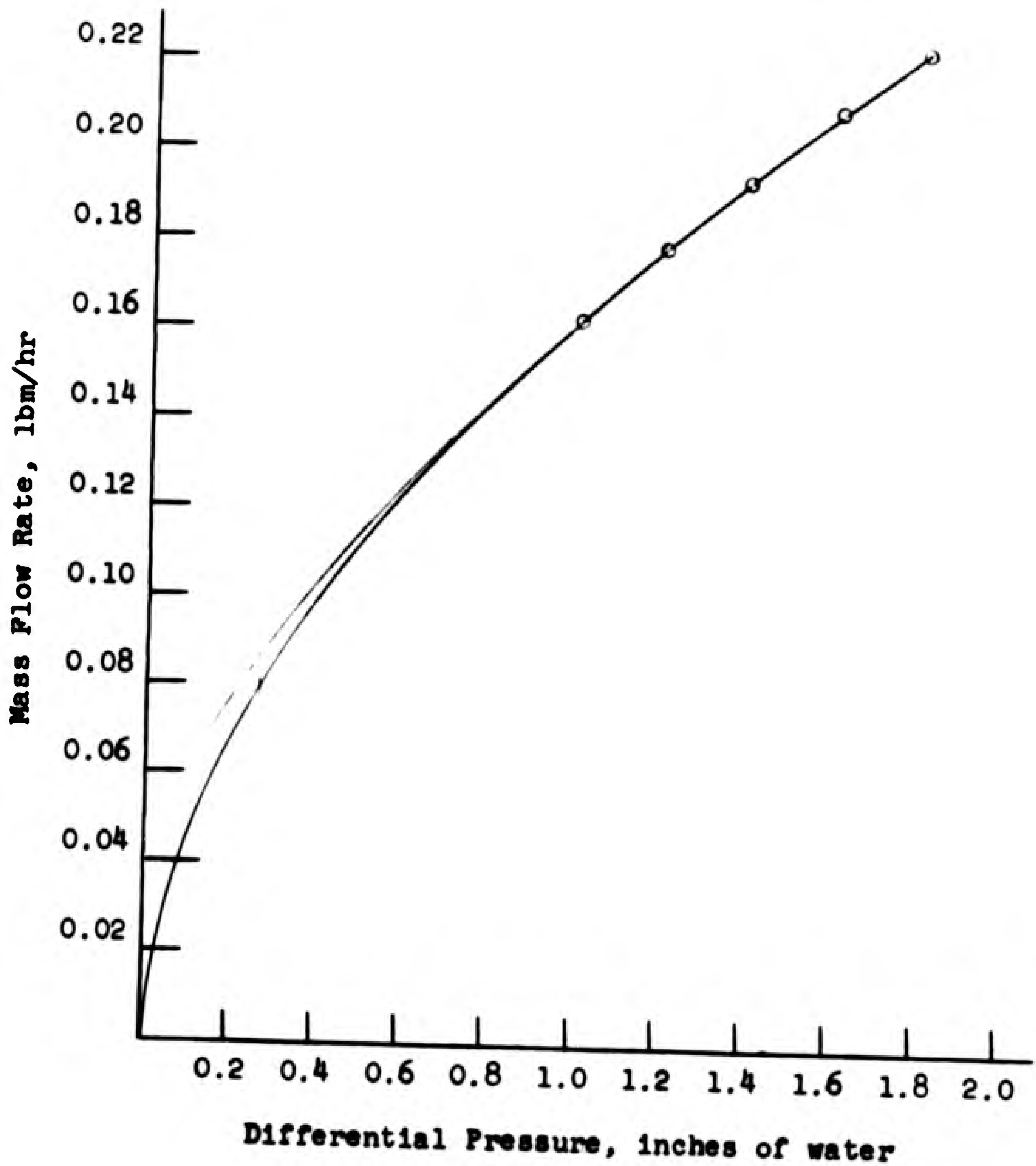
$d \sim 0.040$ inch (heating element thickness)

$$U_{cr} = \frac{(0.2)(3.55)(1144)(0.040)(2.54)}{(8.15)(4.199)}$$

$= 2.42$ cm/sec

APPENDIX B

EXTRAPOLATED CALIBRATION CURVE FOR NOZZLE NUMBER 1



APPENDIX C

A TYPICAL AIR VELOCITY CALCULATION

Barometric pressure = 29.09 inches of mercury

Air temperature = 74°F

Barometric pressure correction for temperature = 0.12 in. Hg

Wet bulb temperature = 61°F

Vapor pressure = 0.4 in. Hg

Nozzle number = 2

Differential pressure = 0.297 inches of water

Mass flow rate = 5.423 lbm air/hr

The correction factor is $F = (T_1'P_2/T_1P_2')$

$T_1' = 520°R$

$P_2 = 29.09 - 0.4 - 0.12 = 28.57$ in. Hg

$T_1 = 534°R$

$P_2' = 29.92$ in. Hg

$$F = \frac{(520)(28.57)}{(534)(29.92)}$$

$F = 0.964$

The corrected mass flow rate is $G = (0.964)(5.423)$ lbm/hr

$G = 5.22$ lbm/hr

Substituting the perfect gas law, $P = \rho RT$, into $G = \rho VA$ and solving for V,

$V = GRT/PA$

where $P = 14.7\ lbf/in^2$

V = air velocity in ft/sec

$R = 53.34\ lbf/lbm\text{-}{}^\circ R$

T = plenum chamber air temperature

$A = 11.99\ in^2$ (tube cross-section area)

$$V = \frac{(5.22)(53.34)(534)}{(14.7)(11.99)(3600)}\ \text{ft/sec}$$

$V = 0.234$ ft/sec

$V = (0.234)(30.48)$ cm/sec

$V = 7.14$ cm/sec