

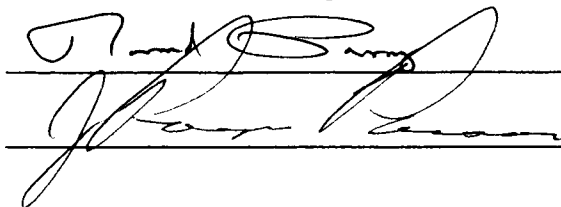
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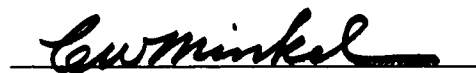


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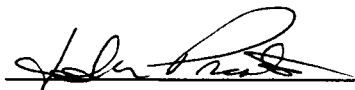
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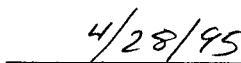
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**Acoustic Enhancement of the
Rate of Heat Transfer Over a Flat Plate**

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

John M. Preston

May 1995

Dedication

This thesis is dedicated to my grandfather, Reverend C. Walker Sessions and my great-aunt, Miss Ellinor G. Preston. Their lives have taught me about the joy of the life-long pursuit of learning.

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ABSTRACT

Increasing the rate of heat transfer can improve product quality and lower energy cost of many industrial processes. Pulsating flow produced by sound waves have been used to increase the rate of heat transfer in some situations. Infrasound, sound waves below the audible limit, has been used to increase the rate of heat transfer from small diameter, wire rods. The Aluminum Company of America, ALCOA, has proposed to use infrasound to increase the rate of heat transfer in aluminum ingot furnaces. This study examined the effects of infrasound on the rate of heat transfer from a two-dimensional region.

A standing wave of infrasound was produced in the rectangular neck of a Helmholtz resonator by two loudspeakers. One side of the neck was formed by a copper plate maintained at a constant temperature. Thermocouples imbedded in the back of the copper plate were used to measure the plate temperature. Heat flux gages mounted on the inside surface of the copper plate measured the rate of heat transfer from the plate to the air provided by a blower connected to the neck of the Helmholtz resonator.

The rate of heat transfer from a flat plate was increased by infrasound. Infrasound produced the largest increase in the rate of heat transfer when compared to natural convection. Infrasonic effects decreased with increasing air velocity when compared to forced convection. The effect of infrasound on the rate of heat transfer was shown to be a function of the air velocity inside the neck, the hydraulic diameter and the sound pressure level inside the neck. Infrasonic enhancement of the rate of

heat transfer inside aluminum ingot furnaces does not appear practical at this time since the improvement is negligible at high fluid velocities.

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CHAPTER 1

INTRODUCTION AND LITERATURE REVIEW

Introduction and Background

The manufacture of many aluminum products involves the rolling of aluminum ingots at elevated temperatures. The ingots are heated in indirect gas-fired, forced convection furnaces and approximately twenty four hours are required to reach rolling temperatures. To increase production rates and/or reduce energy costs, various ideas have been proposed to shorten the heating time. One such idea is the use of low frequency sound waves, infrasound, to increase the rate of heat transfer from the heated air to the ingots.

Statement of Intent

This study investigated the effects of infrasound on the rate of heat transfer from a flat plate to an airstream flowing between the plate of interest and another parallel plate. The objectives of this study were the determination of the local heat transfer coefficient and Nusselt number as a function of: (1) distance from the infrasonic source, parallel to the direction of propagation of the sound wave; (2) distance from the center line of the plate, perpendicular to the direction of propagation of the sound wave; (3) plate width, and (4) airflow velocity. Also examined were the average heat transfer coefficient and the average Nusselt number as a function of: (1) plate width, (2) airflow velocity, and (3) frequency and amplitude of infrasound.

Use of Sound Waves in Heat Transfer

In many heating and cooling applications, a high rate of heat transfer is beneficial in minimizing energy costs and/or increasing the rate of production. Typically, the rate of heat transfer between a fluid and a solid surface is increased by using high fluid velocity. Using increasingly higher fluid velocities to increase the rate of heat transfer means each particle of fluid is in proximity to the surface or object of concern for a decreasingly shorter length of time. Although the overall heat transfer between the fluid and the solid surface is increased by higher fluid velocities, the heat absorbed by the fluid particles, individually, is lowered. If the heat transfer fluid is subjected to a pulsating flow, each fluid particle can remain in proximity with the surface or object for a longer period of time, reducing the overall power required to propel the fluid. Additionally, the pulsating flow disrupts the boundary layer, promoting turbulent flow and increasing the time in which a defined volume of air remains in proximity with the surface or object.

A flow pulse can be created by a small pressure disturbance in a fluid. The pressure disturbance will propagate at the speed of sound in the fluid. As the disturbance is transmitted, the regions of compression and rarefactions move through the fluid. This is referred to as a traveling sound wave. The traveling sound wave imparts an instantaneous velocity, U , and pressure, P , to each fluid particle. The instantaneous velocity and pressure change of the fluid particles are normally described by sine waves, Equation 1.1 and 1.2, respectively, if the pressure disturbances are generated by a harmonic oscillator.

$$U(x,t) = -A \omega \cos(cx - \omega t) \quad (1.1)$$

$$\Delta p(x,t) = \Delta p_{\max} \sin(cx - \omega t) \quad (1.2)$$

where c represents the speed of sound in the fluid, ω represents the circular frequency (radian per second), A represents the maximum amplitude of displacement and $\Delta p_{\max}$ is the maximum pressure change. The velocity wave is 90° out of phase with the pressure wave. Sound pressure is measured in decibels, dB, which relates the root-mean-square sound pressure to a reference sound pressure by Equation 1.3:

$$\text{Sound Pressure Level, } SPL = 20 \log_{10} \frac{P_{rms}}{P_{ref}} \quad (1.3)$$

The standard reference sound pressure level, P_{ref} , is $20 \mu\text{N/m}^2$ and P_{rms} is the root-mean-square of the peak pressure amplitude.

If a traveling sound wave is generated in a closed tube of finite length, the wave will be reflected from the closed end and propagate in the opposite direction. The original wave and the reflected wave will add vectorially. When the original and reflected wave are in phase, the maximum sound pressure and velocity will occur, which is termed resonance. A point in the tube that does not experience any pressure variations as a result of the sound wave is referred to as pressure node; a point that experiences the largest pressure variations is called a pressure antinode. In Figure 1.1, Point A represents a pressure node and Point B represents a pressure antinode. Note that velocity nodes and antinodes are similarly defined. At resonance a standing sound wave is produced and cyclic changes in the speed and direction of the fluid particles

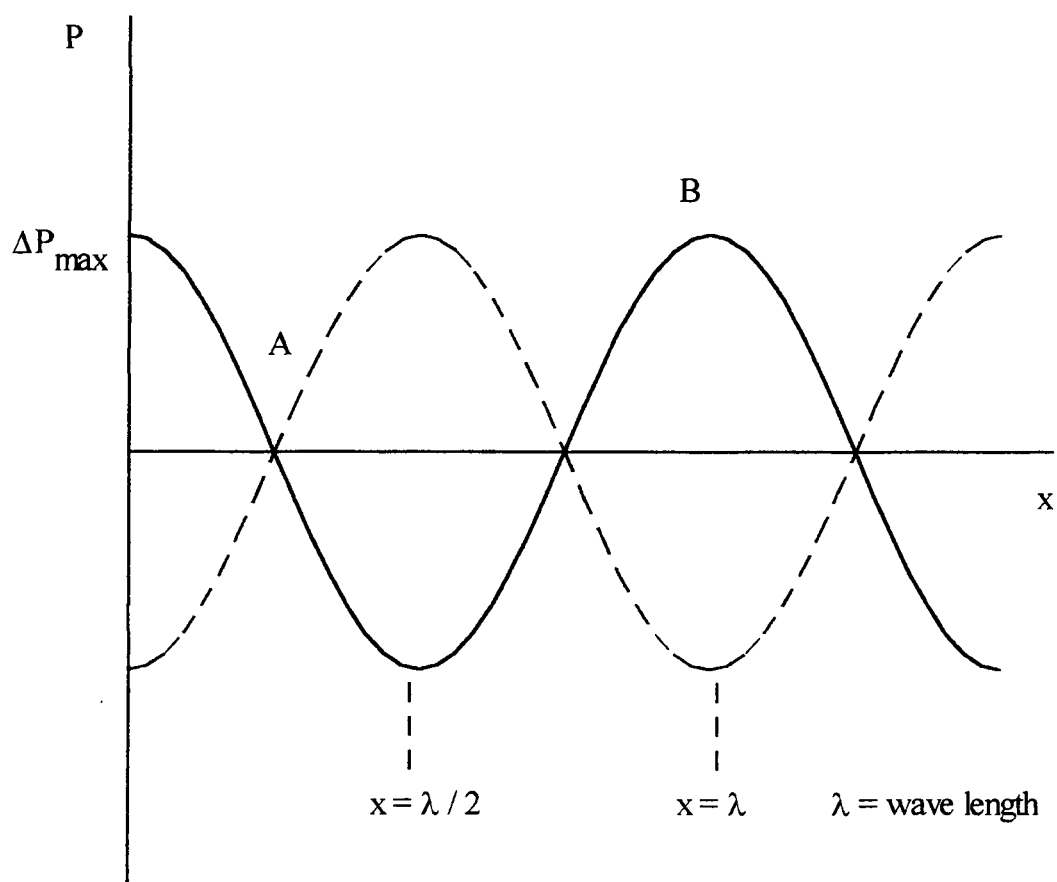


Figure 1.1: Pressure Variation of Imposed Sound Field.

Source: Dunn, J. R., *The Influence of Resonant Acoustic Vibrations on Through-Flow Drying of Tufted Textile Materials*, M. S. Thesis, Georgia Institute of Technology, 1964.

occur; however, the location of the nodes and antinodes remain constant with respect to the tube. The resonant frequency is a fixed, natural parameter of the tube.

Applications of Sound Waves in Heat Transfer

In recent attempts to use sound waves to increase the rate of heat transfer, frequencies below 20 Hz have been used because 20 Hz is the lower limit of human hearing and therefore, these frequencies are not an irritant. Sound frequencies below 20 Hz are normally considered infrasound. Presently, the use of infrasound has been concentrated in two areas, cooling small objects and some cleaning applications. The sound waves are generated by infrasound generators, which produce energy levels of several hundred watts and a sound pressure level of 130 dB at a distance of one meter from the source. Infrasonik, Inc. manufactures a series of high power, industrial, low frequency sound generators under the trademark name of Infracone™. The Infracone consists of a patented exigator and a resonance tube, one quarter of the wavelength of the generated sound. The exigator generates infrasound by pulsating air from a compressor or blower into the resonance tube. The resonance tubes are connected end to end to produce a closed tube, one half the wavelength of the generated sound wave, with the exigators at opposite ends [Olsson, 1982]. When the exigators are activated 180° out of phase, a velocity node is formed in the middle, where the tubes are connected. The exigators generate a maximum air particle velocity of 20.4 m/s (66.9 ft/s) at the velocity node and a maximum sound pressure of 9000 Pa (1.305 psi) at the pressure node at 20 Hz. Infrasonik's research has concentrated on the effects of

infrasound in a small region and little is known about how infrasound affects the rate of heat transfer over a large region such as a flat plate.

It has been proposed by the Aluminum Company of America, ALCOA, to use infrasound to improve the rate of heat transfer in aluminum ingot furnaces. Each indirect gas-fired, forced convection furnace holds two carts with four ingots, 53.34 cm x 188 cm x 685.8 cm (21" x 74" x 270"), per cart. The ingots are stacked on top of each other with small spacer blocks, approximately 6.35 cm (2.5 in.) high, between the ingots. Each furnace contains two 70.1 m³/s (150,000 cfm) fans overhead providing air circulation. These fans produce an air velocity of 50.8 m/s (10,000 fpm) out of the sidewalls into the region between the ingots. The region between the ingots was modeled by the test equipment, as shown in Figure 1.2.

Literature Review

Infrasonic generators have been used in wire rod cooling, with the infrasonic units placed on roller conveyors [Malmgren, 1991]. The units require 2.8 meters (9.2 ft) of conveyor length with an active zone, 1.2 meters (3.9 ft) in length and produce sound waves at 20 Hz and a sound pressure level of approximately 170 dB. The distribution of heat transfer within the active zone was not described, but the highest cooling rates were assumed to occur at the center of the active zone. Inside the active zone each air particle remained in proximity with the wire rod for a much longer time than it would be with conventional air, as illustrated in Figure 1.3.

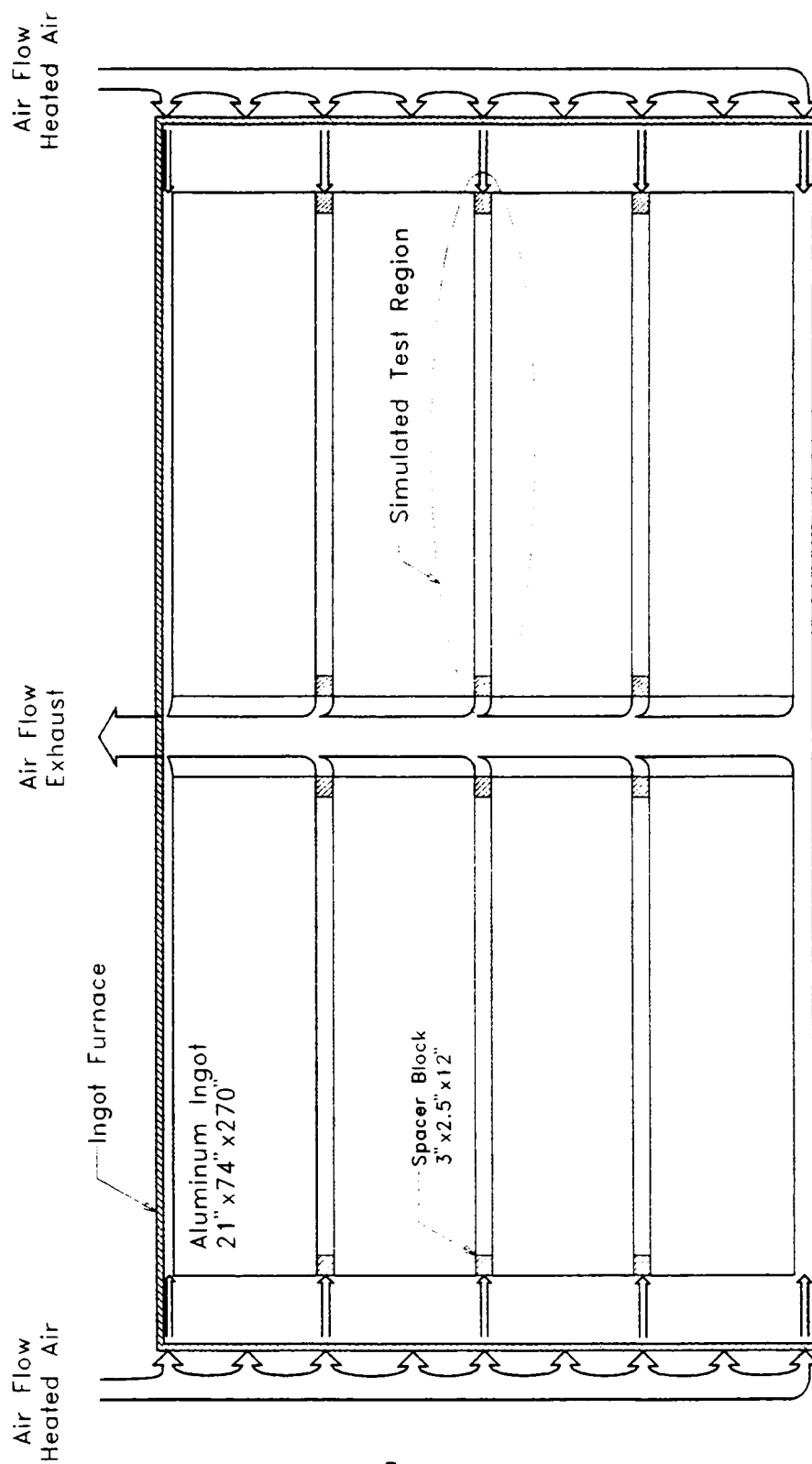


Figure 1.2: Aluminum Ingot Furnace; ALCOA, Tennessee Operations, End View.

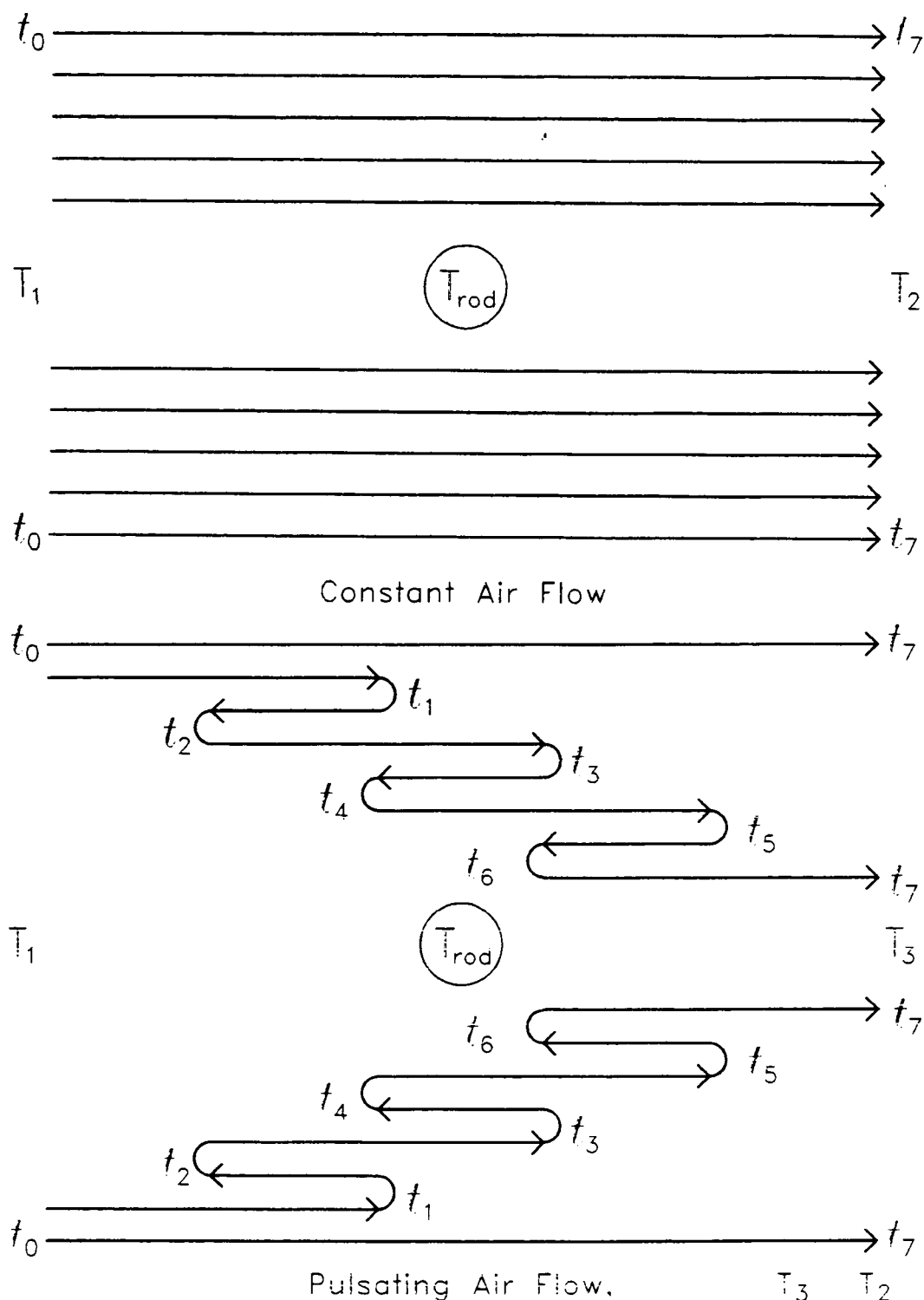


Figure 1.3: Comparison of Constant and Pulsating Air Flow.

Note: Diagram is not intended to indicate air flow in the vertical direction.

Adapted from Malmgren, N-G, "Recent developments in controlled cooling systems for wire rod - Infrasonic cooling," *Iron and Steel Engineer*, January 1991, pp. 45-48.

Infrasonik, Inc. reports an average heat transfer coefficient, $\bar{h}$, of 1700 W/m²·K (300 Btu/h·ft²·°F) and average Nusselt number, $\overline{Nu}$, of 480 for the cooling of wire rods, diameter ≤ 10 mm, using infrasonic generators with the same frequency and sound pressure level as above [Lasday, 1989]. This can be compared with an average Nusselt number of 50 obtained with a steady air velocity of 20 m/s over a cylinder.

In a study reported by Woods, the heat transfer from a cylinder in cross flow was increased by using sound waves at 18 Hz and approximately 170 dB. The most significant increase in the rate of heat transfer occurred at steady flow Reynolds numbers less than 40,000. Compared with conventional fan systems, various combinations of infrasound and steady air flow reduced the power requirements by 12% to 75%, depending on the required rate of heat transfer [Woods, 1992].

The effect of pulsating flow on heat transfer from a flat plate was investigated by Bayley, Edwards, and Singh. The pulsating flow was produced by a butterfly valve connected to a variable speed D.C. motor downstream of a centrifugal blower. The frequency of the pulsations was controlled by varying the speed of the D.C. motor, and the amplitude of the pulsations was governed by the diameter of the butterfly valve. The amplitude of pulsation was reported by using a non-dimensional pressure parameter, P , defined as half of the pressure amplitude divided by the dynamic head based on the mean air velocity. For critical values of this parameter, between 12 and 19, the Nusselt number was shown to increase proportionally to the non-dimensional pressure raised to the 0.23 power over the flow rates examined [Bayley et al., 1961].

The drying rate of tufted carpet samples was shown to increase linearly as a function of the ratio of sound Reynolds number to the Reynolds number based on the volumetric flow rate in the test section. The sound Reynolds number was based on the maximum velocity of air particles in a resonant acoustic field. The drying rate increased a maximum of 22 per cent at a Reynolds number ratio of 0.325 for flow rates between 2.04 m/s (6.70 ft/sec) and 5.24 m/s (17.18 ft/sec), a sound frequency of 300 Hz and sound pressure level of approximately 140 dB [Dunn, 1964].

CHAPTER 2

EXPERIMENTAL APPARATUS

Helmholtz Resonator

A standing wave is formed in the neck of a Helmholtz resonator by producing sound waves at the resonant frequency at or near the neck of the resonator. A Helmholtz resonator consists of a rigid-walled cavity and a neck as shown in

Figure 2.1. The resonance frequency of a Helmholtz resonator is governed by Equation 2.1 [Thompson, 1972]:

$$f = \frac{c}{2\pi} \sqrt{\frac{A}{VL'}} \quad (2.1)$$

where:

f	= resonant frequency	[Hz]
c	= speed of sound in air, 340	[m/s]
A	= cross-sectional area of the neck entrance	[m ²]
V	= volume of the resonance chamber	[m ³]
L'	= effective length of the neck	[m]

The effective length of the neck, L' , is a function of the physical length of the neck and the size and type of opening of the neck. For a flanged outer end the effective length is:

$$L' = L + 1.7a \quad (2.2)$$

where:

L	= physical length of the neck	[m]
a	= radius of neck entrance	[m] [Kinsler et al., 1982]

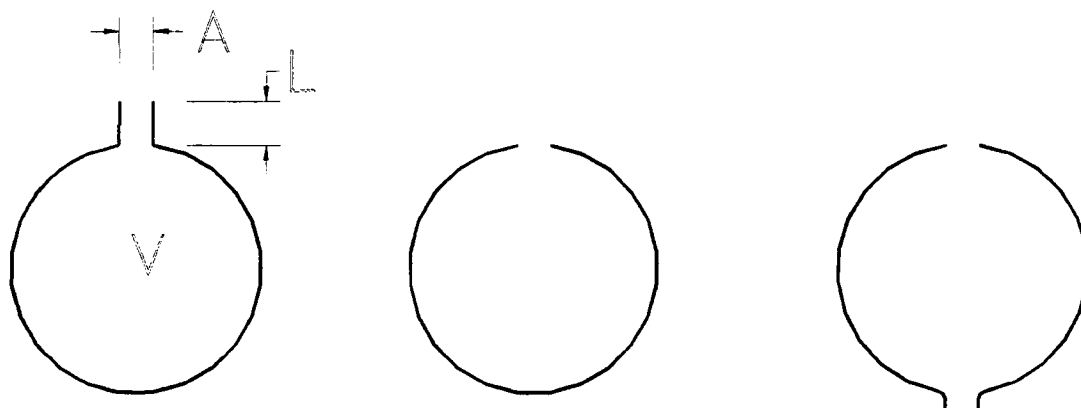


Figure 2.1 Helmholtz Resonators.

Adapted from Kinsler, L. E., Frey, A. R., Coppens, A. B., and Sanders, J. V., *Fundamentals of Acoustics*, 3rd edition, John Wiley and Sons, Inc., New York, 1982, pp. 225-228.

The neck region, where the amplitude of acoustic waves is the highest, was used to model the region between the aluminum ingots. Based on recommendations from Infrasonik, Inc. the cross sectional area of the neck was limited to 0.01 m^2 (15.5 in^2) and the physical length of the neck to 0.8 m (31.5 in) when using loudspeakers for sound generation [Sandström, 1994]. The inside dimensions of the neck, or channel were 0.0635 m (2.5 in) high and 0.8 m (31.5 in) long. For the first series of tests the inside width of the channel was 0.1575 m (6.2 in) and for the second and third series the channel width was 0.287 m (11.3 in) and 0.4369 m (17.2 in), respectively. The cross-sectional area, 0.0635 m by 0.1575 m ($2.5 \text{ in} \times 6.2 \text{ in}$), of the entrance to the channel was not varied with channel width. Using the dimensions for the channel size listed above and a

proposed resonance frequency of 20 Hz, the interior volume of the resonance chamber was calculated to be 0.0835 m^3 (5094 in^3), using Equations 2.1 and 2.2.

In an effort to increase the amplitude at resonance, speakers were attached to a box with the same interior volume as the resonance chamber. The experimental apparatus was constructed of 1.905 cm ($3/4$ in) thick plywood and laminated on the inside to form an acoustically hard surface. The corners were sealed to prevent any leakage. The experimental apparatus is shown in Figures 2.2 and 2.3. The infrasound was generated by two Blaupunkt Pro Component 15 inch diameter speakers. The speakers were driven by a Gold Star Function Generator connected to a Crown 460 CSL™ Professional Power Amplifier. A list of the equipment used is shown in Appendix A.

Constant Temperature Plate

The experiment was designed to measure the rate of heat transfer between the top of the channel of the Helmholtz resonator and the air in the channel. The top of the channel was made from 110 copper, 0.00953 m thick by 0.762 m long by 0.475 m wide ($0.375 \text{ in} \times 30 \text{ in} \times 18.7 \text{ in}$). The copper plate was maintained at a constant temperature by circulating heated water from a constant temperature bath through copper tubing soldered to the back of the copper plate, as shown in Figure 2.4.

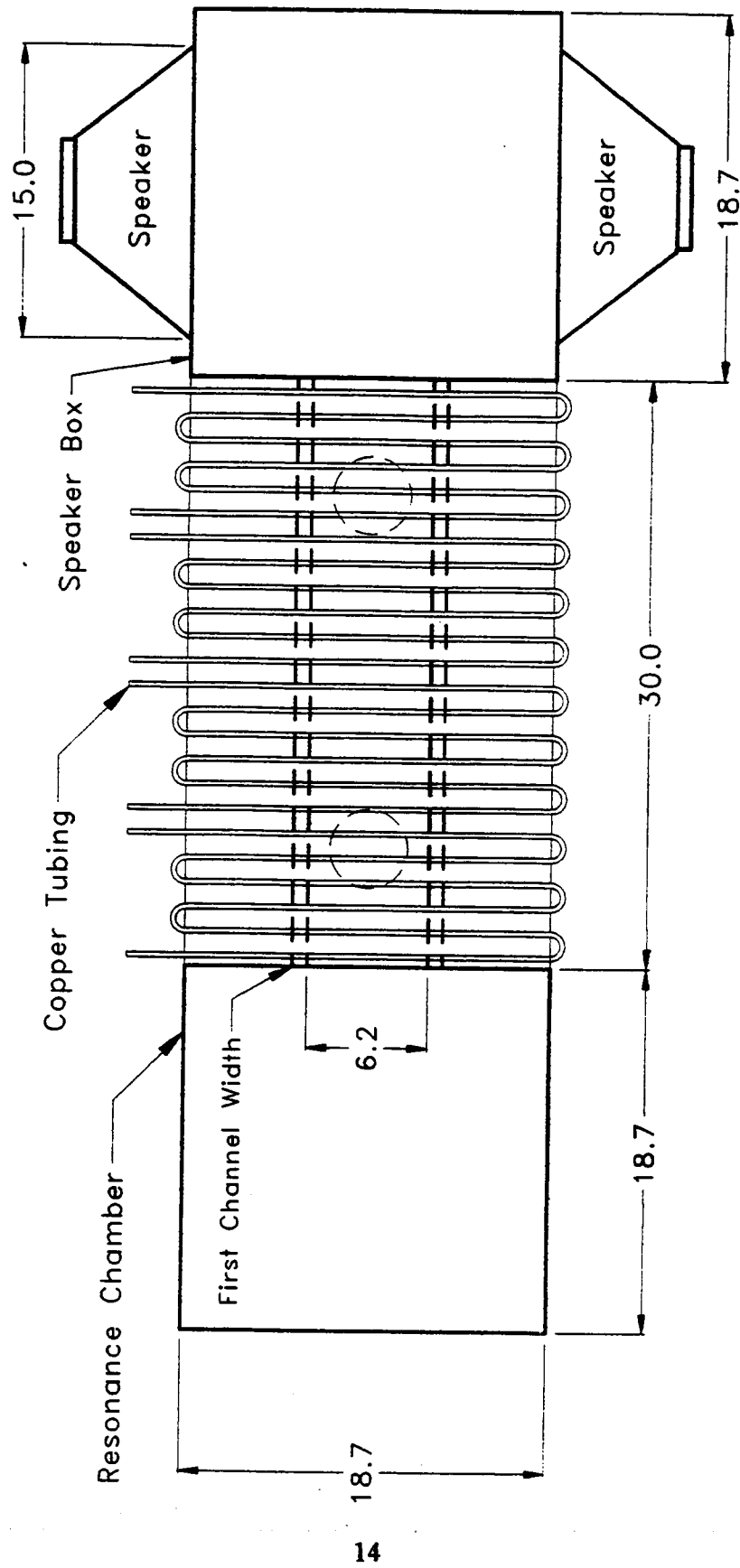


Figure 2.2: Experimental Apparatus, Top View.
Note: Dimensions in inches.

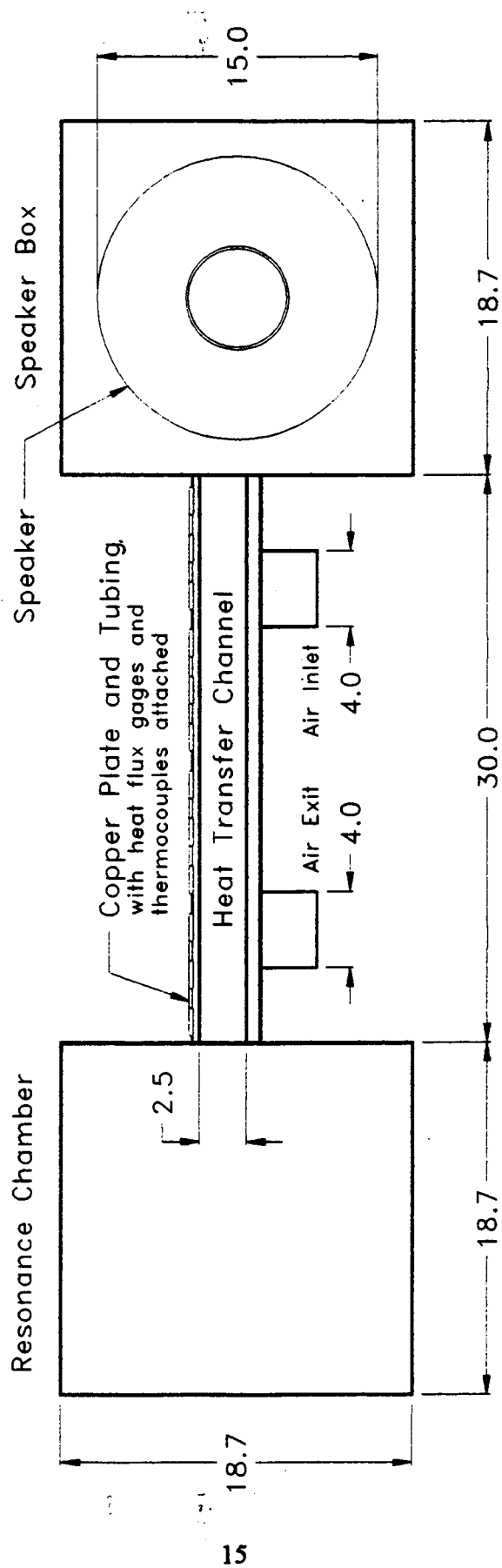


Figure 2.3: Experimental Apparatus, Front View.
 Note: Dimensions in inches.

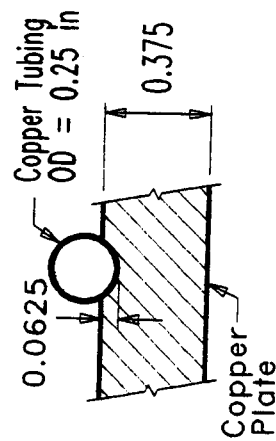


Figure 2.4: Copper Tubing Arrangement.

Note: Dimensions in inches.

Air Flow

Regardless of the temperature of the copper plate, stagnant air inside the apparatus will reach a condition approaching conduction and heat transfer will be limited without additional, fresh air. To measure a steady state rate of heat transfer an air flow was introduced to the system through a 10.16 cm (4 in) diameter hole in the bottom of the channel with the center 10.16 cm (4 in) from the inside edge of the speaker box. The air exited the system through another 10.16 cm (4 in) diameter hole in the bottom of the channel with the center 10.16 cm (4 in) from the inside edge of the resonance chamber. Air flow was provided by a $0.236 \text{ m}^3/\text{s}$ (500 cfm) blower connected to the air inlet by a flexible aluminum duct and varied by attaching a damper to the blower.

EQUIPMENT EVALUATION AND INSTRUMENTATION

The system frequency was designed to resonate at 20 Hz. The experimental system resonant frequency was determined by varying the input frequency of a sine wave input to the speakers and recording the maximum sound pressure level at each frequency. The output of the function generator was sent to the speakers through the power amplifier. The amplified signal was also sent to a Tektronix 5103W Oscilloscope, used to measure the peak voltage. The peak voltage was used to determine the input power to the speakers, using Equation 3.1.

where:

P	= input power	[W]
V _p	= peak voltage	[V]
R	= speaker resistance, 4 ohms	[Ω]

18

the channel with a frequency resolution of 0.25 Hz. Initially, a few resonance tests were conducted with the microphone placed on top of the channel; however, this distorted the results and did not provide an accurate indication of the resonant frequency. The input frequency test range was set by approximating the resonant frequency audibly as the input frequency was varied. During each resonance test the maximum sound pressure level for each frequency was recorded as the input frequency was increased, and then decreased to identify any hysteresis in the function generator. The results of repeated measurements for a particular frequency were averaged to establish a relationship between sound pressure level and input frequency. No sharp peaks were present in the maximum sound pressure level as a function of the input frequency as had been expected; however, a region of highest sound pressure level was observed for each channel width. The variability of the results may have been a function of the resolution of the FFT. Increasing the resolution of the FFT to 0.125 Hz required data analysis over an eight second time interval, which unacceptably lowered the response time of the signal analyzer. The resonant frequency of the system for the first channel width, 15.75 cm (6.2 in), was approximately 36.50 Hz, as shown in Figure 3.2. The variability of the results increased for the second channel width, 28.70 cm (11.3 in), and the resonant frequency increased to approximately

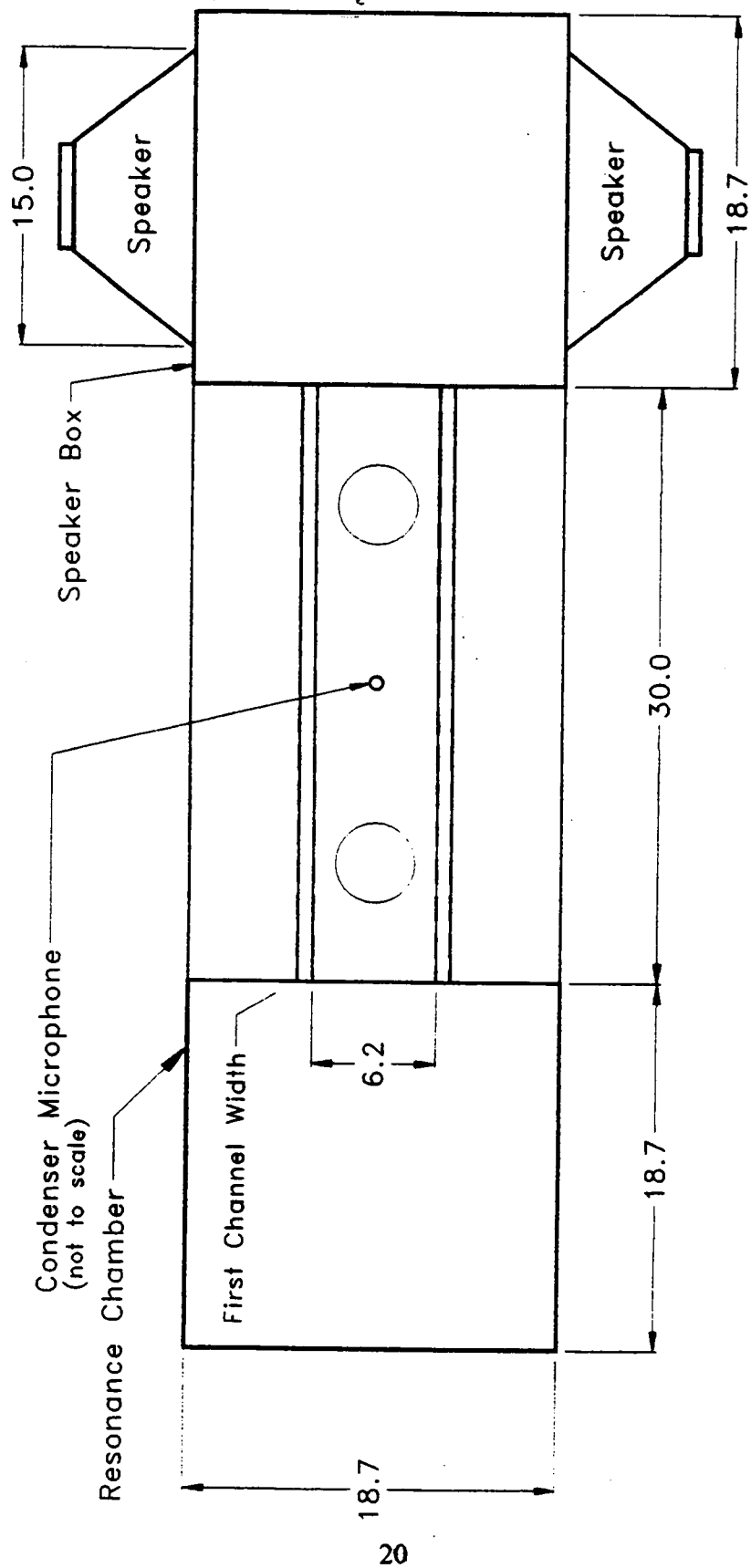
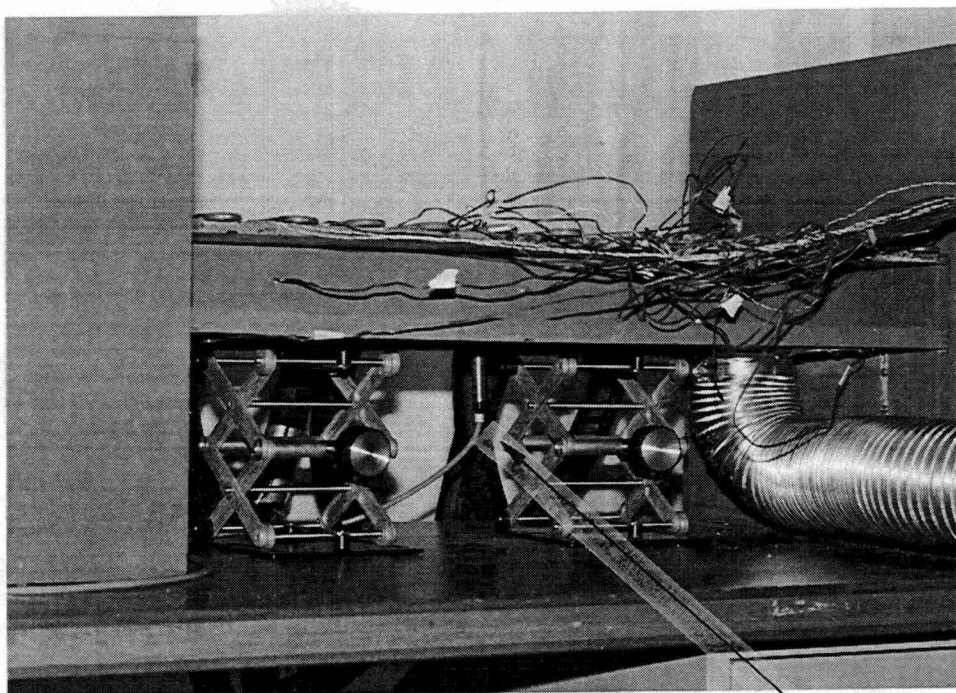
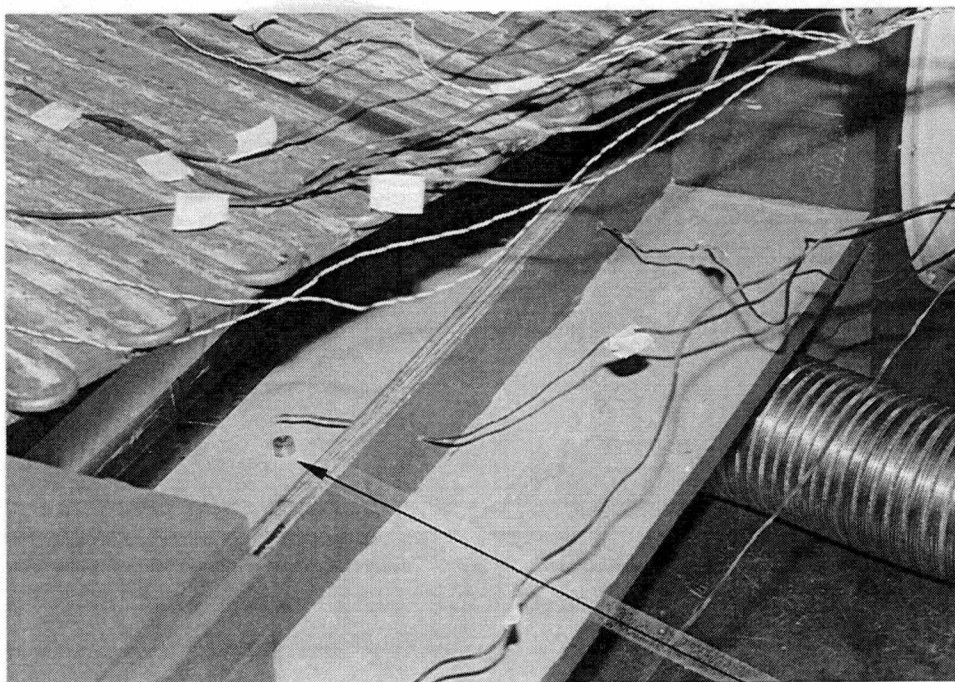


Figure 3.1: Microphone Arrangement.
Note: Dimensions in inches.



(b) Microphone Placement (outside the channel).

Microphone Support



(c) Microphone Placement (inside the channel).

Condenser Microphone

Figure 3.1: continued.

Maximum Sound Pressure Level as
a Function of Input Frequency
Channel Width, 15.75 cm (6.2 in)

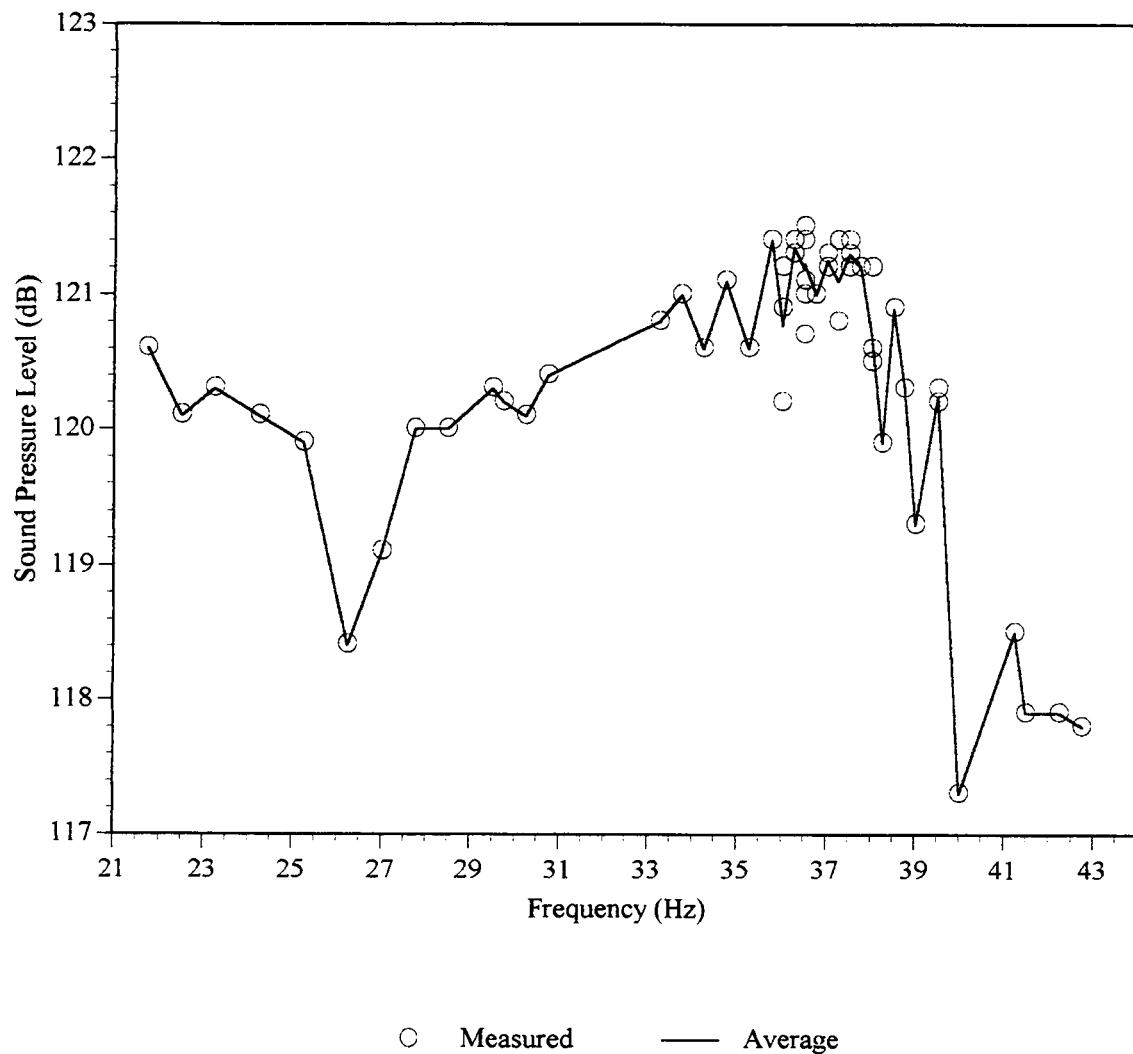


Figure 3.2: Resonance Test for the First Channel Width.

36.75 Hz, Figure 3.3. The maximum sound pressure level for the third channel width, 43.69 cm (17.2 in), was not measured at frequencies below 28 Hz because the most appear to change as the channel width was increased from 28.70 cm (11.3 in) to 43.69 cm (17.2 in). The width of the channel entrance remained constant, 15.75 cm (6.2 in), as the width of the channel was varied. Consequently, Equation 2.1 could not be used to predict a change in the resonant frequency of the system for the second and third channel widths, where the cross sectional area of the channel was larger than that of the channel entrance. The relationship between sound pressure level and frequency was not as apparent for the third channel width, Figure 3.4. The resonant frequency of the system is a function of the speed of sound in air, Equation 2.1, which is governed by Equation 3.2 [Munson et al., 1990]:

$$c = \sqrt{RTk} \quad (3.2)$$

where:

c	= speed of sound in air	[m/s]
R	= ideal gas constant, 286.9	[J/kg·K]
T	= air temperature	[K]
k	= specific heat ratio of air, 1.4	

As the temperature of the copper plate was raised, the air temperature inside the channel rose, increasing the theoretical resonant frequency of the system. Figures 3.5 and 3.6 show the sound pressure level variation with input frequency for the second channel width with copper plate maintained at 80 °C (176 °F) and 90 °C (194 °F), notable changes in the sound pressure level occurred at frequencies greater than 28 Hz for the first and second channel widths. The resonant frequency of the system did not

Maximum Sound Pressure Level as
a Function of Input Frequency
Channel Width, 28.70 cm (11.3 in)

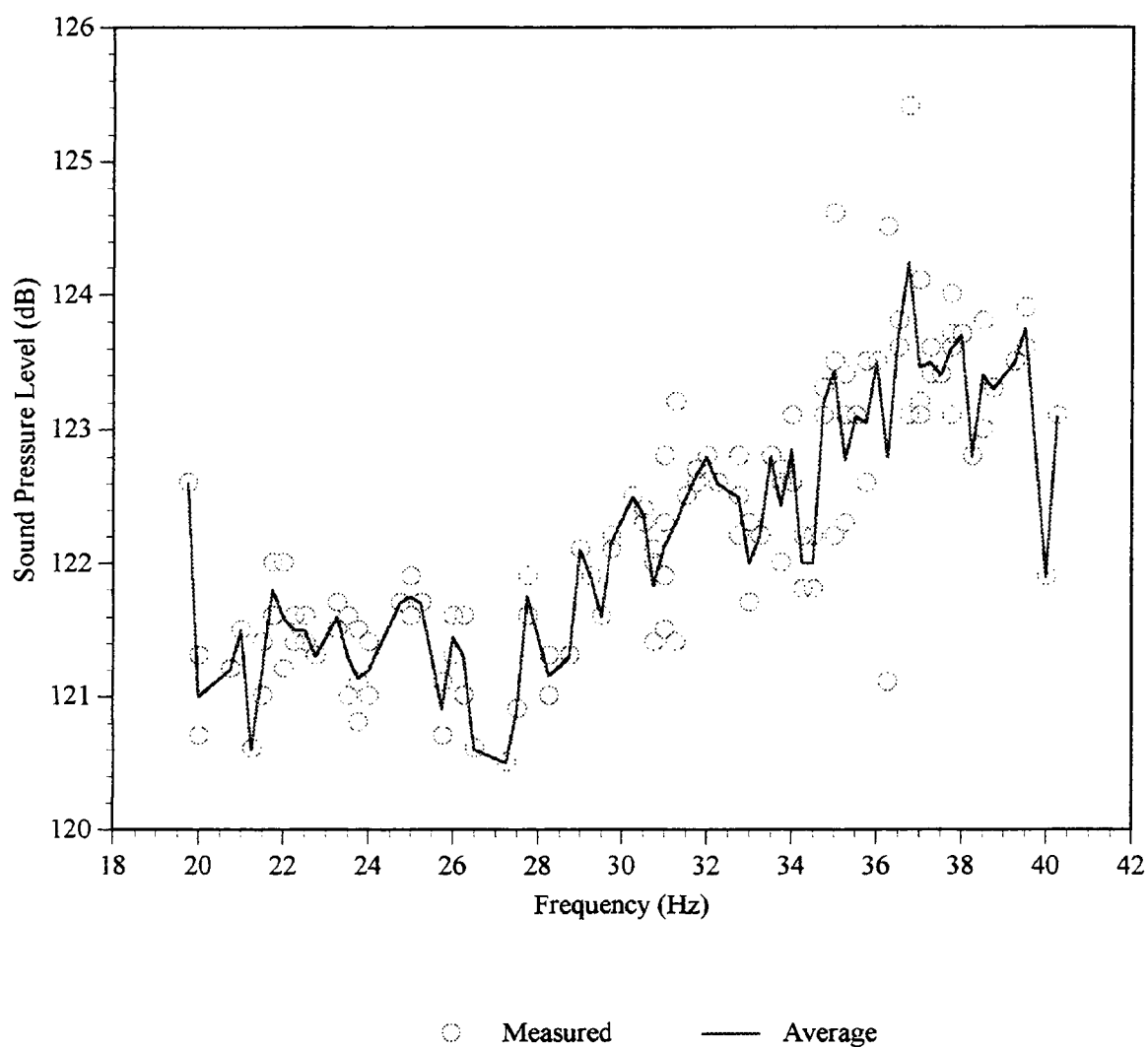


Figure 3.3: Resonance Test for the Second Channel Width.

Maximum Sound Pressure Level as
a Function of Input Frequency
Channel Width, 43.69 cm (17.2 in)

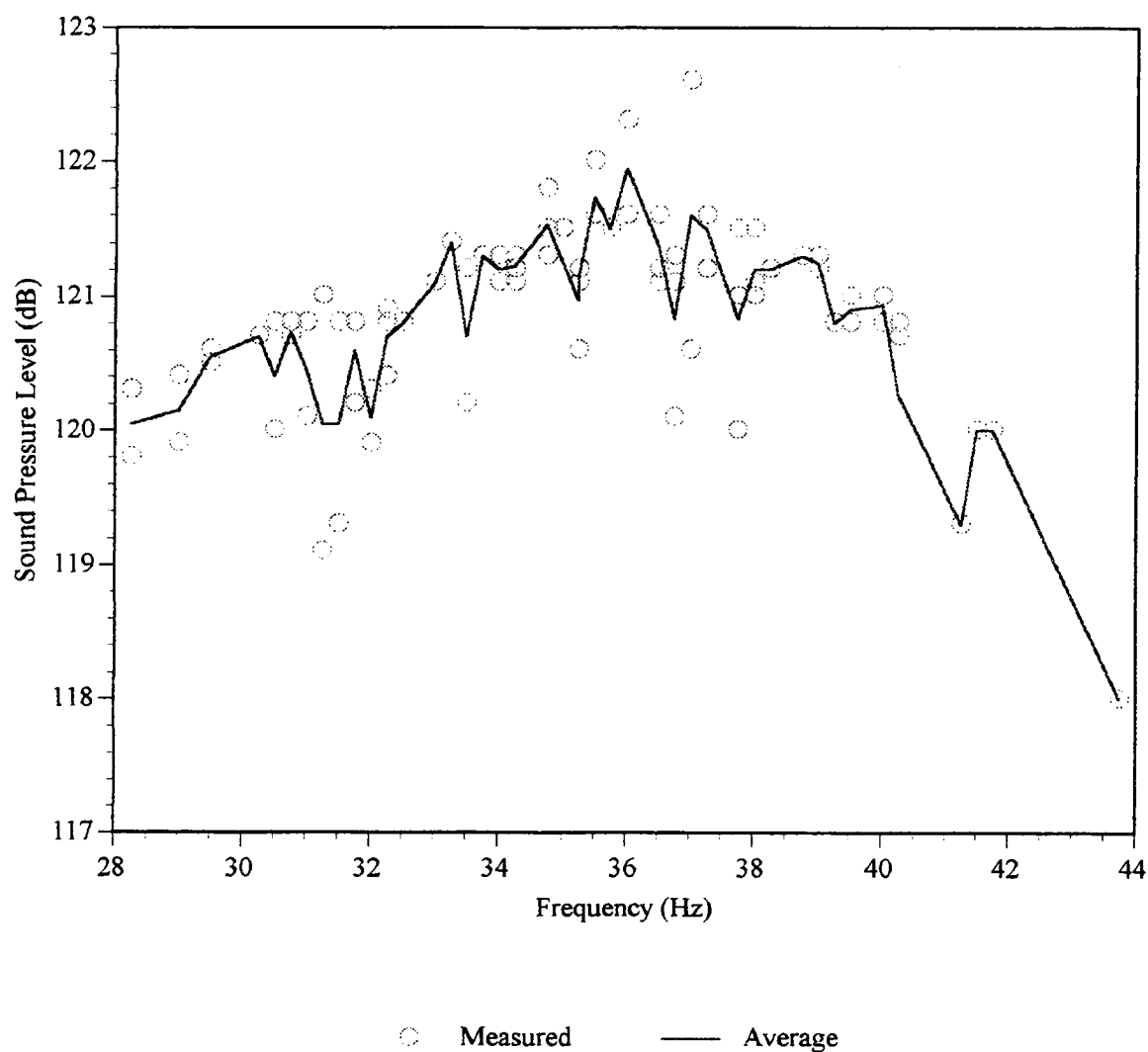


Figure 3.4 Resonance Test for the Third Channel Width.

Maximum Sound Pressure Level as
a Function of Input Frequency
Channel Width, 28.70 cm (11.3 in)
Plate Temperature, 80 °C (176 °F)

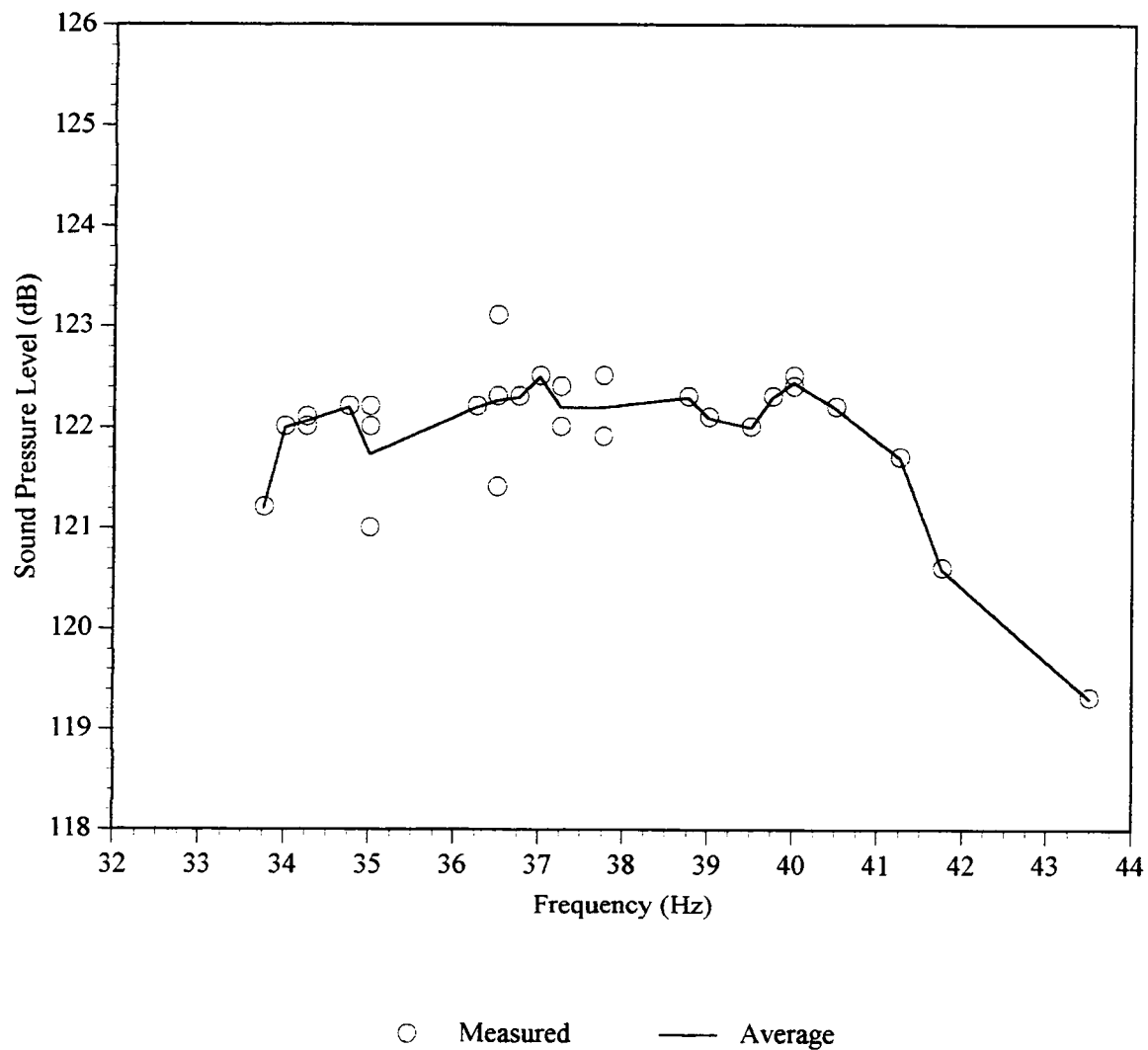


Figure 3.5: Resonance Test for Second Channel Width, Plate Temperature of 80 °C.

Maximum Sound Pressure Level as
a Function of Input Frequency
Channel Width, 28.70 cm (11.3 in)
Plate Temperature, 90 °C (194 °F)

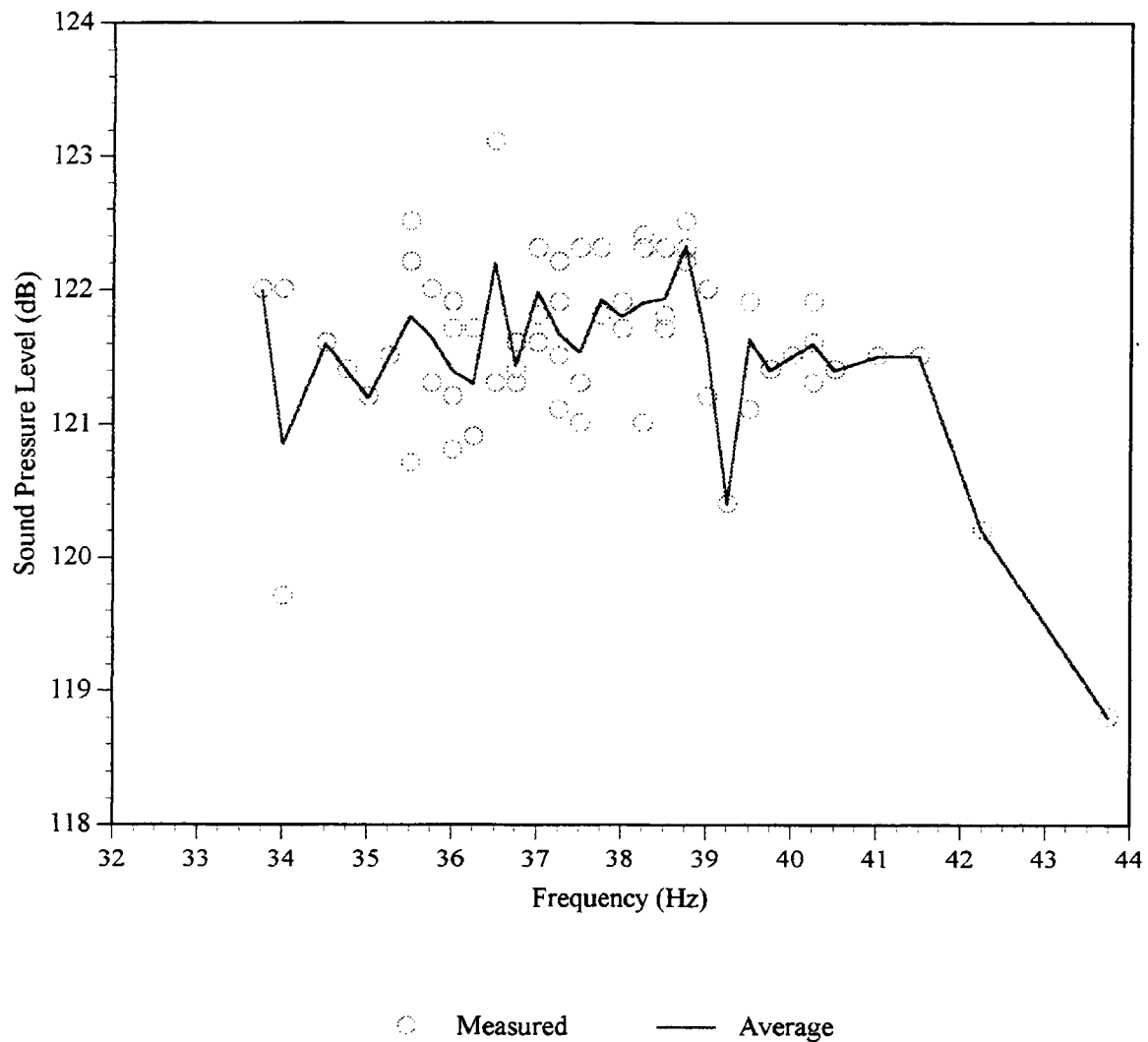


Figure 3.6: Resonance Test for Second Channel Width, Plate Temperature of 90 °C.

respectively. Results from a resonance test for the third channel width with the copper plate maintained at 90 °C (194 °F) are shown in Figure 3.7. For a temperature change from 40 °C (104 °F) to 90 °C (194 °F), the speed of sound in air changes approximately 12%. This change produced a small change in the resonant frequency of the system which was not detectable.

A hot-film anemometer was used to determine the effect of the frequency of the sound waves on the air velocity inside the channel. Connected through a constant resistance bridge to the Brüel and Kjær Dual Channel Signal Analyzer, the hot-film anemometer measured the air velocity along the channel center line at the center and 25.4 cm (10 in) to the left and right of the center. A Fast Fourier Transform of the applied bridge voltage showed the amplitude of the first harmonic of the input frequency was always larger than the fundamental frequency because the bridge voltage was always positive which produced a rectified sine wave signal with a frequency of twice the input frequency. The resonant frequency of the system was not apparent as the anemometer voltage was compared with the input frequency, Figure 3.8, for the first channel width. Since it was not possible to establish a relationship between the anemometer voltage and the input frequency for the first channel width, the hot-film anemometer was not used for the second and third channel widths. Sound frequencies between 18 Hz and 28 Hz were used for the hot-film anemometer tests. This range was set from the results of the resonance tests conducted with the microphone placed on top of the channel. These results suggested a resonant frequency of approximately 24 Hz, the resonance frequency of the speakers, which

Maximum Sound Pressure Level as
a Function of Input Frequency
Channel Width, 43.69 cm (17.2 in)
Plate Temperature, 90 °C (194 °F)

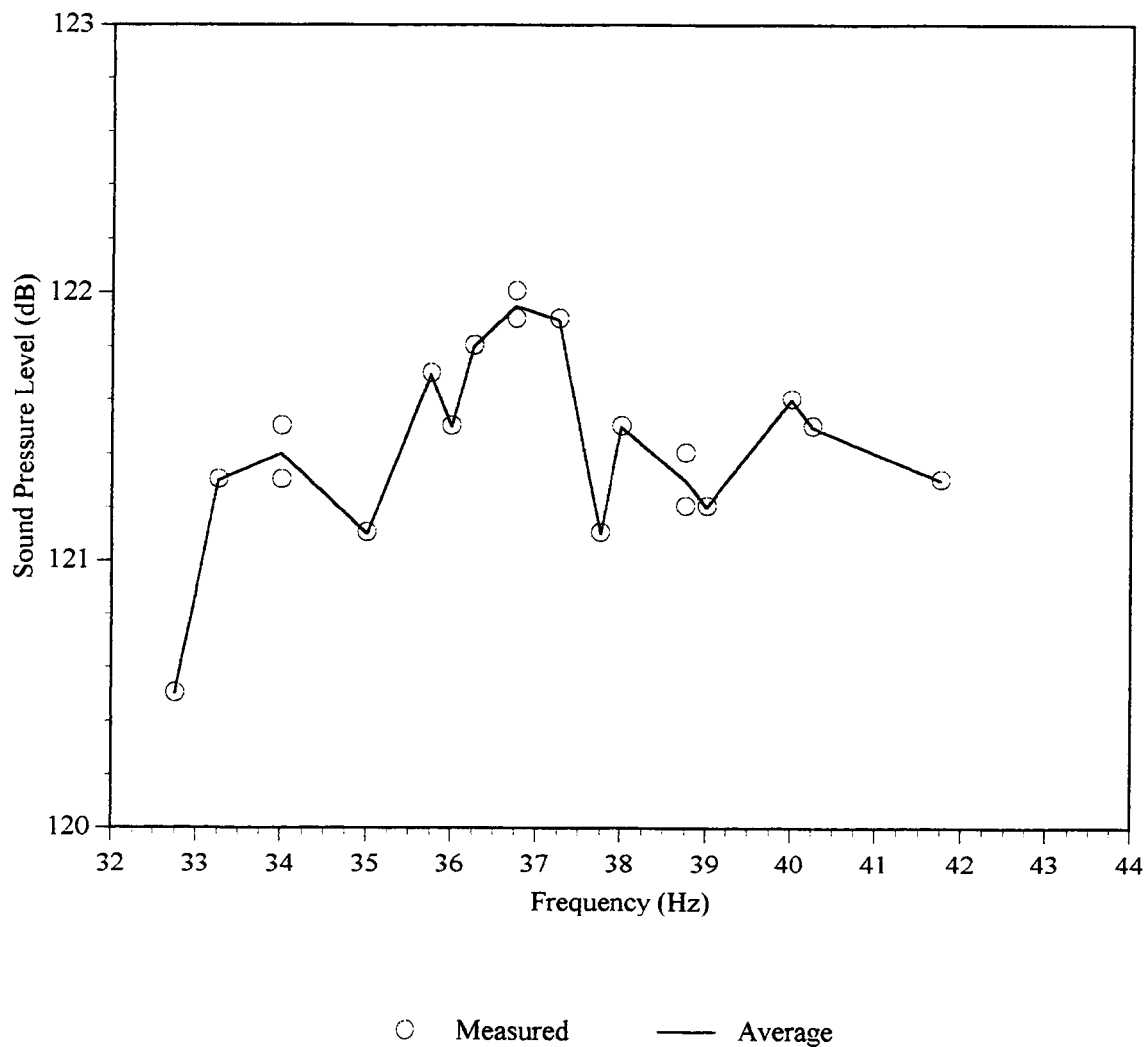
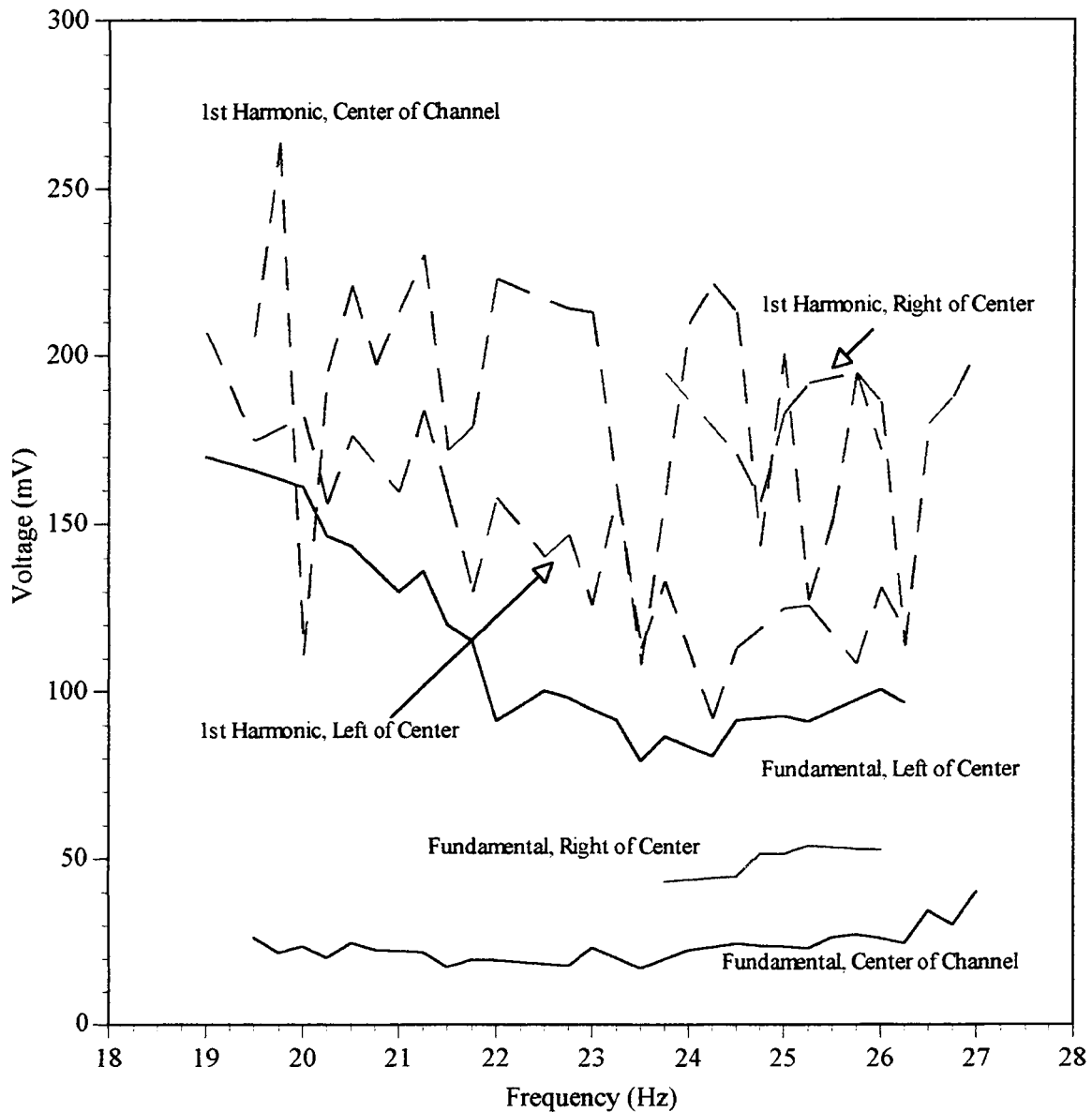


Figure 3.7: Resonance Test for Third Channel Width, Plate Temperature of 90 °C.

Hot-Film Anemometer Output Voltage as a Function of the Input Sound Frequency



Hot-Film Anemometer located at the center of the channel, 25.4 cm (10 in) to the right of the center, and 25.4 cm (10 in) to the left of center.

Figure 3.8: Hot-Film Anemometer Resonance Test for the First Channel Width.

overshadowed the system resonant frequency. This effect was eliminated by placing the microphone inside the channel. The natural frequency of the copper plate, 73.8 Hz, and the plywood channel bottom, 107.6 Hz should not have effected the resonance tests when the microphone was inside the channel, Appendix B.

Channel Air Velocity

Air velocity in the channel was controlled by adjusting the damper position on the blower. Six damper positions were used; open, three-quarters open, half open, one-quarter open, one-eighth open and closed. Lower flow rates were produced by replacing the damper with one of two plates; plate "A" had a single 0.9525 cm (0.375 in) diameter hole and plate "B" had two 0.9525 cm (0.375 in) diameter holes. Using a Kiel probe and an AM 5000 Digital Vane Anemometer, the air velocity at the channel exit was measured. The Kiel probe measured the difference between the stagnation pressure at the channel exit and atmospheric pressure for the four highest flow rates using an inclined manometer. Seven pressure measurements were taken along the diameter of the channel exit at 1.27 cm (0.5 in) increments. The air velocity at each location was calculated using Equation 3.3 [Munson et al., 1990]:

$$V = \sqrt{\frac{2\Delta P}{\rho}} \quad (3.3)$$

where: V = air velocity [m/s]
 ΔP = pressure difference [N/m²]
 ρ = air density, 1.23 [kg/m³]

From the measured velocity data, Figure 3.9, the average air velocity was assumed to be symmetrically distributed across the channel exit and the air velocity at five radial locations was obtained by averaging the velocity at symmetric locations. Using a fourth order polynomial curve-fit of the discrete velocity measurements, the radial air velocity distribution was obtained for each channel width and the four largest flow rates. The volumetric flow rate in the channel was calculated by integration of the velocity profile over the cross-sectional area of the channel exit, assuming no circumferential variations, Equation 3.4.

$$Q = \int V dA = \int V(r) r dr d\phi = \int V(r) (2\pi r dr) = 2\pi \int_{0.0 \text{ m}}^{0.0508 \text{ m}} V(r) r dr \quad (3.4)$$

where: Q = volumetric flow rate [m³/s]
 V = air velocity [m/s]
 r = channel exit radius [m]

The air velocity inside the channel was calculated by dividing the volumetric flow rate by the cross-sectional area of the channel.

Using the vane anemometer the channel exit air velocity was estimated from the average of four separate measurements taken over varying time intervals for each channel width and flow rate. The air velocity inside the channel was determined by multiplying these measurements by the ratio of the cross-sectional area of the exit to cross-sectional area of the channel. Since the vane anemometer measured the average air velocity, not the distribution, at the exit, the channel air velocities at the lower flow rates were extrapolated from the Kiel probe measurements based a linear curve fit of the air velocities calculated from the vane anemometer, Appendix D. For the second

Velocity Profile at Channel Exit Measured with a Kiel Probe

Blower Damper Fully Open

Channel Width, 15.75 cm (6.2 in)

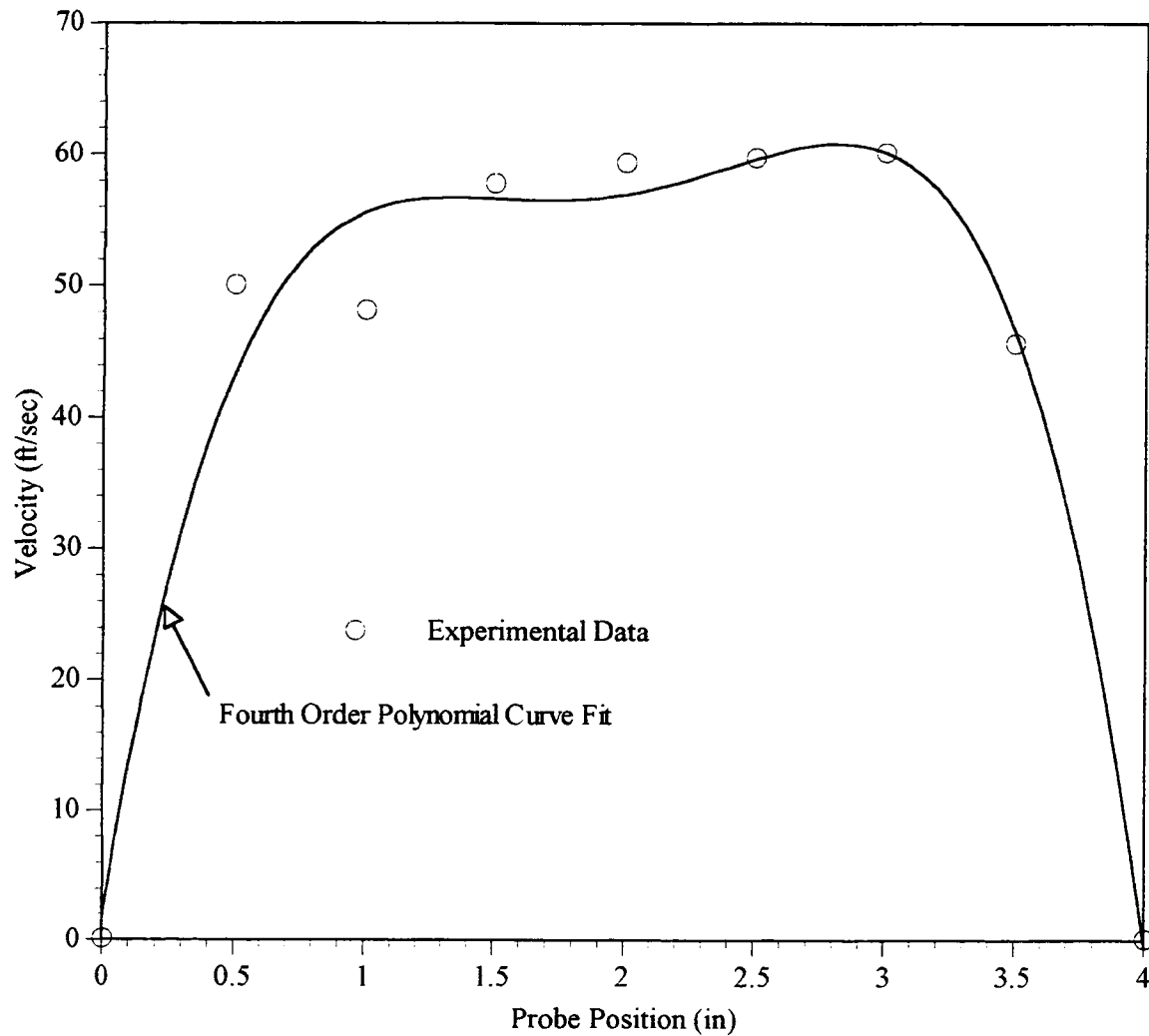


Figure 3.9: Air Velocity Distribution at the Channel Exit.

and third channel widths, an accurate correlation between the Kiel probe measurements and the vane anemometer measurements was obtained; however, the coefficient of determination, R^2 , for the first channel width was low, 0.777. The projected channel air velocities for the first channel width were used because these values more closely corresponded to projected average air velocities used for the second and third channel widths, as shown in Figure 3.10. The low air velocities experienced in the first channel width, 15.75 cm (6.2 in), indicated the frictional effects of the walls limited the effectiveness of the blower. The largest channel width, 43.69 cm (17.2 in), had the lowest air velocities, indicating a significant dispersion of the blower output. In terms of air velocity, an optimal channel width between 15.75 cm (6.2 in) and 43.69 cm (17.2 in) is suggested by the high air velocities experienced for the intermediate channel width, 27.70 cm (11.3 in).

Sensor Placement

The local temperature of the copper plate was measured using Type T thermocouples mounted on the outside of the plate in holes drilled to within 1.5875 mm (0.0625 in) of the inside surface. Six, self-generating thermopile transducers (heat flux gages) mounted on the inside surface of the plate were used to measure the heat transfer from the plate to the air. The heat flux gages were enclosed in a polyimide film, 35.05 mm by 28.45 mm (1.36 in $\times$ 1.12 in). Assuming the standing sound wave produced in the channel and its effects on the rate of heat transfer were symmetric about the long axis of the channel, the thermocouples and

Air Velocity Inside the Channel

Based on Kiel Probe and Vane Anemometer Measurements

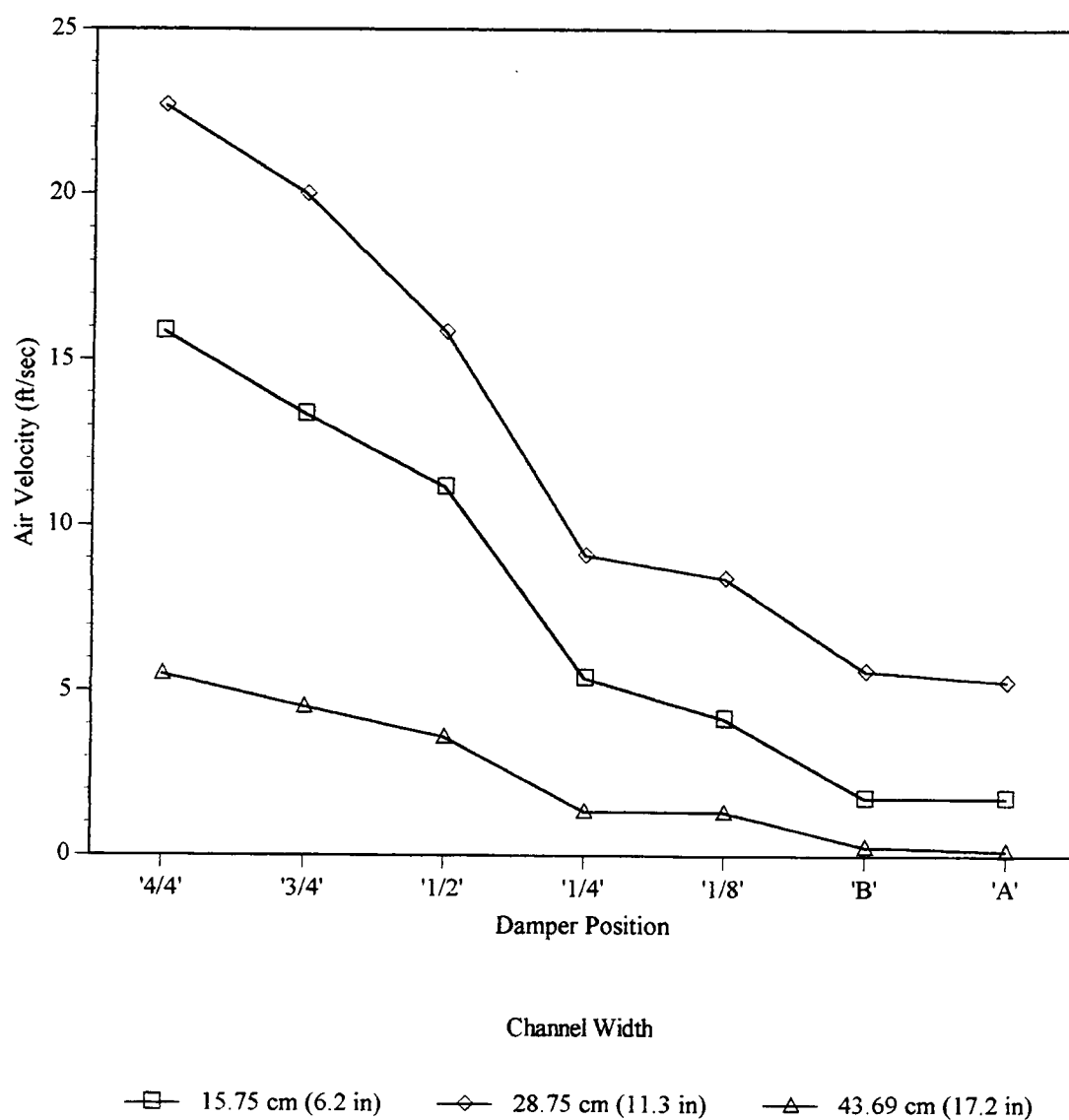


Figure 3.10: Average, Channel Air Velocity as a Function of Damper Position.

heat flux gages were positioned on one side of the long axis of the plate. Thermocouples located at the channel entrance and exit, at the center of the channel and 25.4 cm (10 in) to the left and right of the center position measured the air temperature inside the channel, as shown in Figure 3.11. One heat flux gage was mounted at center of the plate, Position 6, directly under the center thermocouple, where the effects of the infrasound were predicted to be the largest, as shown in Figure 3.12. For each channel width the position of the remaining heat flux gages on the copper plate was varied. Three heat flux gages were placed along a line parallel to long axis of channel and 2.159 cm (0.85 in) from the channel wall, with one gage at the midpoint of the line and the other gages 25.4 cm (10 in) to the left and right of the midpoint, Positions 2, 7, and 11. For the first channel width the remaining two gages were placed 2.159 cm (0.85 in) from the other channel wall, 25.4 cm (10 in) to the left and right of the short axis of the channel, Positions 1 and 10. These gages were used to verify the symmetric effects assumption. For the second and third channel widths, these gages were mounted between the long axis of the channel and the gages near the wall, 25.4 cm (10 in) to the left and right of the short axis of channel. These positions and the corresponding numbers are shown in Figure 3.12.

Data Acquisition

The thermocouples and heat flux gages were connected to a Hewlett Packard 3497A Data Acquisition / Control Unit, which was controlled by a 286 personal computer. An ice bath was used for a thermocouple reference junction. The data

acquisition system measured the output voltage from the thermocouples and heat flux gages at twenty to thirty channels per second. A computer program was developed to control the data acquisition and convert the output voltages into temperature or heat flux measurements, Appendix C. During each test all measurements were repeated fifty times and averaged to reduce the effect of any fluctuations. Based on the damper position and the calibration of blower, the average air velocity inside the channel was recorded for each test.

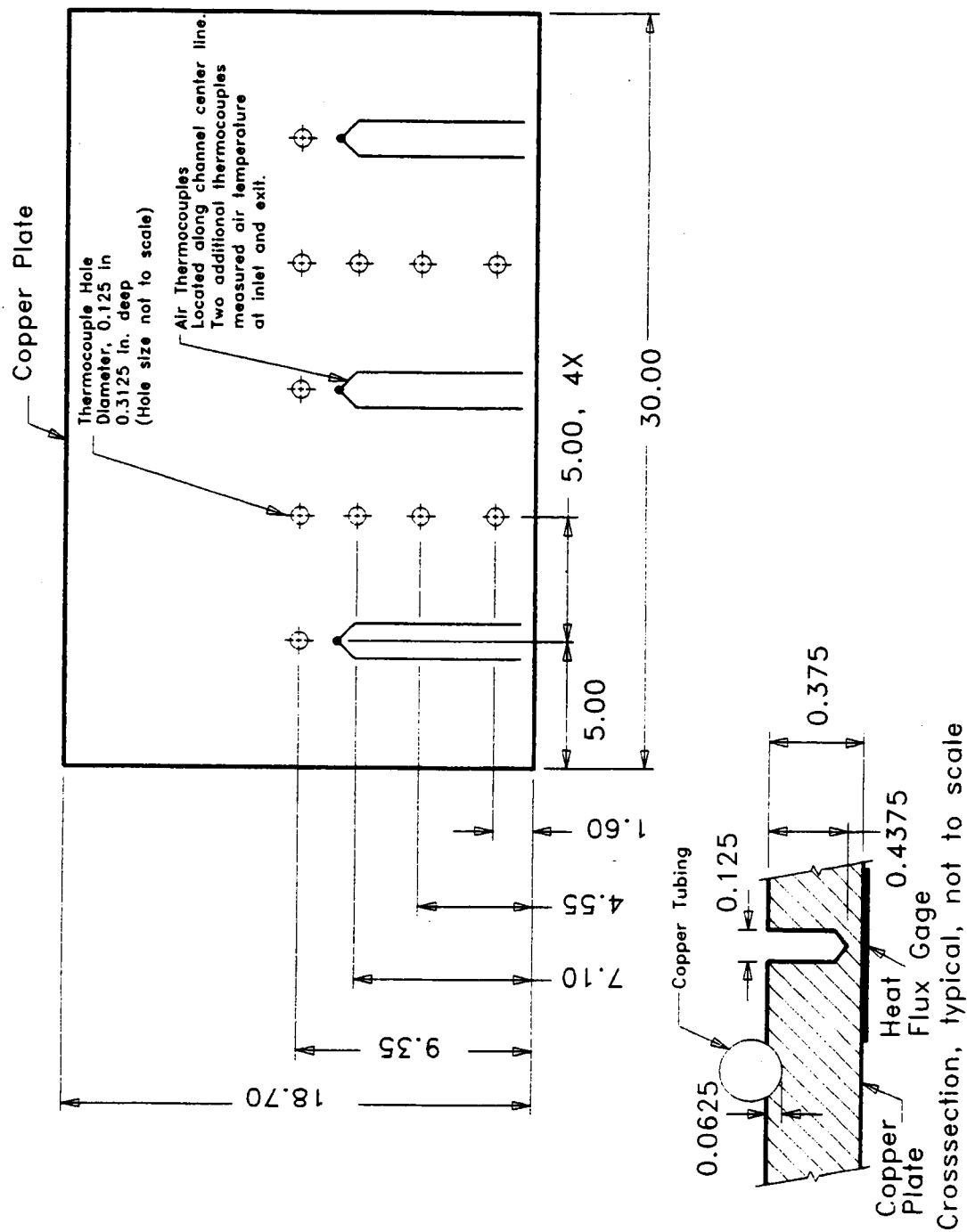


Figure 3.11: Thermocouple Positions.
Note: Dimensions in inches.

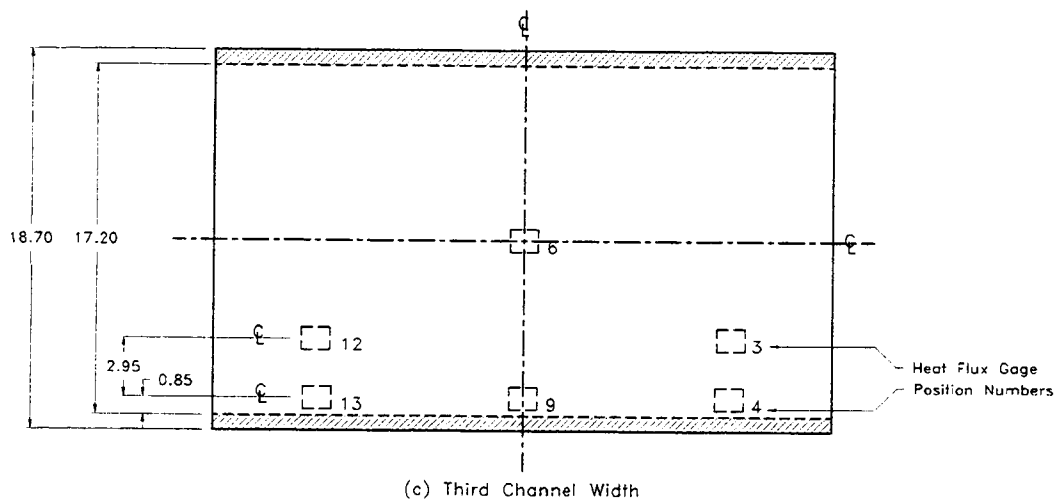
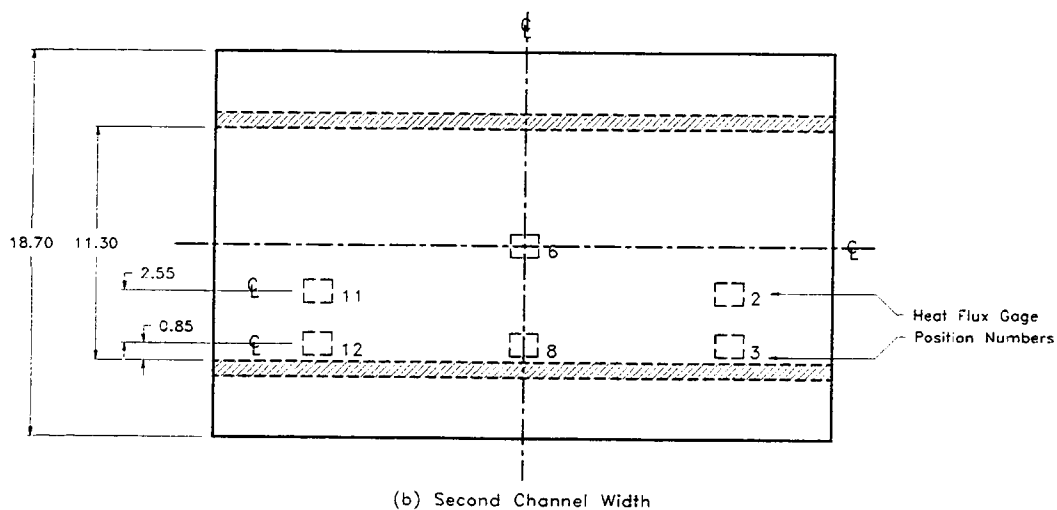
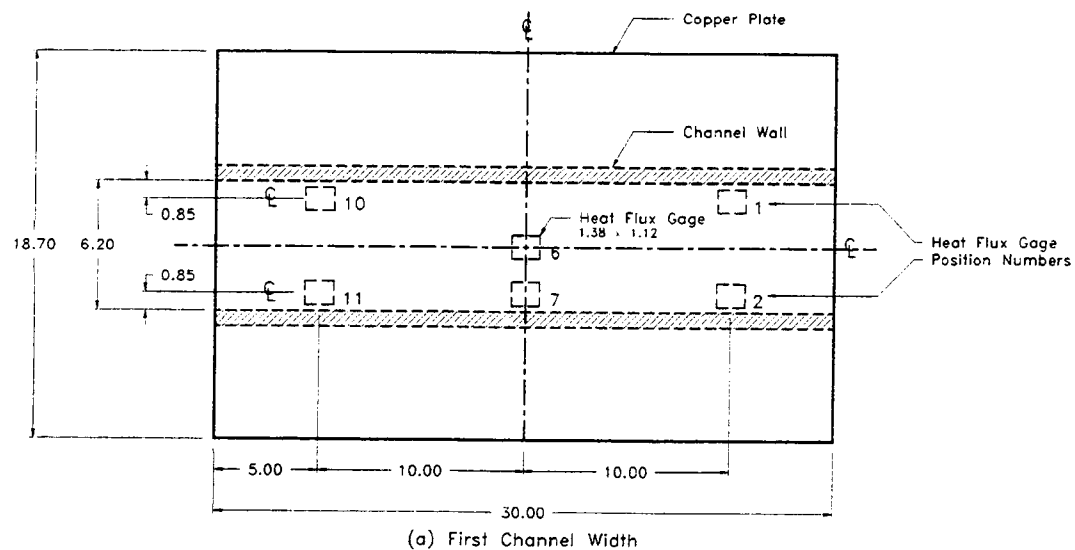


Figure 3.12: Heat Flux Gage Locations.
Note: Dimensions in inches.

CHAPTER 4

EXPERIMENTAL PROCEDURE

Test Conditions

To evaluate the effects of the infrasound on the rate of heat transfer for each channel width; 15.75 cm (6.2 in), 28.70 cm (11.3 in), and 43.69 cm (17.2 in), four different plate temperatures; 40 °C (104 °F), 60 °C (140 °F), 80 °C (176 °F), and 90 °C (194 °F) and eight different air flow rates were used. The air velocity inside the channel was controlled by adjusting the damper on the blower or replacing the damper with plates to restrict the flow to the blower inlet, the Reynolds numbers corresponding to each damper position and each channel width are listed in Table 4.1.

Table 4.1: Reynolds Numbers based on Hydraulic diameter for each Channel Width.

Damper Position	Channel Width		
	15.75 cm (6.2 in)	28.70 cm (11.3 in)	43.69 cm (17.2 in)
Closed	0	0	0
Plate "A"	3085	10716	307
Plate "B"	3075	11391	587
1/8 Open	7343	17088	2863
1/4 Open	9527	18465	2876
1/2 Open	19666	32137	7785
3/4 Open	23539	40551	9758
Open	27975	45961	11894

Calculations

The local heat transfer coefficients were calculated at each heat flux gage location, based on the average plate temperature of the surrounding thermocouples and the average air temperature inside the channel, Equation 4.1:

$$h_x = \frac{q_x''}{T_{plate} - T_{air}} \quad (4.1)$$

where:

q_x''	= local heat flux from the plate to the air	[W/m ²]
h_x	= local heat transfer coefficient	[W/m ² ·K]
T_{plate}	= local plate temperature	[K]
T_{air}	= local air temperature in the channel	[K]

At each heat flux gage location, a local Nusselt number, based on the hydraulic diameter of the channel, was calculated, Equation 4.2.

$$Nu_x = \frac{h_x D_H}{k} \quad (4.2)$$

where:

Nu_x	= local Nusselt number	
h_x	= local heat transfer coefficient	[W/m ² ·K]
D_H	= hydraulic diameter of the channel	[m]
k	= thermal conductivity of the air in the channel	[W/m·K]

The hydraulic diameter of the channel was defined as four times the ratio of the cross-sectional area divided by the wetted perimeter [Munson et al., 1990]. An empirical

formula was used to calculate the thermal conductivity of the air in the channel based on the average air temperature in the channel [Figliola and Beasley, 1991].

CHAPTER 5

DISCUSSION OF RESULTS

Heat Transfer Without Acoustic Enhancement

The experimentally measured Nusselt numbers were verified by comparing the results without acoustic enhancement to values predicted by theoretical equations. In the absence of additional air flow and infrasound, natural convection effects govern the heat transfer from the copper plate to the air. Assuming the ends of the channel are sealed, the Nusselt number for a shallow enclosure and heated upper surface is determined by Equation 5.1, [Bejan, 1984] which is the lower limit of heat transfer given by conduction.

$$Nu = \frac{Q}{kH\Delta T/L} = 0.364 \frac{L}{H} Ra_H^{1/4} \quad (5.1)$$

where:

Nu	= Nusselt number	
Q	= heat flux	[W/m]
k	= thermal conductivity of air	[W/m·K]
H	= channel height, 6.35 cm (2.5 in)	[m]
ΔT	= temperature difference between the plate and the air	[K]
L	= channel length, 80 cm (31.5 in)	[m]
Ra_H	= Rayleigh number, based on channel height	

If the resonance chamber and the speaker box are assumed to be infinite reservoirs at a constant temperature and the air is free to circulate in the resonance chamber and the speaker box, the average Nusselt number for the lower surface of a heated plate is calculated using Equation 5.2 [Incropera and DeWitt, 1990]:

$$\overline{Nu}_L = 0.27 Ra_L^{1/4} \quad (10^5 \leq Ra_L \leq 10^{10}) \quad (5.2)$$

where: $\overline{Nu}_L$ = average Nusselt number based on the characteristic length
 Ra_H = Rayleigh number, based on the characteristic length
 $L \equiv A_s / P$, characteristic length [m]
 A_s = surface area of the plate [m²]
 P = perimeter of the plate [m]

The Nusselt numbers, as a function of the plate temperature, predicted by Equations 5.1 and 5.2 provide an lower and upper range for the average, measured Nusselt number. As seen in Figure 5.1, the experimental values for the first channel width, 15.75 cm (6.2 in), are more closely approximated by Equation 5.1. The relationship between the theoretical results and experimental values for the second and third channel widths was similar, as shown in Appendix D.

Symmetric Effects of Infrasound

The effects of the infrasound were assumed to be symmetric about the long axis of the channel; accordingly, the local Nusselt number calculated from the output of the heat flux gage at Position 1 (as defined in Figure 3.12) should have been equal the local Nusselt number calculated from the output of the heat flux gage at Position 2. Similarly, the local Nusselt numbers corresponding to the heat flux gages at Positions 10 and 11 should have been equivalent. However, the overall average Nusselt number at Position 1 was 25 percent greater than the overall average Nusselt number at Position 2 and the overall average Nusselt number at Position 10 was 20 percent less than the overall average Nusselt number at Position 11. The difference

Nusselt Number as a function of Plate Temperature on the
Lower Surface of a Heated Plate for Natural Convection
Channel Width 15.75 cm (6.2 in)

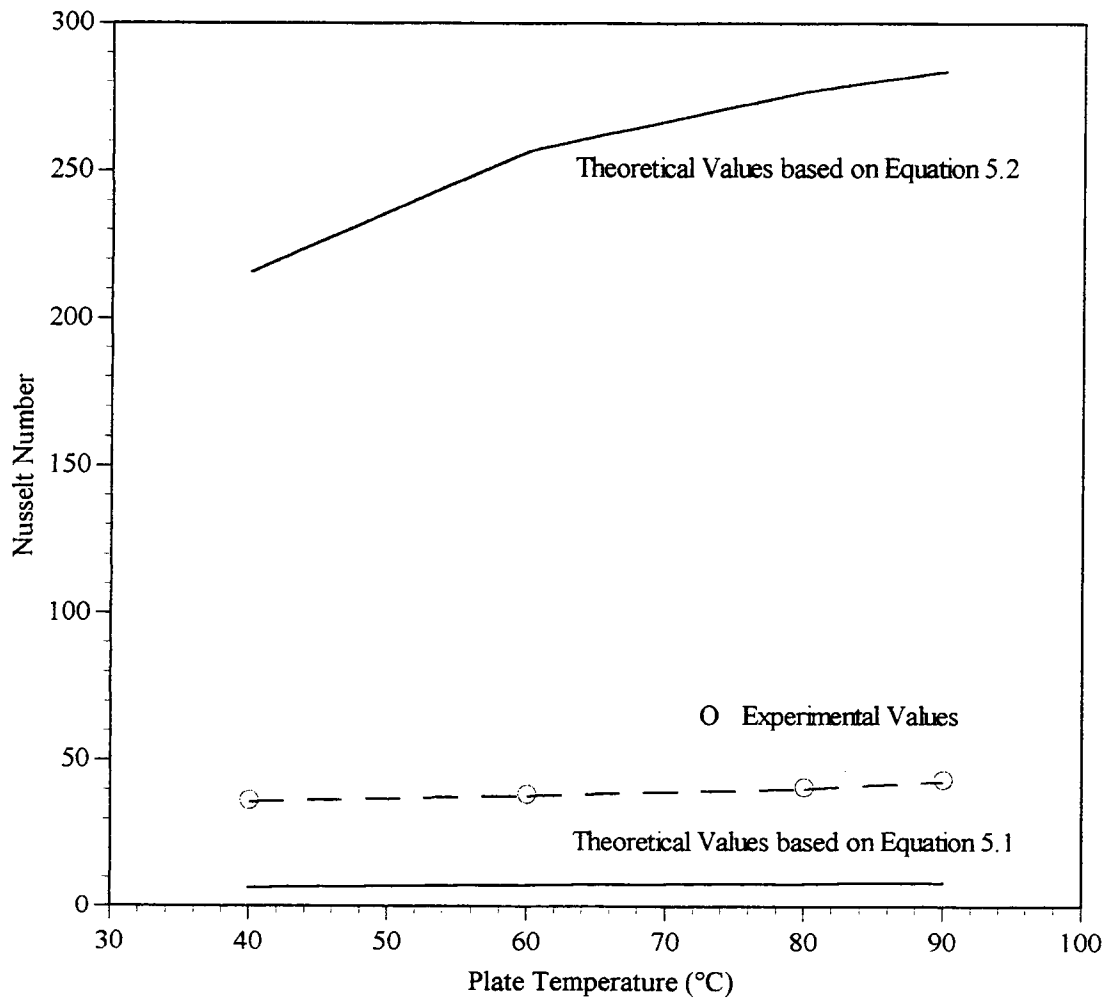


Figure 5.1: Average Nusselt Number as a Function of Plate Temperature.

between the local Nusselt numbers measured at Positions 1 and 2, and 10 and 11 divided by the local Nusselt number at Positions 1 and 10, respectively is shown in Figures 5.2 and 5.3. Positions 1 and 2 were located 15.24 cm (6 in) downstream of where the air flow entered the channel and Positions 10 and 11 were located 15.24 cm (6 in) upstream of the where the air flow exited the channel. The air flow entered the channel normal to the copper plate and the heat flux gages. After entering the channel the direction of the air flow was altered 90 degrees, parallel to the long axis of the channel. At the channel exit the direction of the air flow was again altered 90 degrees. The air flow did not enter the channel symmetrically because of space limitations for the air duct. The difference between the Nusselt number with infrasound and zero air flow near the channel entrance, 75%, and near the channel exit, 28%, may have been caused by a non-symmetric standing wave in the channel. This is most likely the cause of the lack of symmetry found in this study.

Variation of the Local Nusselt Number

A ratio of the Nusselt number with infrasonic enhancement divided by the Nusselt number without infrasonic enhancement was used to evaluate the effects of the infrasound. Based on theoretical velocity distribution inside the channel, the largest rate of heat transfer should occur at the center of the channel. As shown in Figure 5.4, at the first channel width, 15.75 cm (6.2 in) the effectiveness of the infrasound was concentrated at Reynolds numbers below 3000 with the largest effects at zero air flow, indicating infrasound was most effective when compared to natural convection. For

Percent Difference between Nusselt number with Infrasound measured at Symmetric Gage Positions near the Channel Entrance

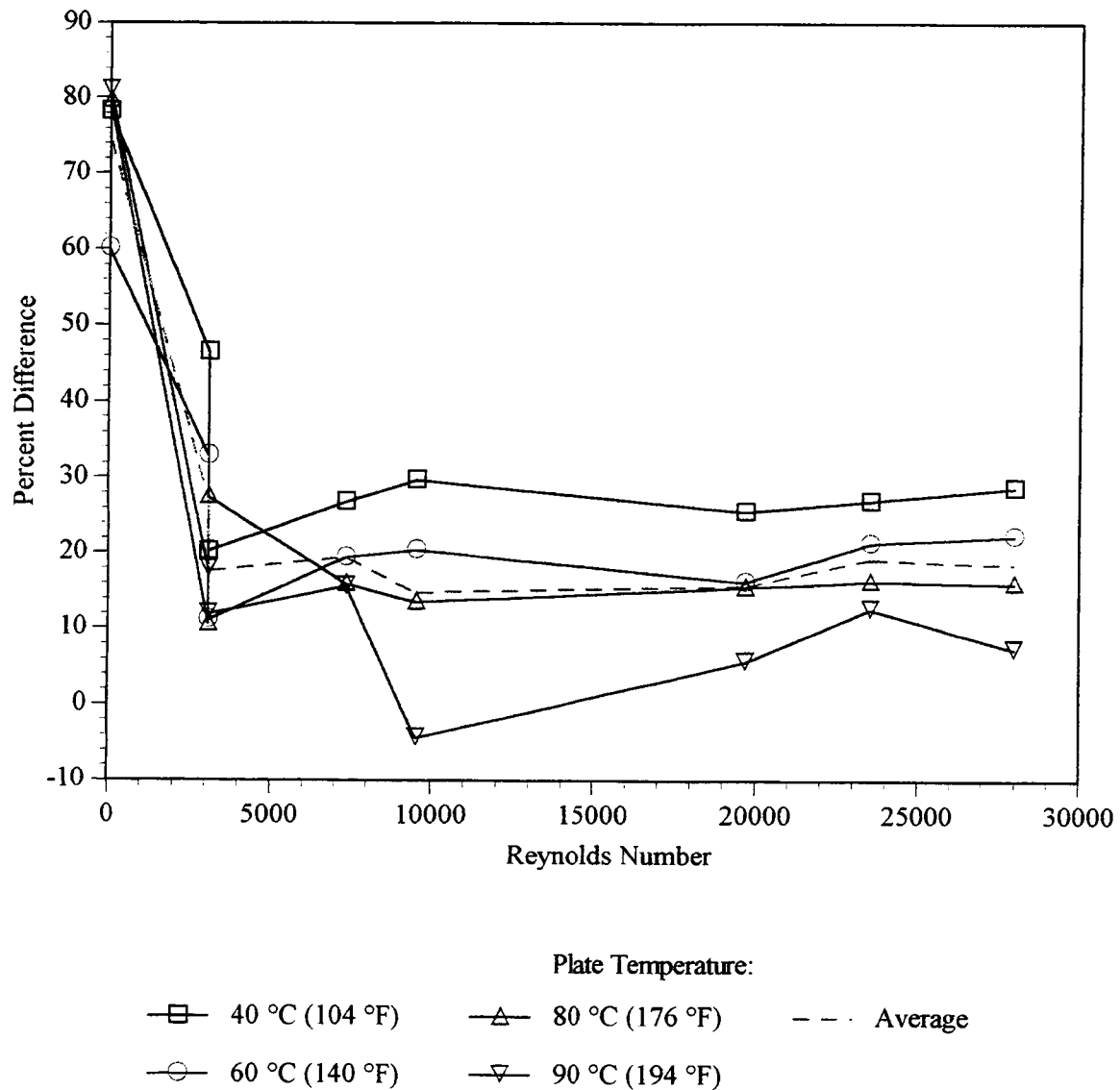


Figure 5.2: Difference between Nusselt numbers near the Channel Entrance.

Percent Difference between Nusselt number with Infrasound measured at Symmetric Gage Positions near the Channel Exit

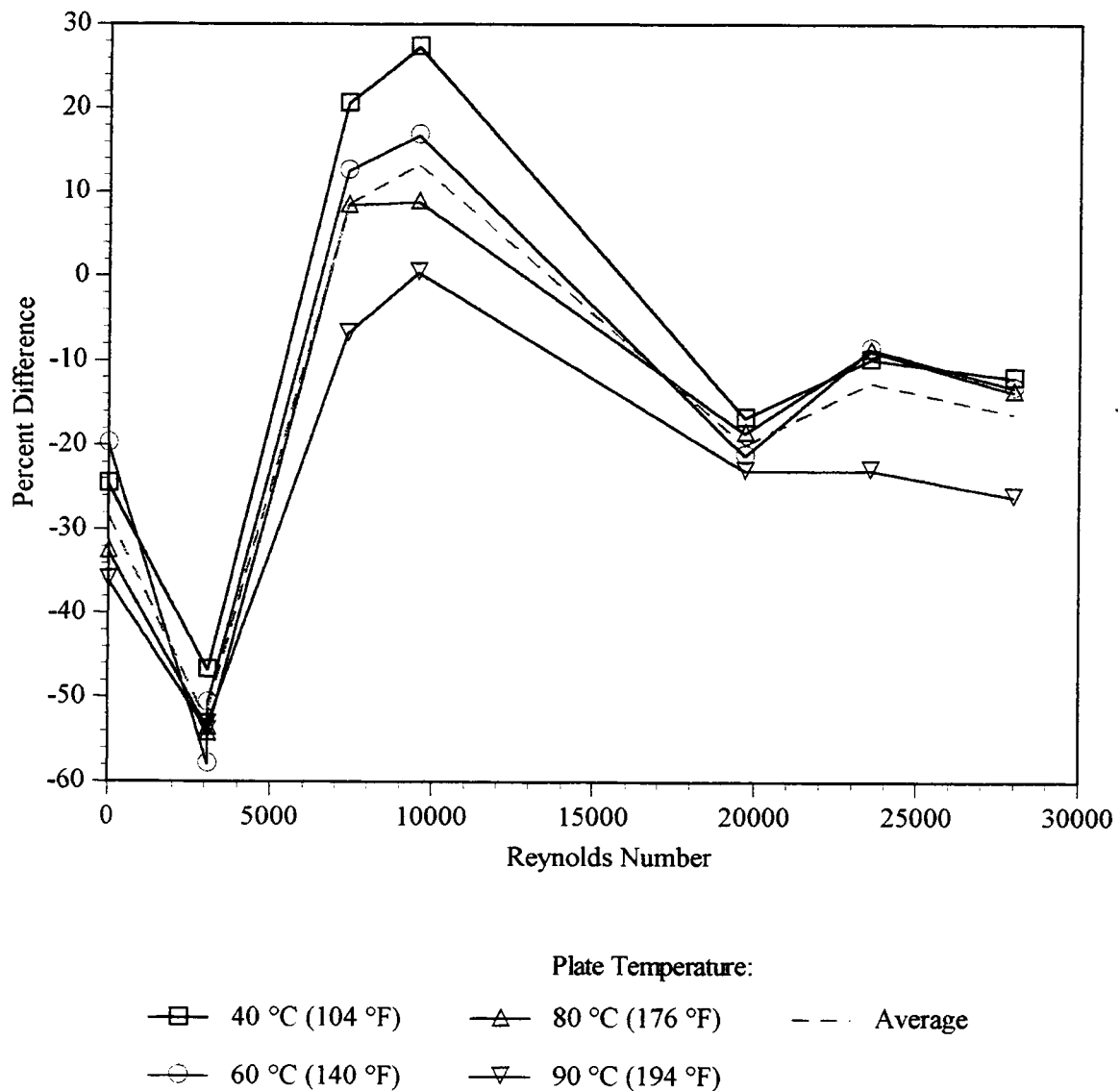


Figure 5.3: Difference between Nusselt numbers near the Channel Exit.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at the Channel Center, Position 6
Channel Width, 15.75 cm (6.2 in)**

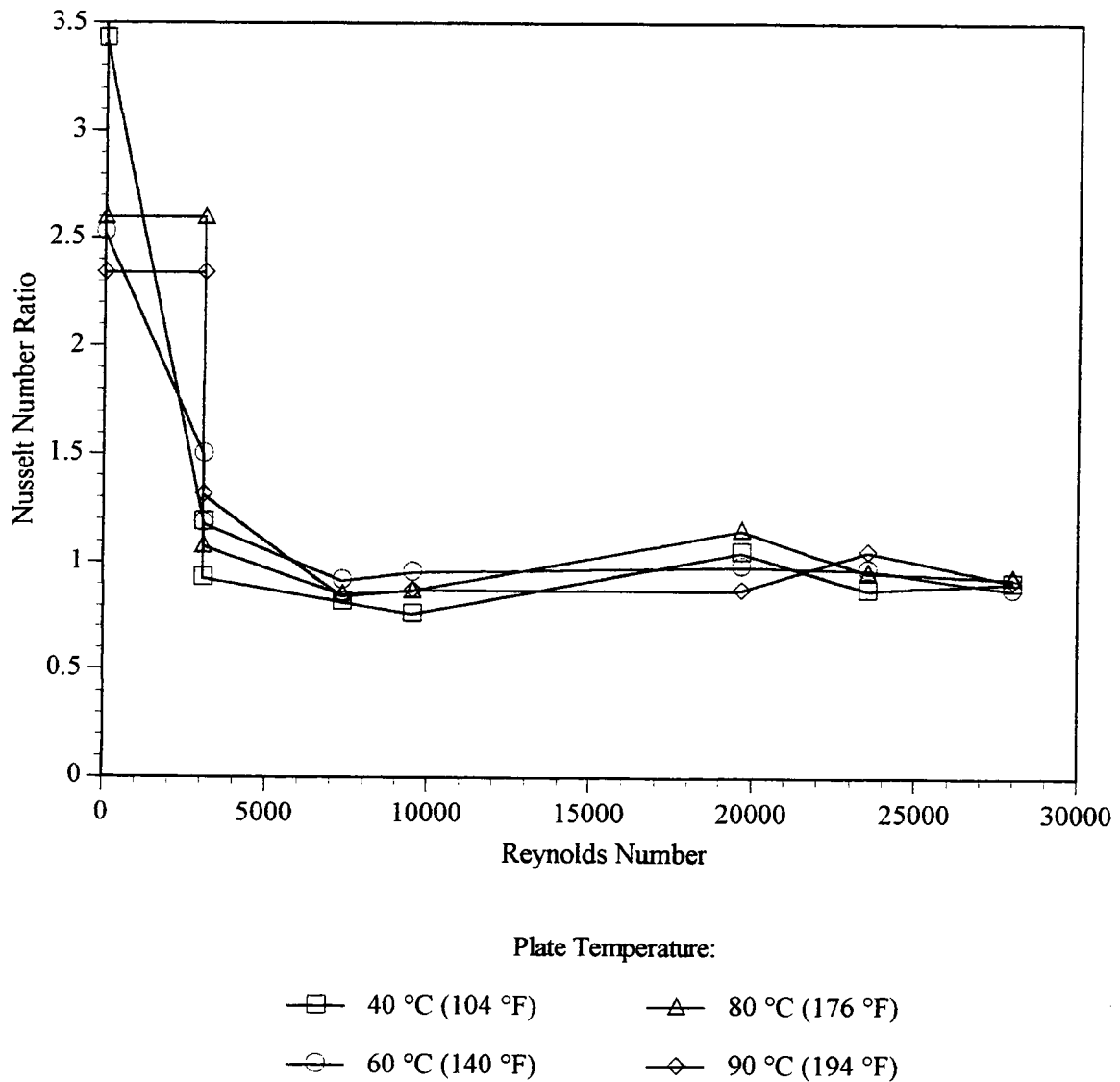


Figure 5.4: Nusselt Number Ratio at the Center of the First Channel Width.

Reynolds numbers greater than 5000, the air flow dominated the effects of the standing wave. Along the short axis of the channel, 6.35 mm (0.25 in) from the channel wall, Position 7, the Nusselt number ratio reached a maximum value for a Reynolds number of approximately 3000. At this position the air flow did not dominate the effects of the infrasound until the Reynolds number was greater than 16,000, Figure 5.5. Near the channel wall, 25.4 cm (10 in), upstream from the short axis of the channel, Positions 1 and 2, the Nusselt number ratio approached unity as the Reynolds number approached 5000, Figures 5.6 and 5.7. The larger Nusselt number ratio at Position 2 was a result of turbulent flow at the channel entrance. Downstream of the short axis of the channel, near the channel wall, Positions 10 and 11, the infrasonic effects extend over a larger Reynolds number range, between 0 and 9000, Figures 5.8 and 5.9, with a possible maximum Nusselt number ratio for a Reynolds number of 3000.

As listed in Table 4.1 the largest Reynolds numbers were experienced for the second channel width, 28.70 cm (11.3 in). Accordingly, the Nusselt number ratio for Reynolds numbers between zero and 10,000 was not measured for the second channel width. As shown in Figure 5.10, the Nusselt number ratio was approximately one as the Reynolds number approached 10,000. The local maximum of the Nusselt number ratio for a constant plate temperature of 60 °C (140 °F) is partially accounted for by the $\pm 8.47\%$ uncertainty in the Nusselt number ratio, Appendix F. At the center of the channel, near the wall, Position 8, Nusselt number ratio reached a maximum values near a Reynolds number of 11,000 and the Nusselt number ratio was greater than one

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 7; Channel Width, 15.75 cm (6.2 in)**

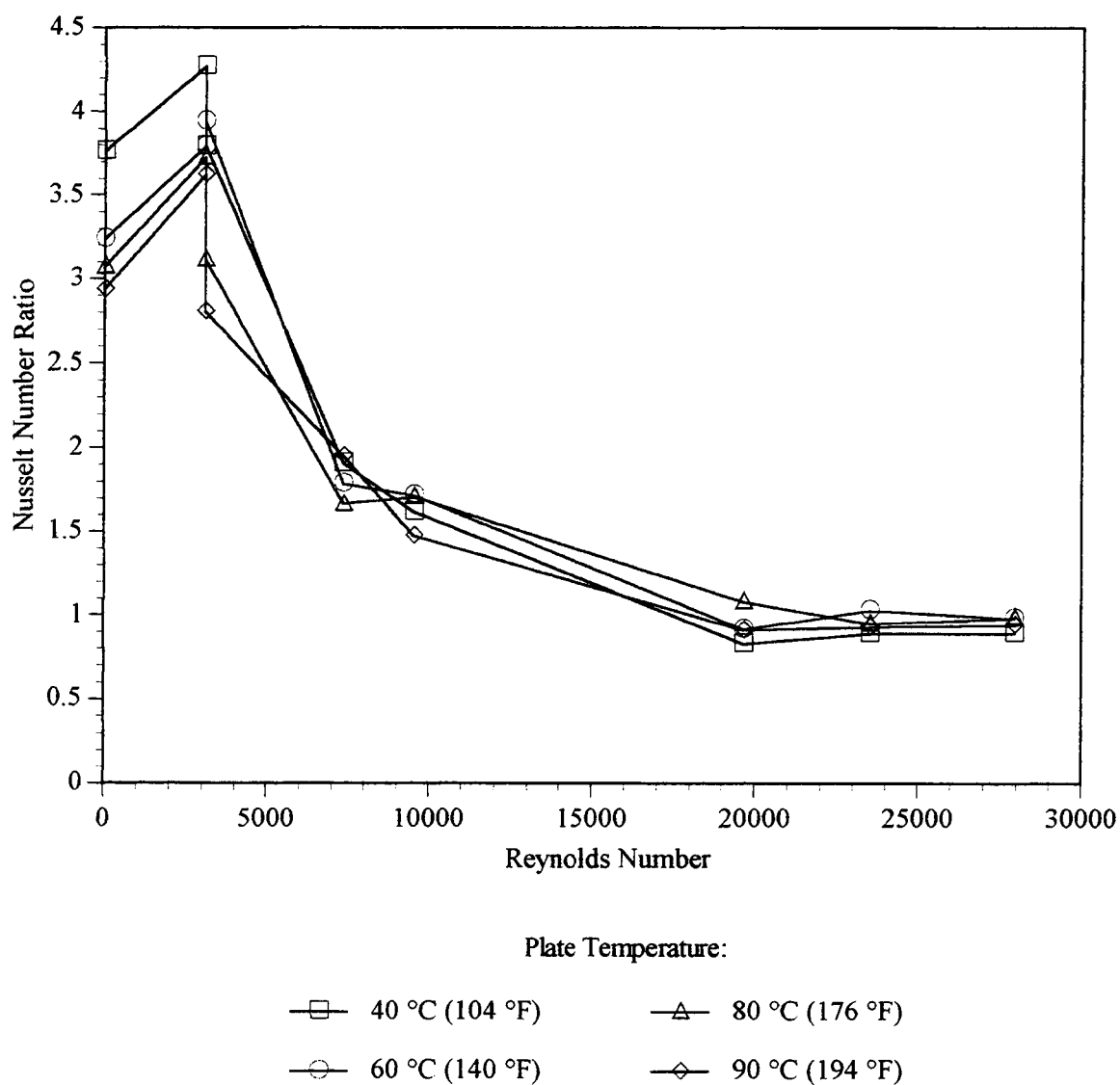


Figure 5.5: Nusselt Number Ratio for the First Channel Width, Position 7.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 1; Channel Width 15.75 cm (6.2 in)**

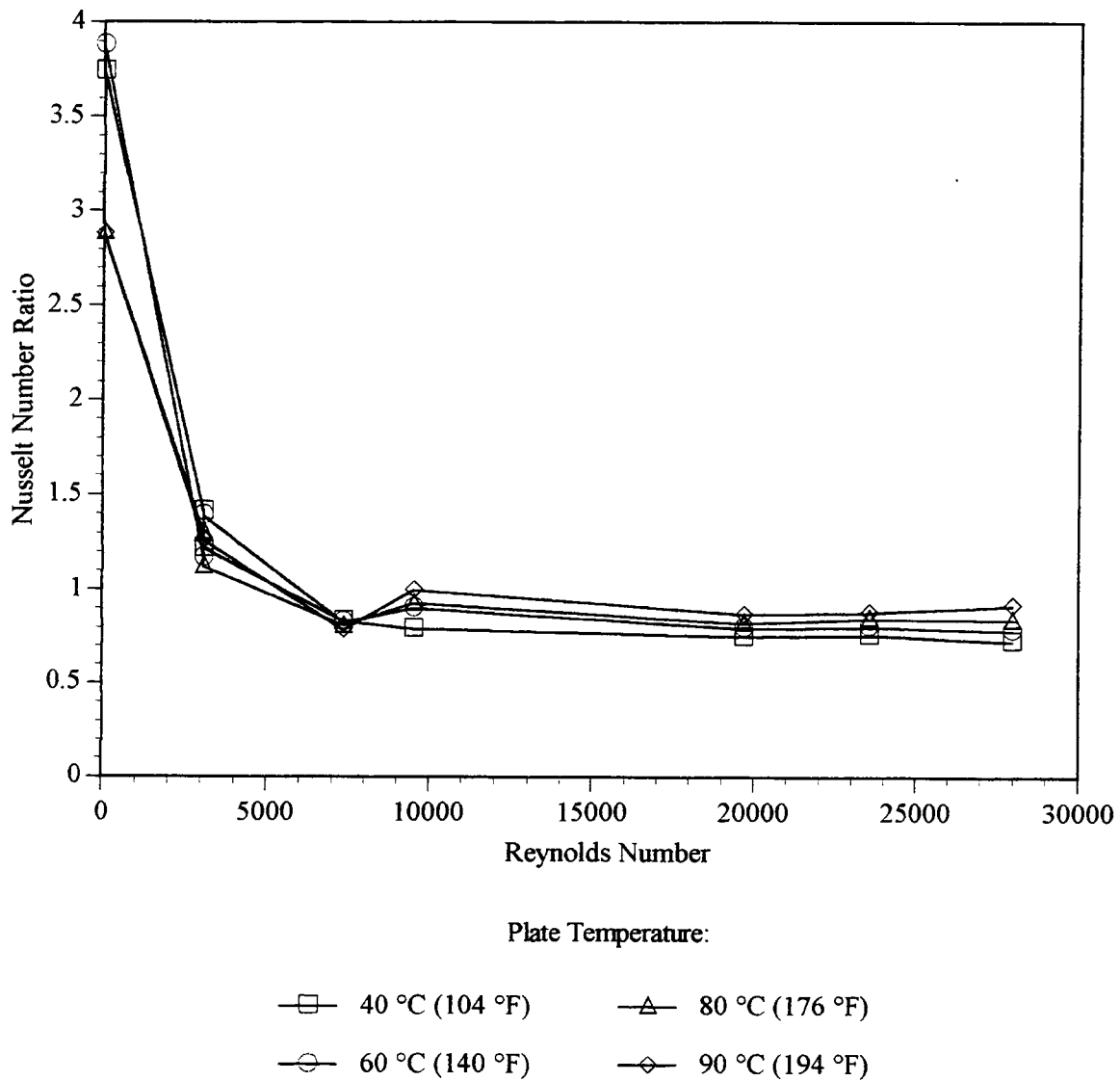


Figure 5.6: Nusselt Number Ratio for the First Channel Width, Position 1.

Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 2; Channel Width 15.75 cm (6.2 in)

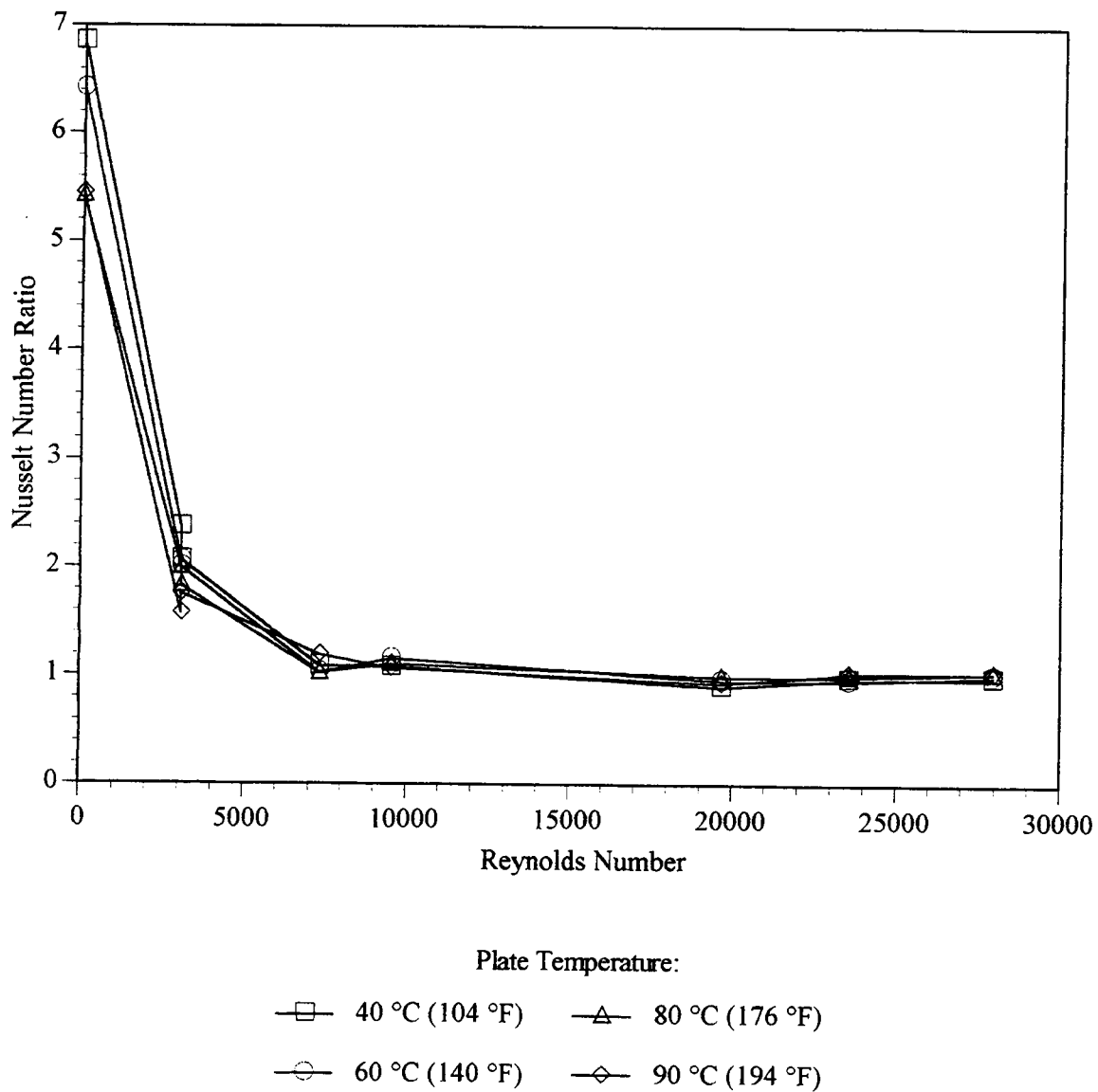


Figure 5.7: Nusselt Number Ratio for the First Channel Width, Position 2.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 10; Channel Width, 15.75 cm (6.2 in)**

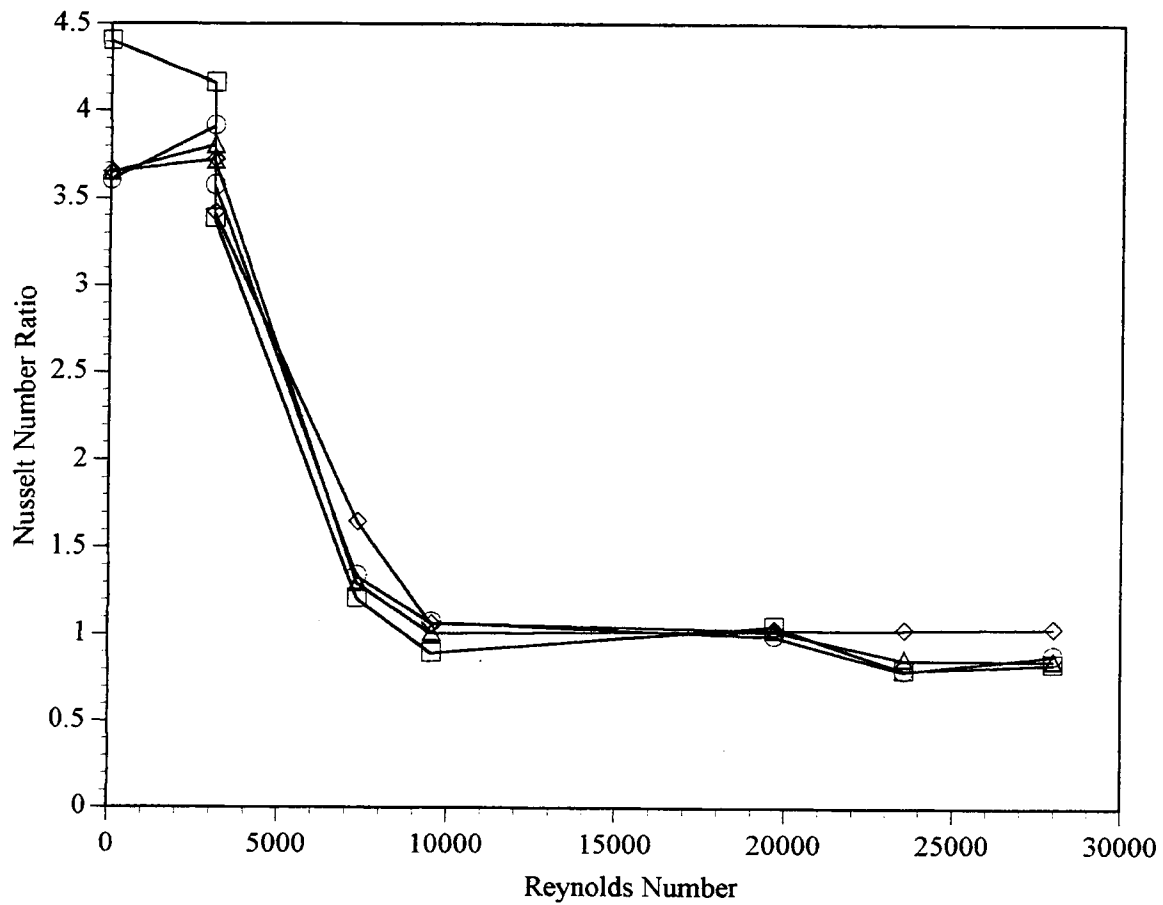


Plate Temperature:



Figure 5.8: Nusselt Number Ratio for the First Channel Width, Position 10.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 11; Channel Width 15.75 cm (6.2 in)**

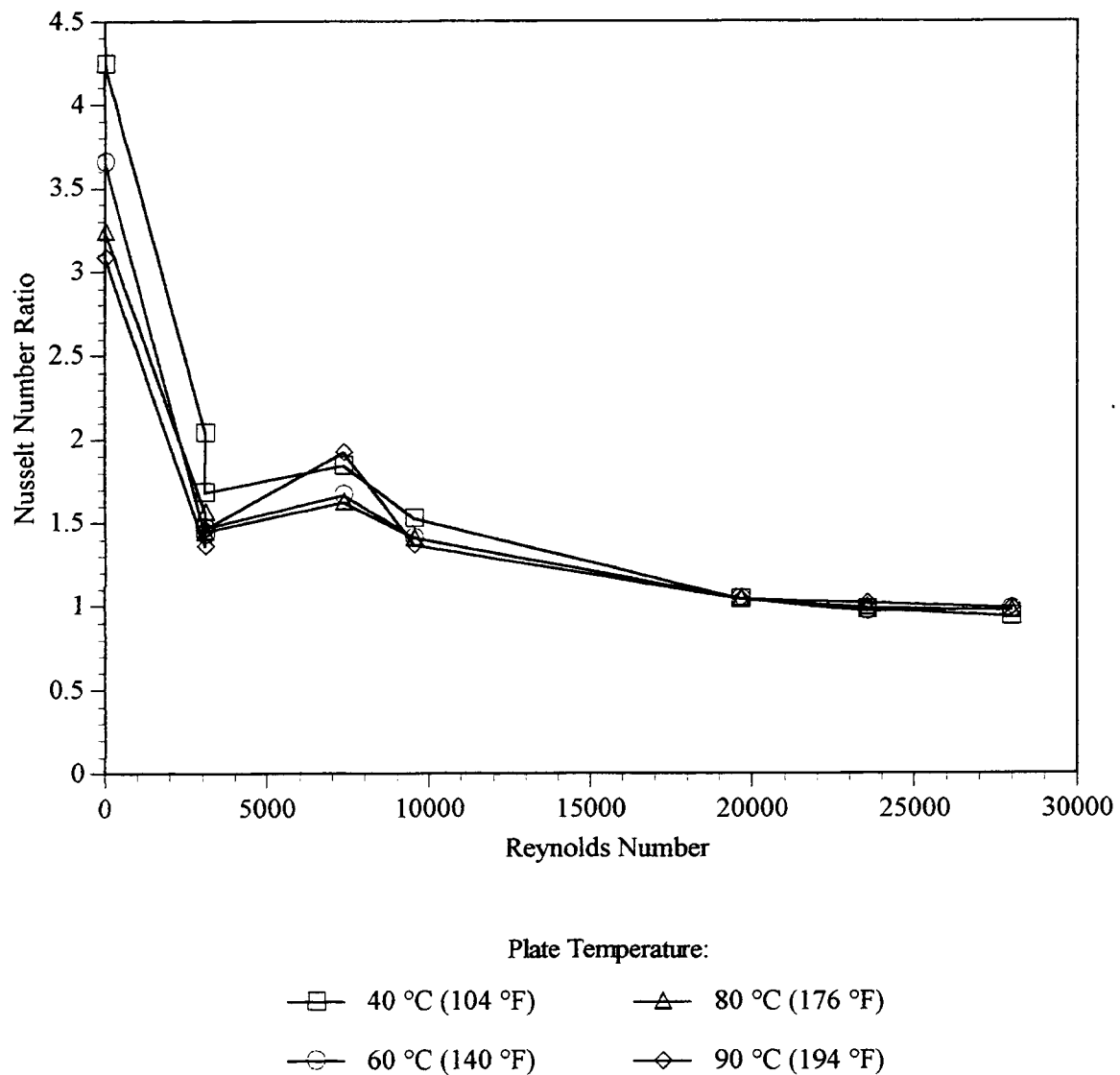


Figure 5.9: Nusselt Number Ratio for the First Channel Width, Position 11.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at the Channel Center; Position 6
Channel Width, 28.70 cm (11.3 in)**

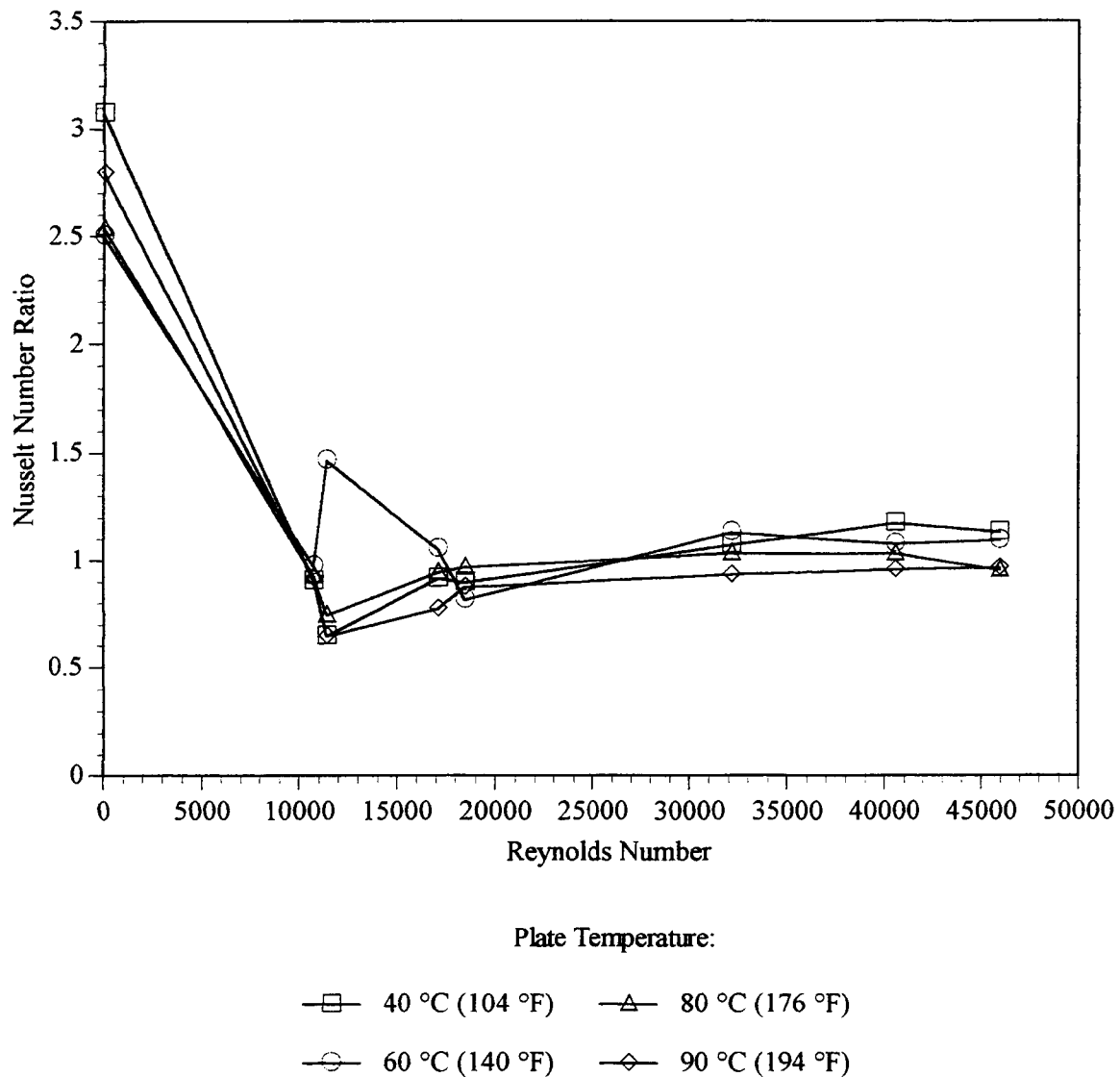


Figure 5.10: Nusselt Number Ratio at the Center of the Second Channel Width.

for Reynolds numbers less than 20,000, Figure 5.11, which is similar to Position 7 for the first channel width, Figure 5.7. As a function of the Reynolds number, the local Nusselt number ratio at the remaining positions for the second channel width exhibited similar trends, with an approximate, effective Reynolds number range of zero to 15,000, as shown in Figures 5.12 - 5.15.

As a result of the low air velocities encountered in the third channel width and channel width, 43.69 cm (17.2 in), the maximum Reynolds number obtained for the third channel width, 11,800, was within the effective Reynolds number range for the second channel width. However, at the center of the third channel, Position 6, the infrasound was only effective over a small range of Reynolds numbers, 0 to 1000, Figure 5.16. This range of effective Reynolds numbers was extended for locations away from the long axis of the channel, as shown in Figures 5.17 - 5.21.

The distribution of the local Nusselt Number ratio for the first channel width, 15.75 cm (6.2 in), was estimated through scaling the measured values and assuming symmetry. Assuming the heat transfer along the walls to be zero, the Nusselt number ratio was forced to zero at the walls and large gradients in the Nusselt number ratio near the side walls were produced, as shown in Figure 5.22. Indicated by contour lines regions where the Nusselt number ratio was constant suggest the infrasound was most effective, relative to conventional forced convection, along the channel walls where the local air velocity was smallest. In contrast, the maximum local Nusselt number with infrasonic enhancement occurred along the channel center line, Figure 5.23.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 8; Channel Width, 28.70 cm (11.3 in)**

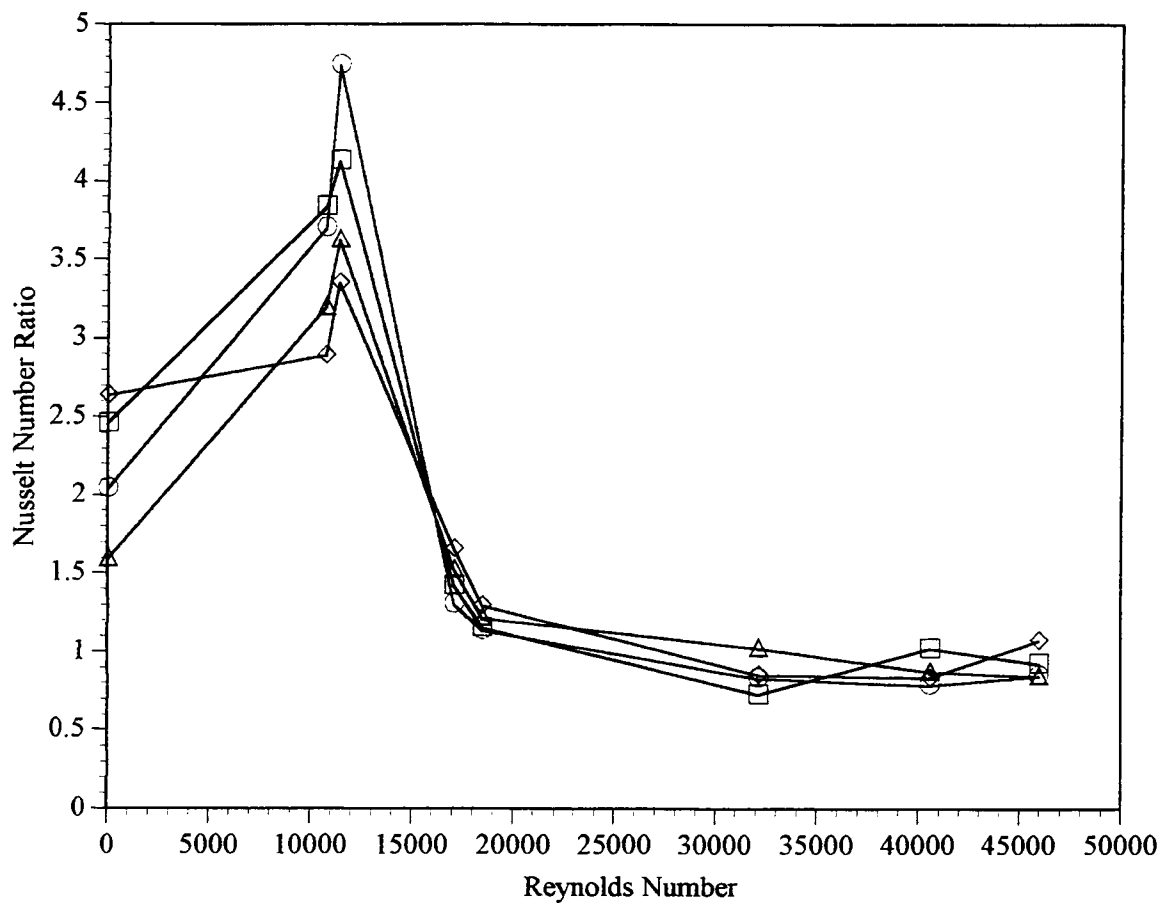


Plate Temperature:



Figure 5.11: Nusselt Number Ratio for the Second Channel Width, Position 8.

Ratio of Nusselt Number with Infrasound to Nusselt Number without Infrasound at Position 2; Channel Width, 28.70 cm (11.3 in)

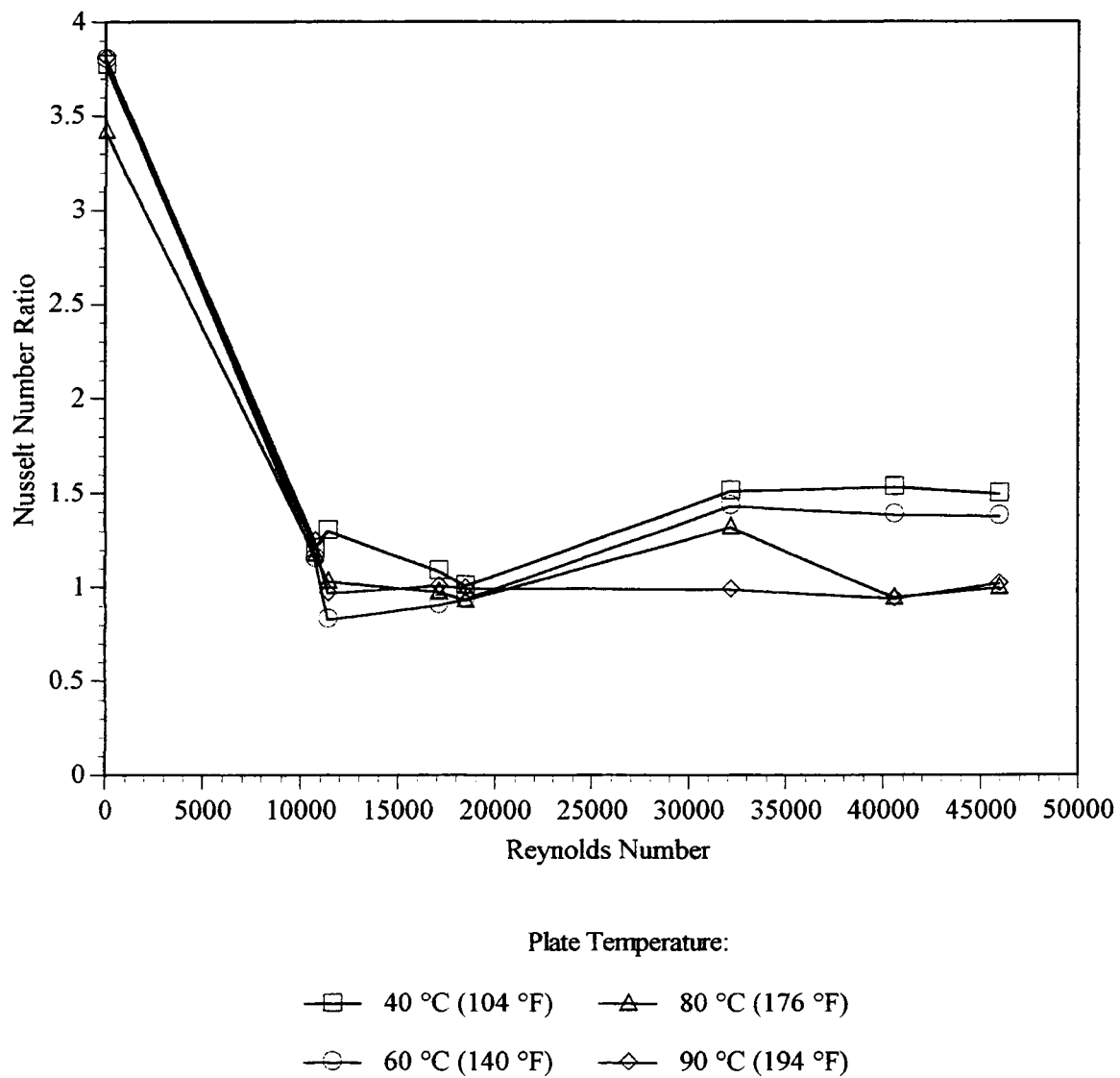


Figure 5.12: Nusselt Number Ratio for the Second Channel Width, Position 2.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 3; Channel Width, 28.70 cm (11.3 in)**

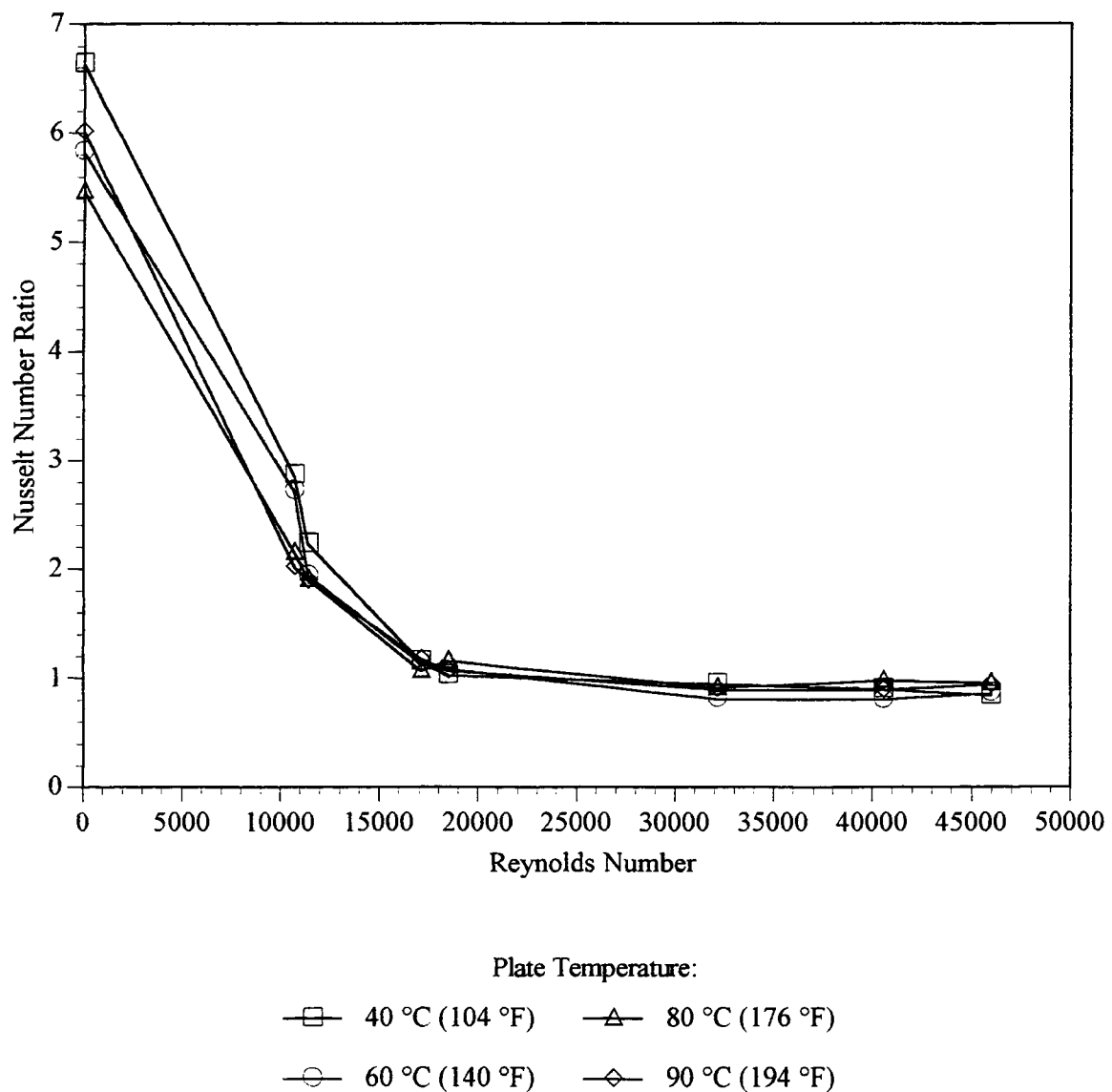


Figure 5.13: Nusselt Number Ratio for the Second Channel Width, Position 3.

**Ratio of Nusselt Number with Infrasonic
to Nusselt Number without Infrasonic
at Position 11; Channel Width, 28.70 cm (11.3 in)**

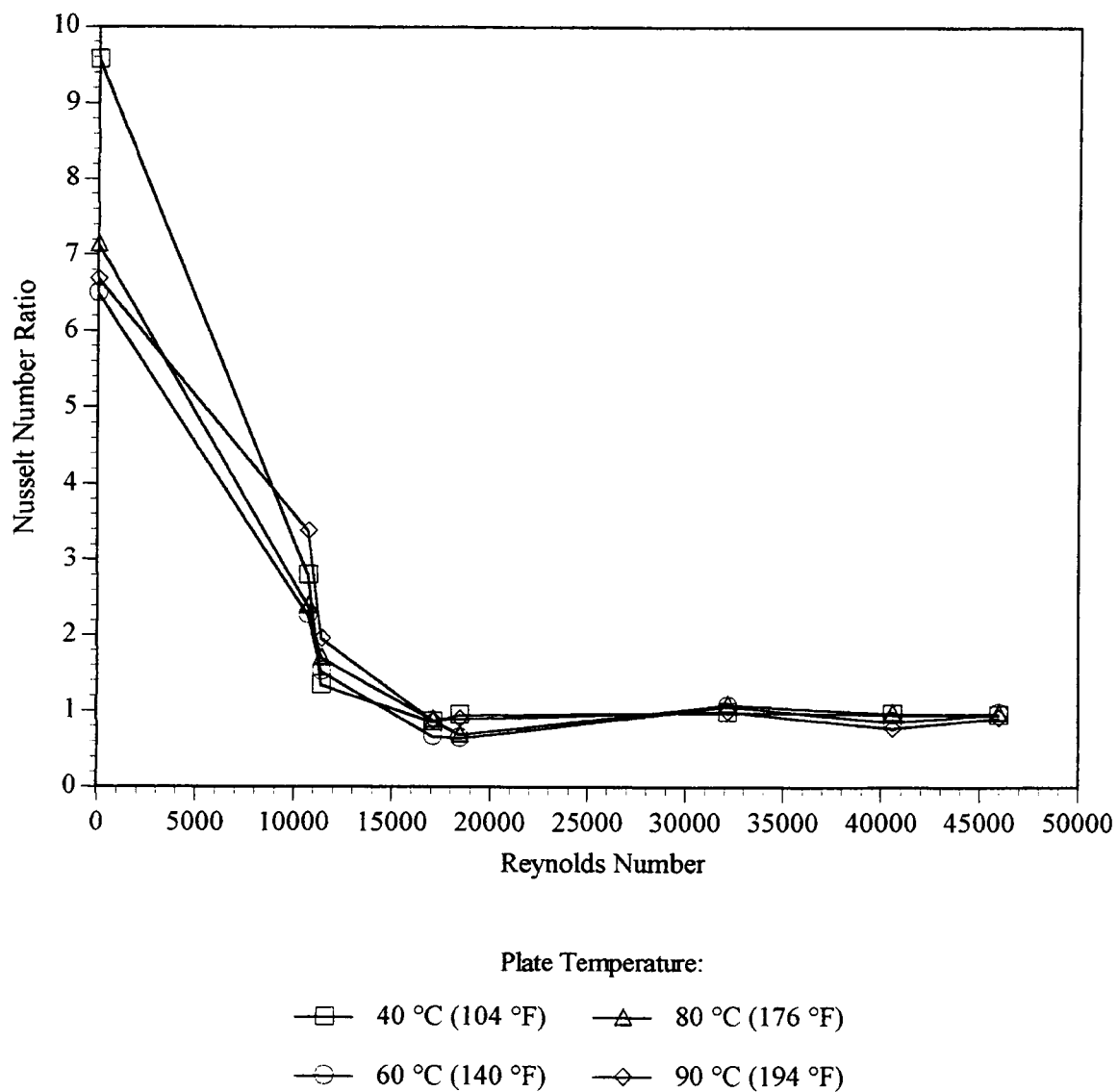


Figure 5.14: Nusselt Number Ratio for the Second Channel Width, Position 11.

Ratio of Nusselt Number with Infrasound to Nusselt Number without Infrasound at Position 12; Channel Width, 28.70 cm (11.3 in)

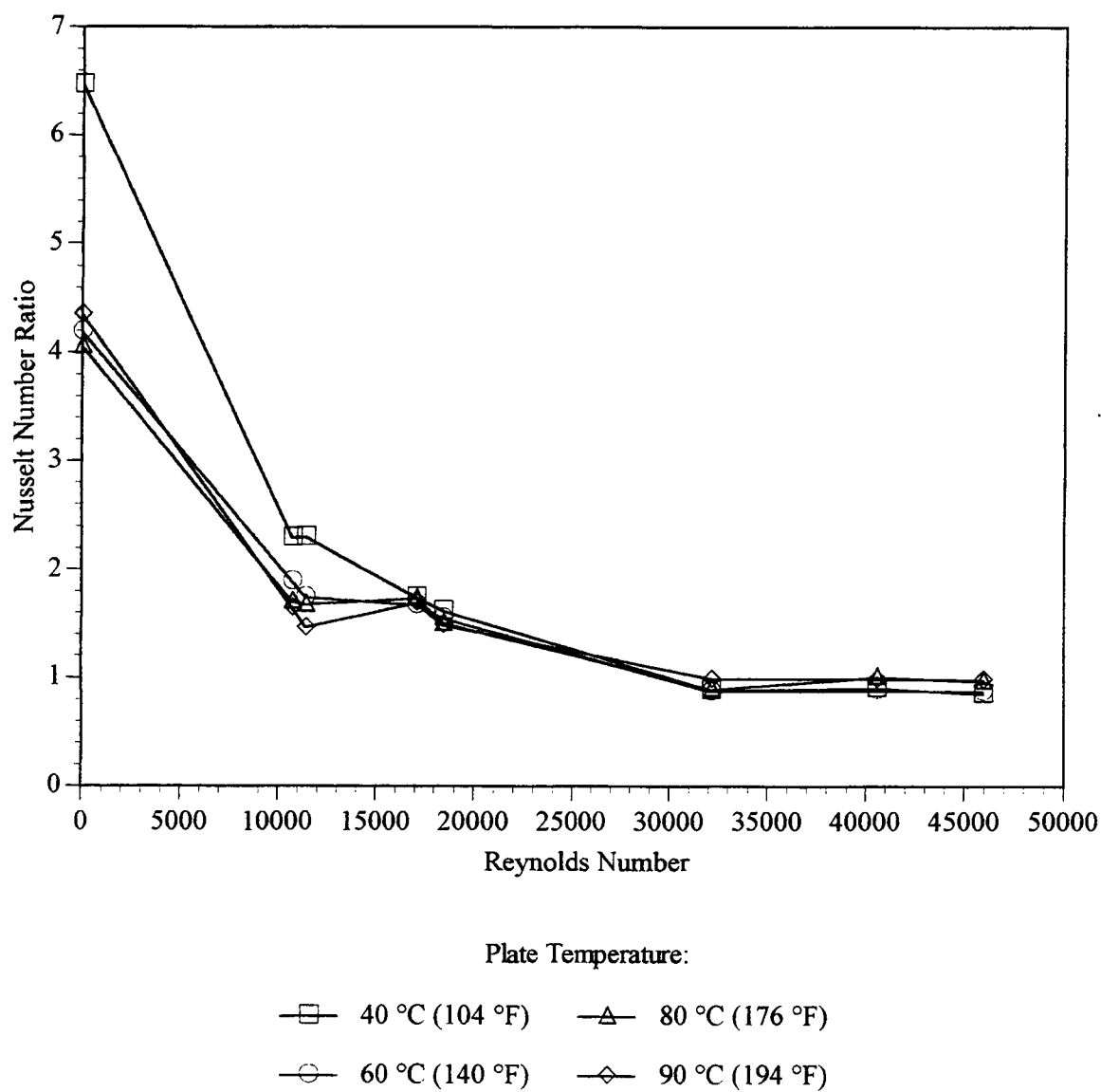


Figure 5.15: Nusselt Number Ratio for the Second Channel Width, Position 12.

Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at the Channel Center, Position 6
Channel Width, 43.69 cm (17.2 in)

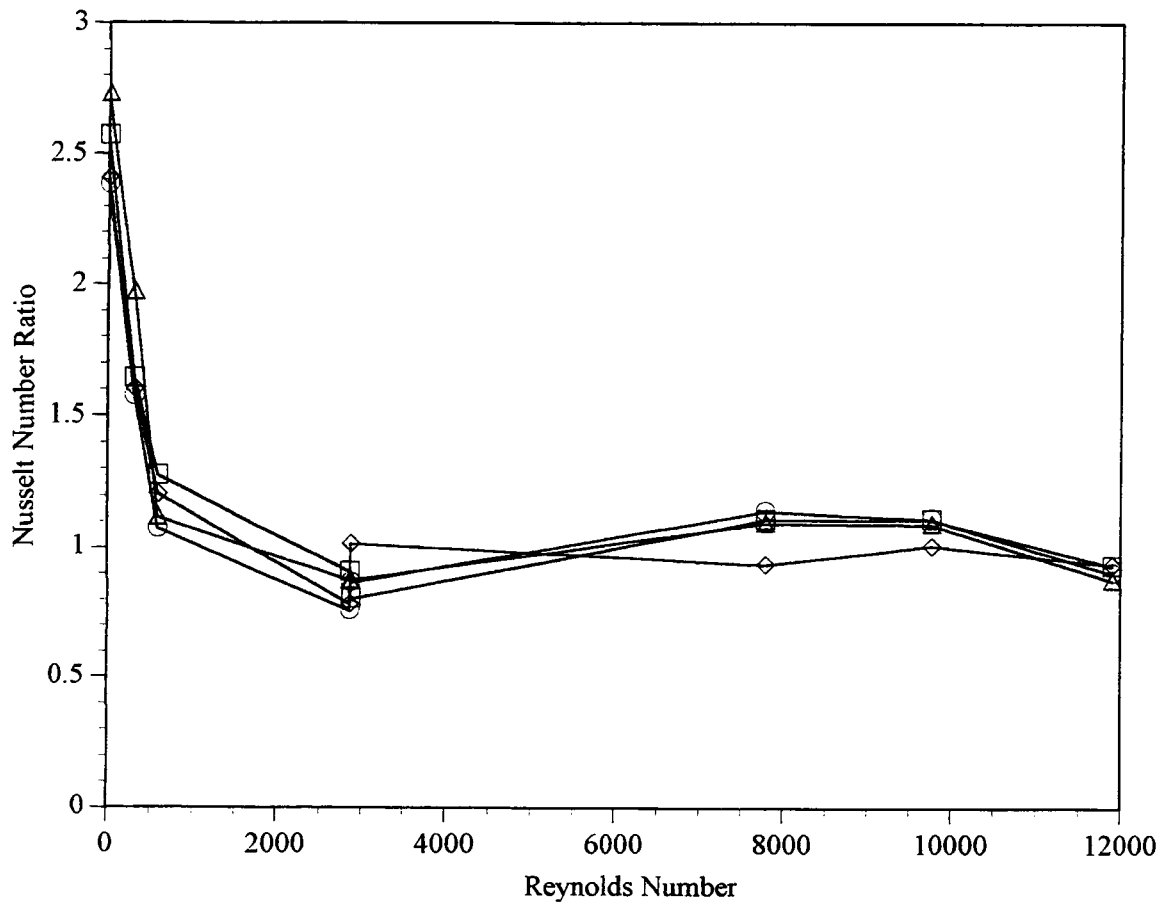


Plate Temperature:

- | | |
|--------------------|--------------------|
| —□— 40 °C (104 °F) | —△— 80 °C (176 °F) |
| —○— 60 °C (140 °F) | —◇— 90 °C (194 °F) |

Figure 5.16: Nusselt Number Ratio at the Center of the Third Channel Width.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 9; Channel Width, 43.70 (17.2 in)**

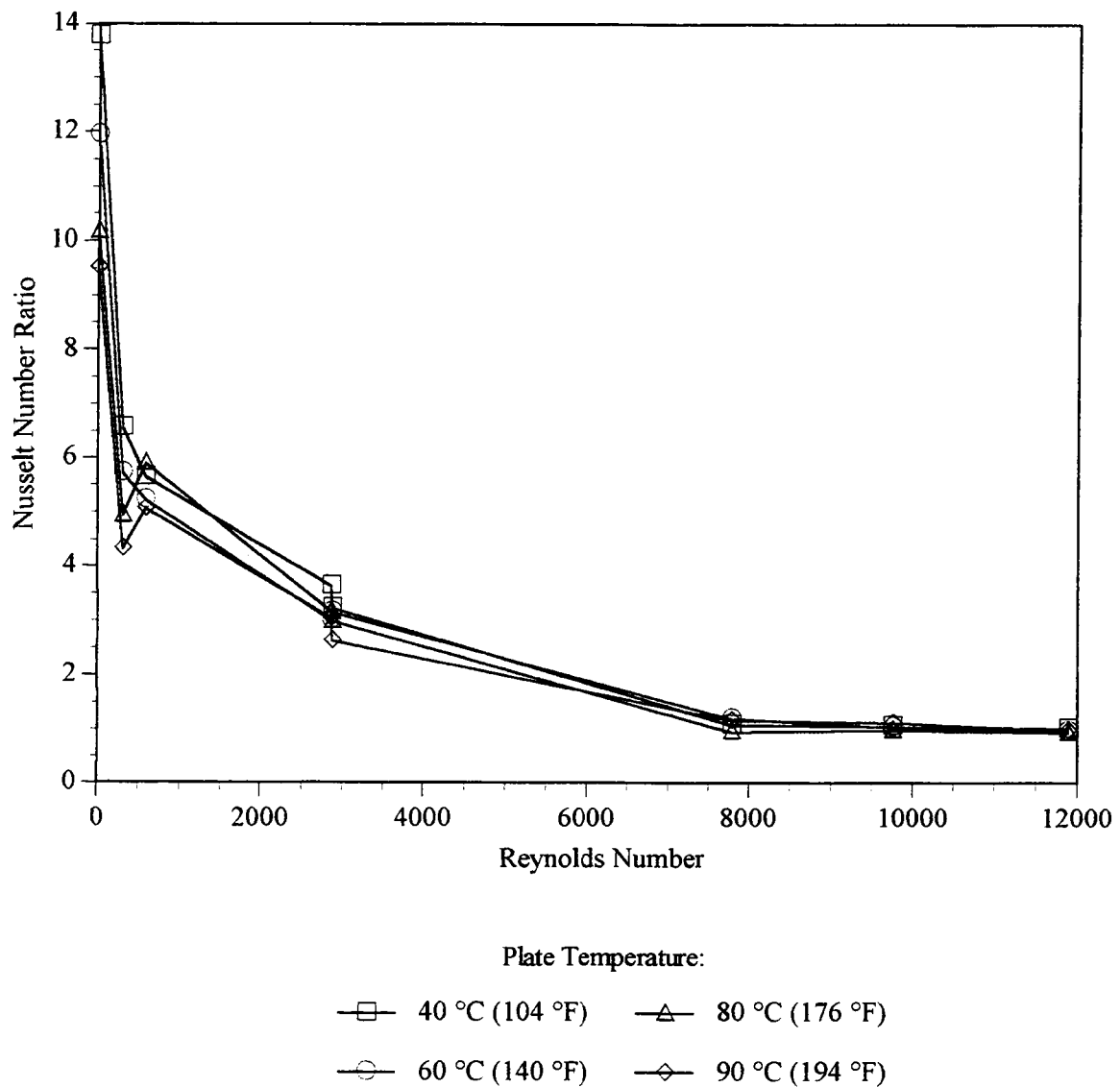


Figure 5.17: Nusselt Number Ratio for the Third Channel Width, Position 9.

Ratio of Nusselt Number with Infrasound to Nusselt Number without Infrasound at Position 3; Channel Width, 43.70 cm (17.2 in)

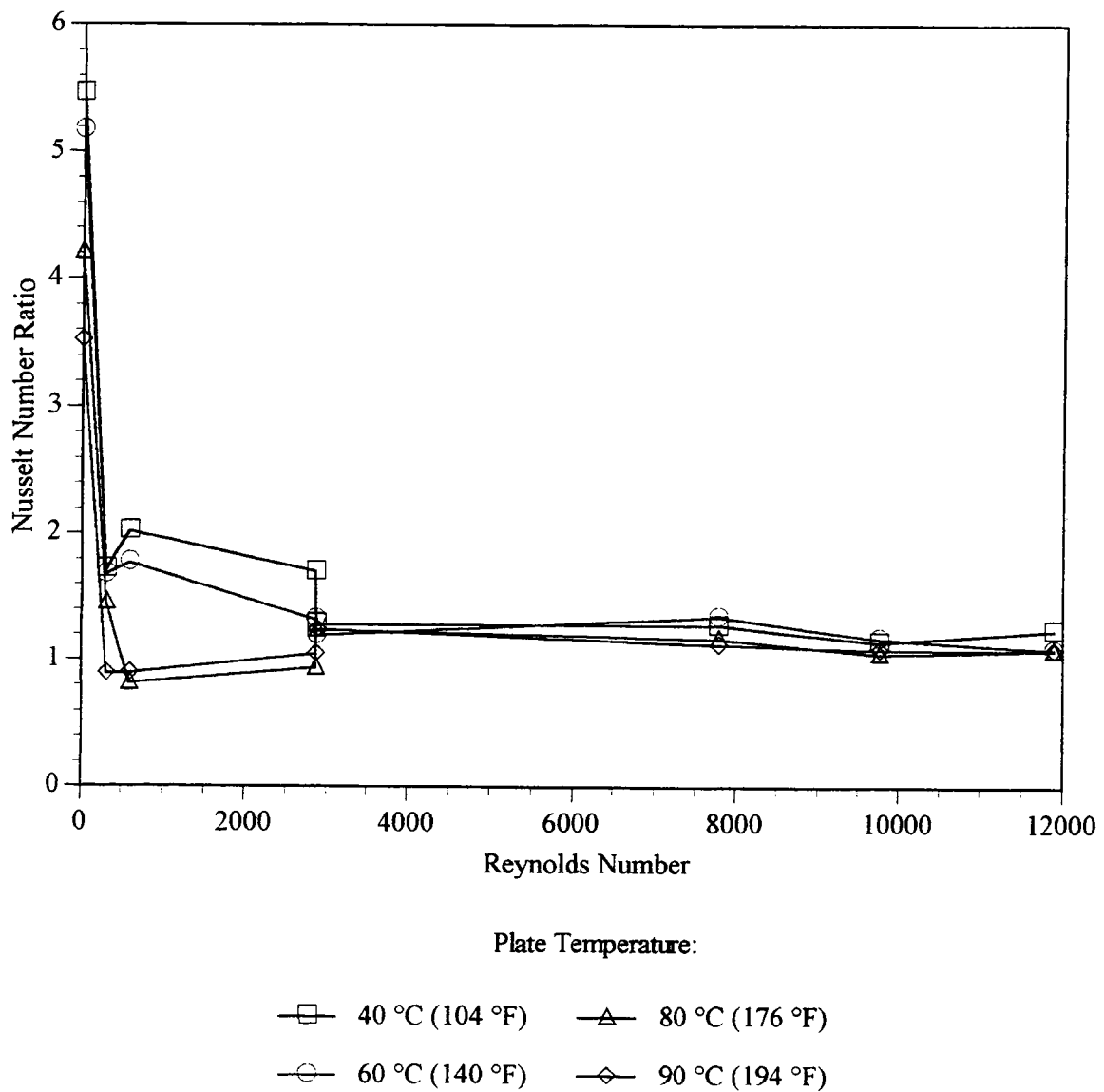


Figure 5.18: Nusselt Number Ratio for the Third Channel Width, Position 3.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 4; Channel Width, 43.70 cm (17.2 in)**

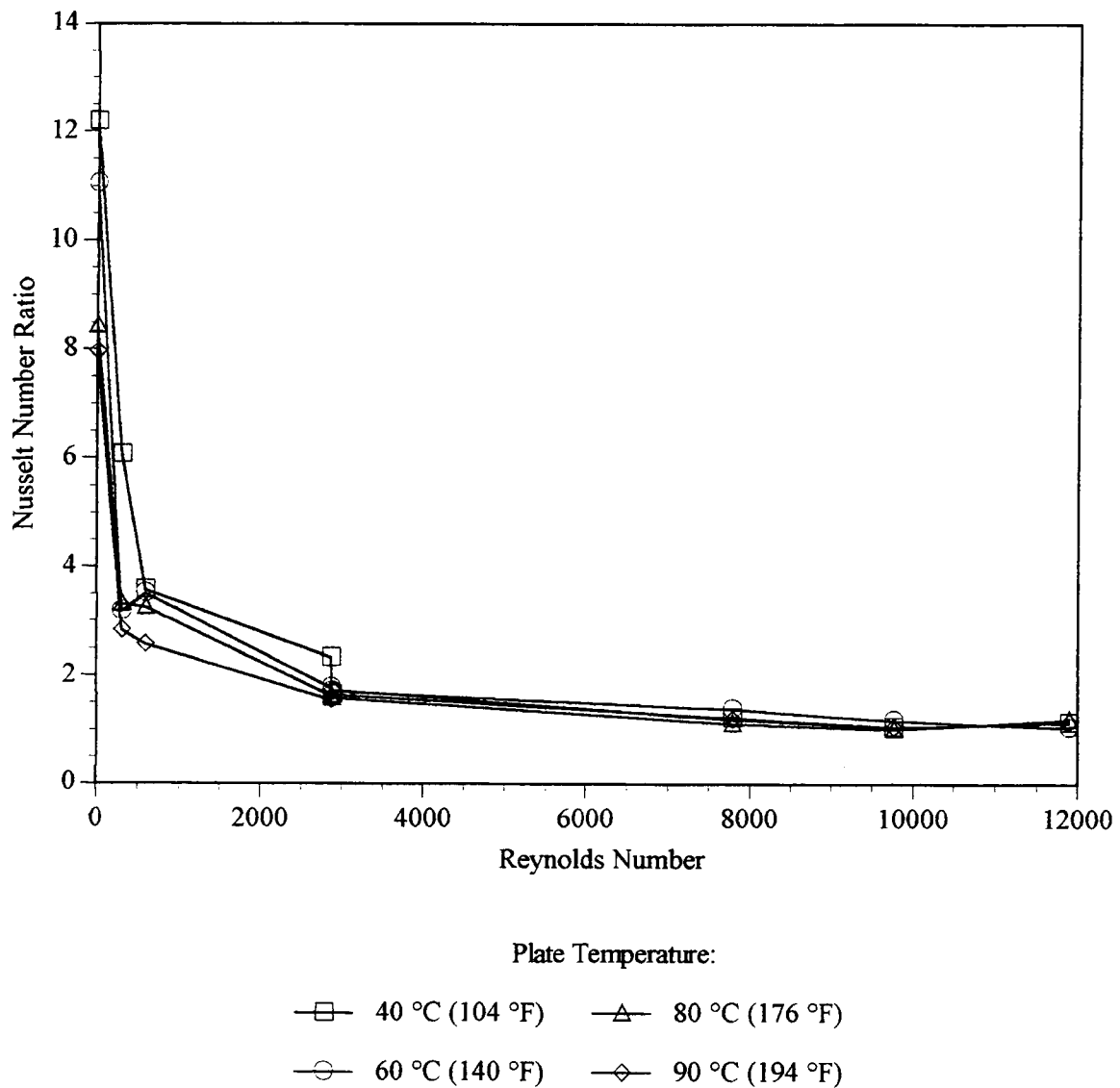


Figure 5.19: Nusselt Number Ratio for the Third Channel Width, Position 4.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 12; Channel Width, 43.70 cm (17.2 in)**

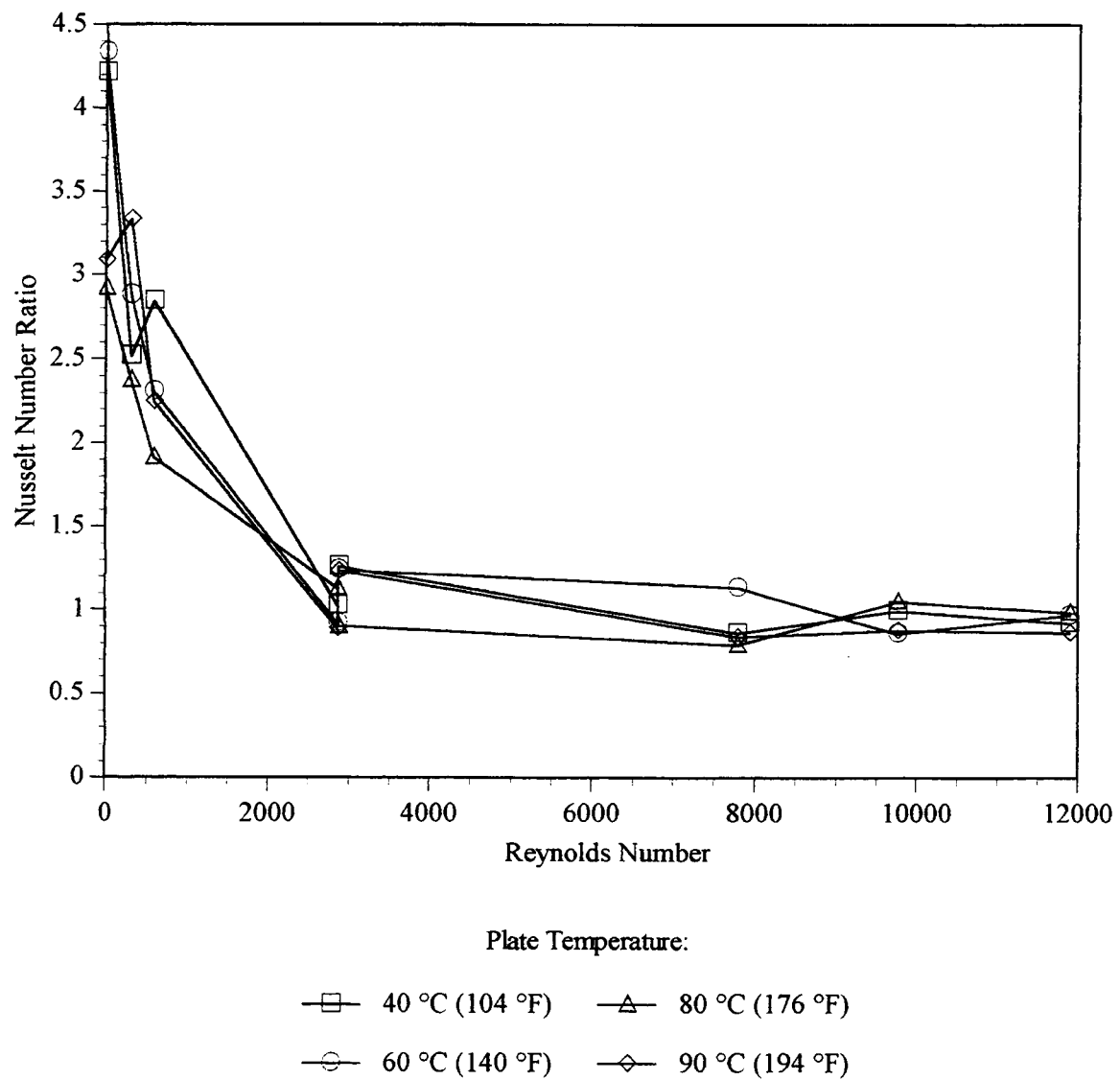


Figure 5.20: Nusselt Number Ratio for the Third Channel Width, Position 12.

**Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
at Position 13; Channel Width, 43.70 cm (17.2 in)**

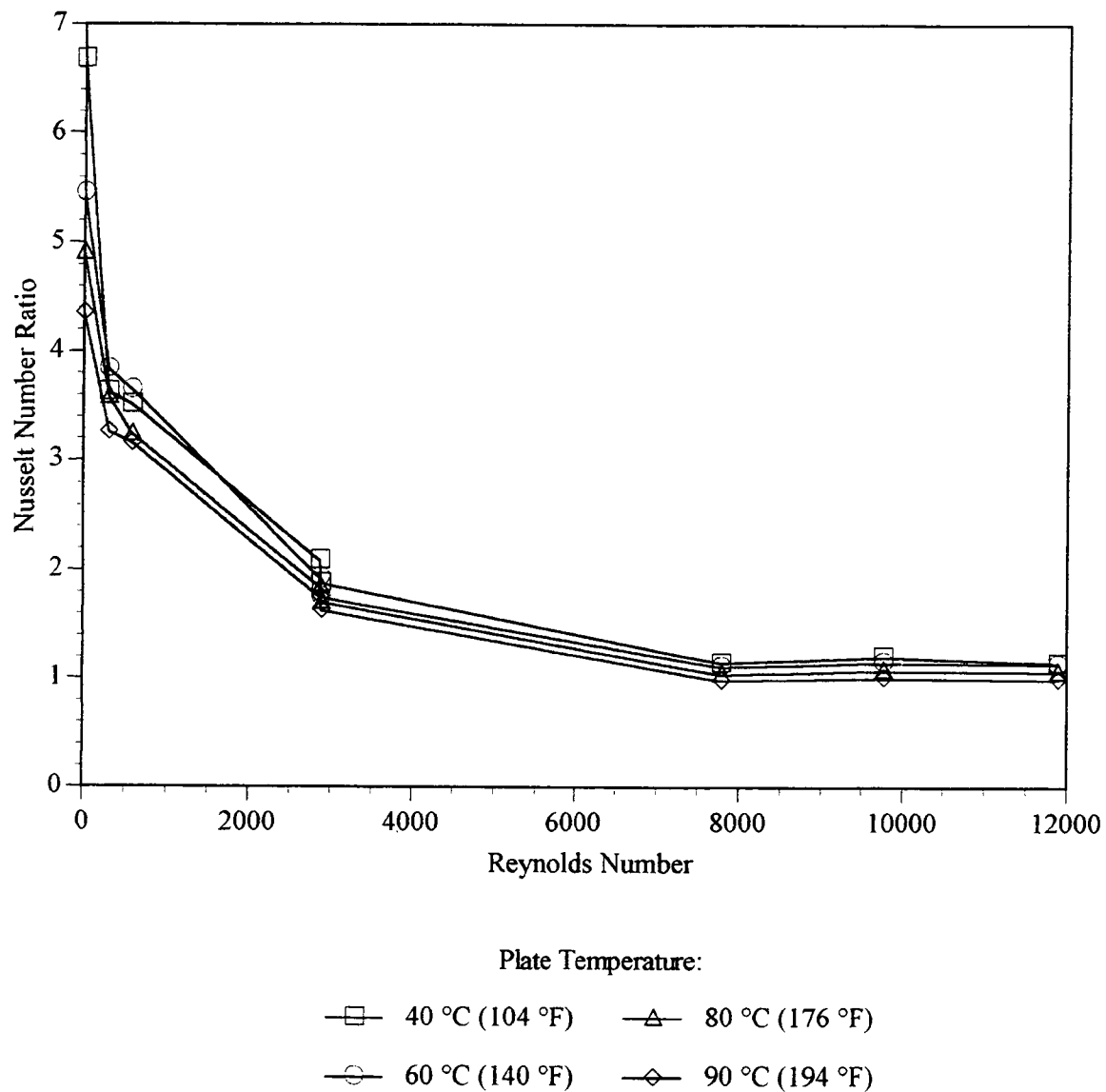


Figure 5.21: Nusselt Number Ratio for the Third Channel Width, Position 13.

Ratio of Nusselt Number with Infrasound to Nusselt Number without Infrasound Channel Width, 15.75 cm (6.2 in)

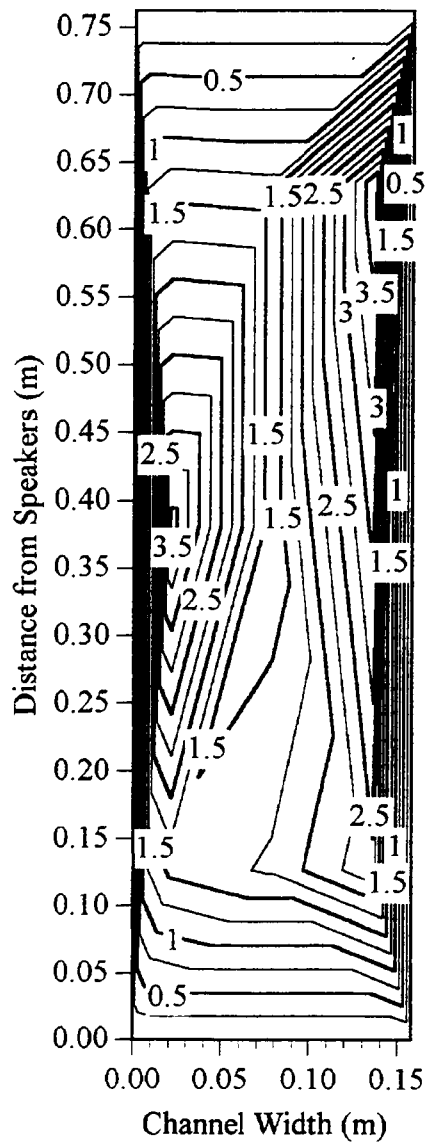


Plate Temperature, 90 °C (194 °F); Reynolds Number, 3085

Figure 5.22: Distribution of the Nusselt Number Ratio over the First Channel Width.

Distribution of the Nusselt Number with Infrasound over the First Channel Width, 15.75 cm (6.2 in)

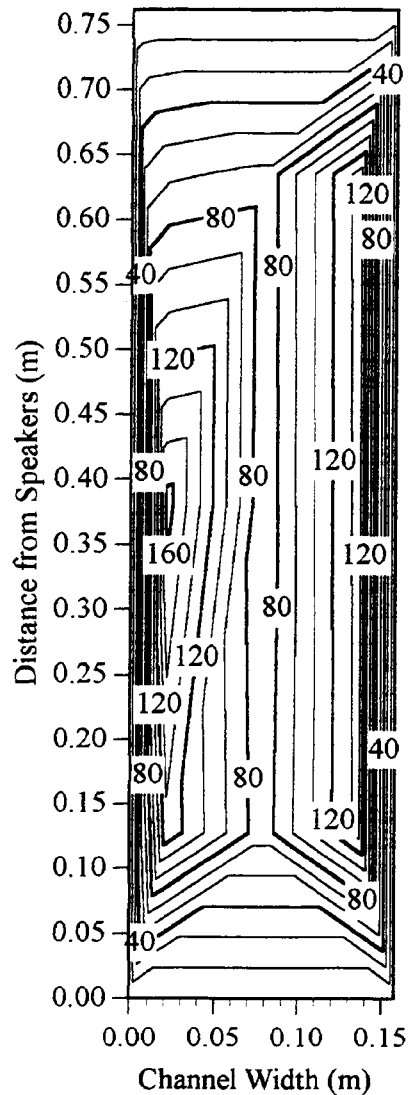


Plate Temperature, 90 °C (194 °F); Reynolds Number, 3085

Figure 5.23: Distribution of the Nusselt Number with Infrasound over the First Channel Width.

For the second channel width, 28.70 cm (11.3 in), the minimum local Nusselt number ratio was along the channel center line and above the channel entrance and exit, Figure 5.24. Similar to the first channel width, the maximum, local Nusselt number with infrasonic enhancement occurred along the channel walls, Figure 5.25. The largest values of the local Nusselt number ratio were experienced for the third channel width, 43.69 cm (17.2 in), with a distribution similar to the first two channel widths, Figure 5.26. The local Nusselt number with infrasonic enhancement was the largest along the channel center line, Figure 5.27.

Variation of the Average Nusselt Number

For the first channel width, 15.75 cm (6.2 in), as the Reynolds number approached 10,000 the average Nusselt number ratio approached unity, Figure 5.28. The Nusselt number ratio was not significantly affected by the plate temperature, the average standard deviation of the Nusselt number ratio as function of plate temperature was 0.103. An empirical relationship between the average Nusselt number ratio, Nu_{ratio} , and the Reynolds number, Re_D , was developed with the least squares regression of the base ten logarithm of the Nusselt number ratio and base ten logarithm of the Reynolds number, Equation 5.3:

$$Nu_{ratio} = 7.74 Re_D^{-0.189} \quad R^2 = 0.914 \quad (5.3)$$

The error bars of the best fit line reflect a $\pm 7.82\%$ uncertainty in the Nusselt number ratio for the first channel width, Appendix F.

Ratio of Nusselt Number with Infrasound to
Nusselt Number without Infrasound
Channel Width, 28.70 cm (11.3 in)

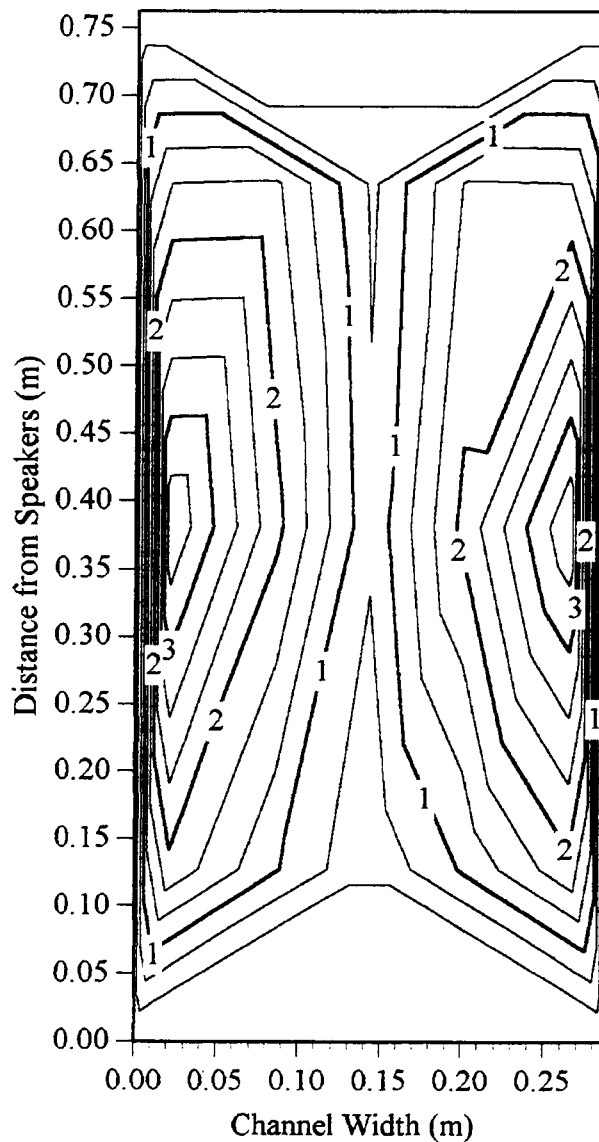


Plate Temperature, 80 °C (176 °F); Reynolds Number, 11400

Figure 5.24: Distribution of the Nusselt Number Ratio over the Second Channel Width.

Distribution of the Nusselt Number with Infrasound over the Second Channel Width, 28.70 cm (11.3 in)

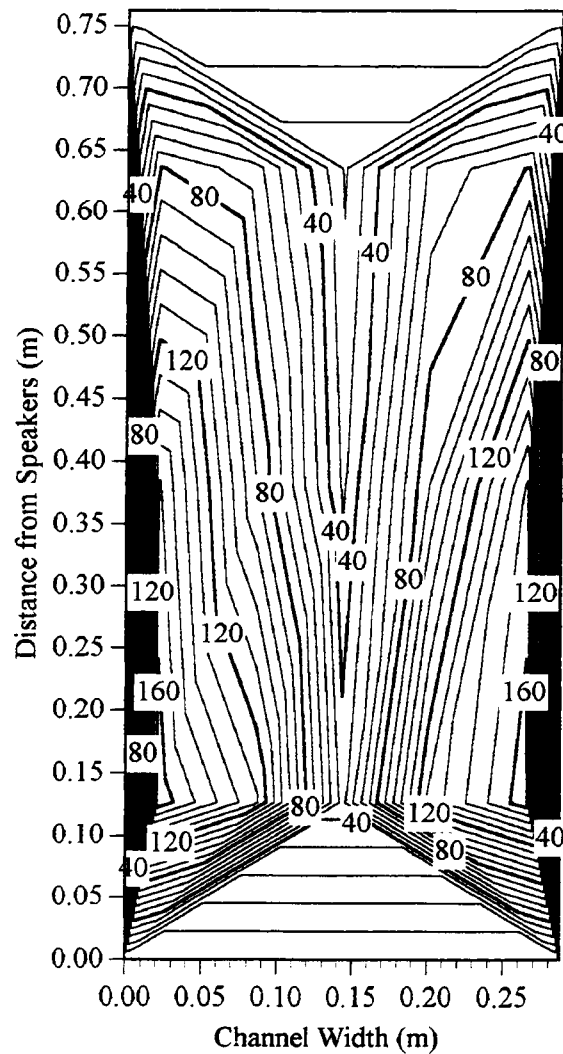


Plate Temperature, 80 °C (176 °F); Reynolds Number, 11400

Figure 5.25: Distribution of the Nusselt Number with Infrasound over the Second Channel Width.

Ratio of Nusselt Number with Infrasound to
Nusselt Number without Infrasound
Channel Width, 43.69 cm (17.2 in)

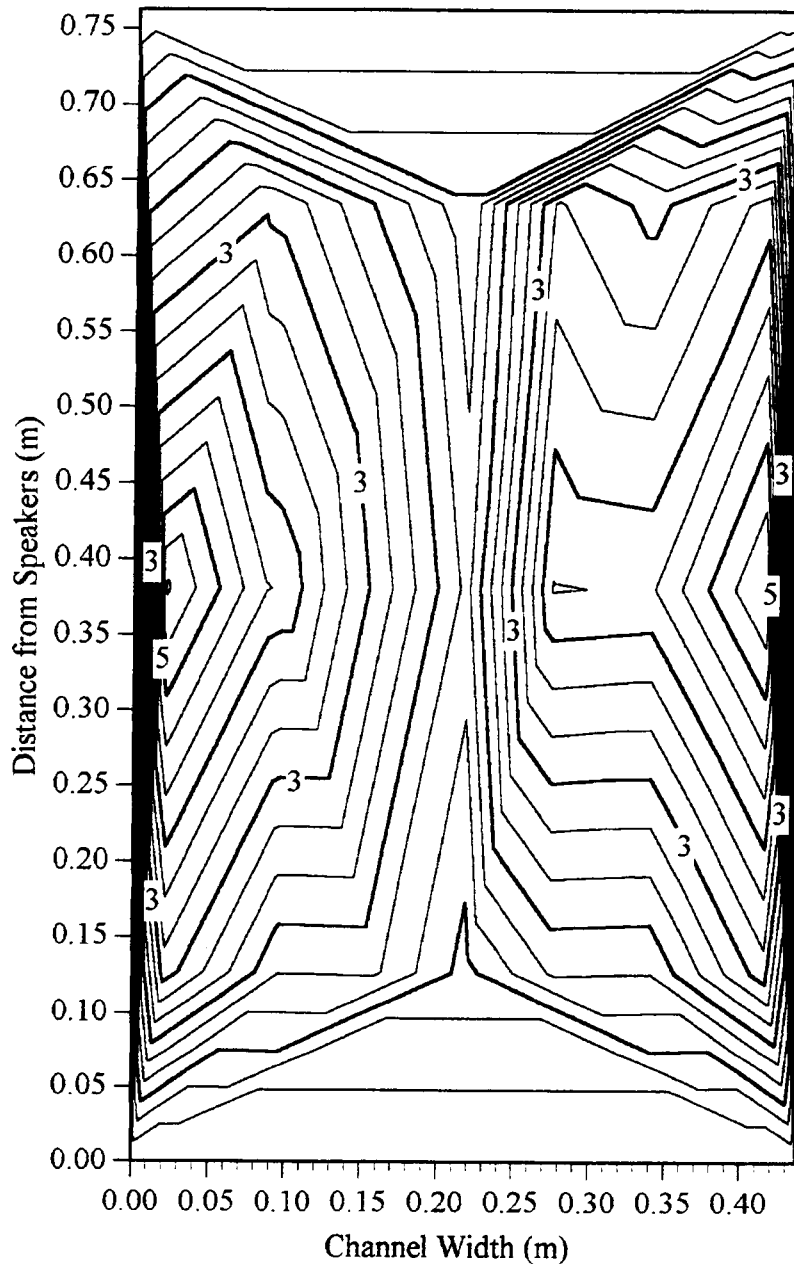


Plate Temperature, 60 °C (140 °F); Reynolds Number, 307

Figure 5.26: Distribution of the Nusselt Number Ratio over the Third Channel Width.

Distribution of the Nusselt Number with Infrasound over the Second Channel Width, 43.69 cm (17.2 in)

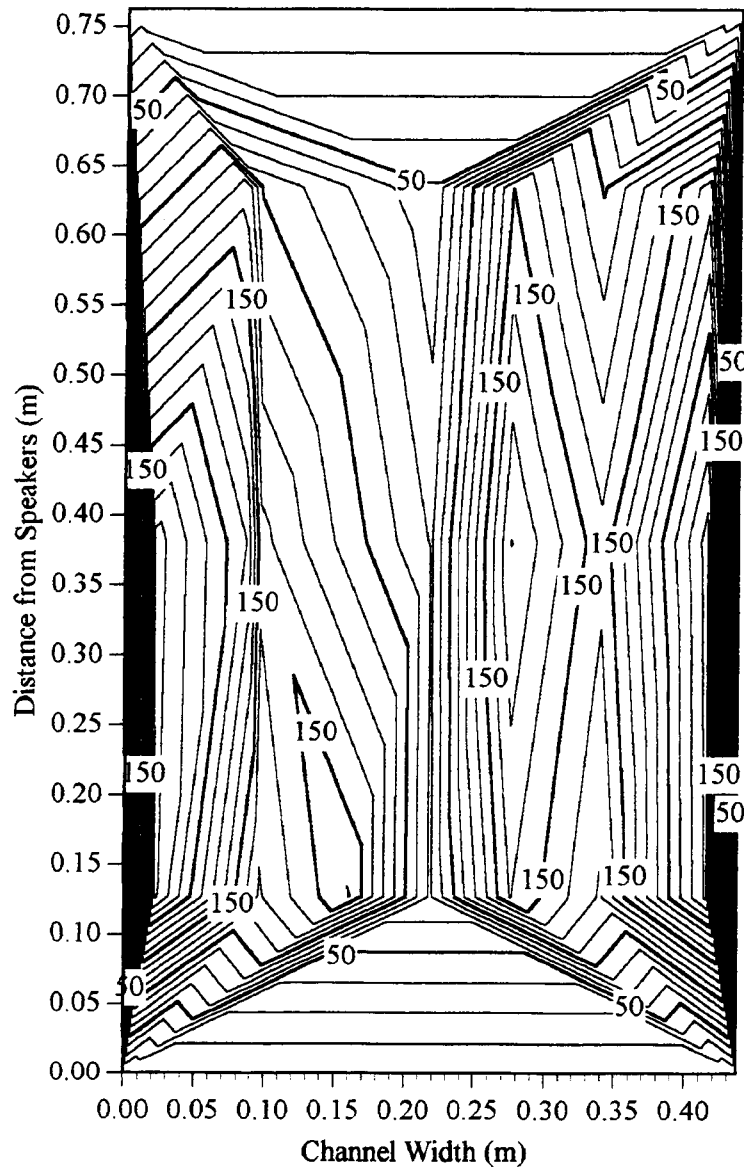


Plate Temperature, 60 °C (140 °F); Reynolds Number, 307

Figure 5.27: Distribution of the Nusselt Number with Infrasound over the Third Channel Width.

Average Ratio of Nusselt Number with Infrasound to Nusselt Number without Infrasound Channel Width, 15.75 cm (6.2 in)

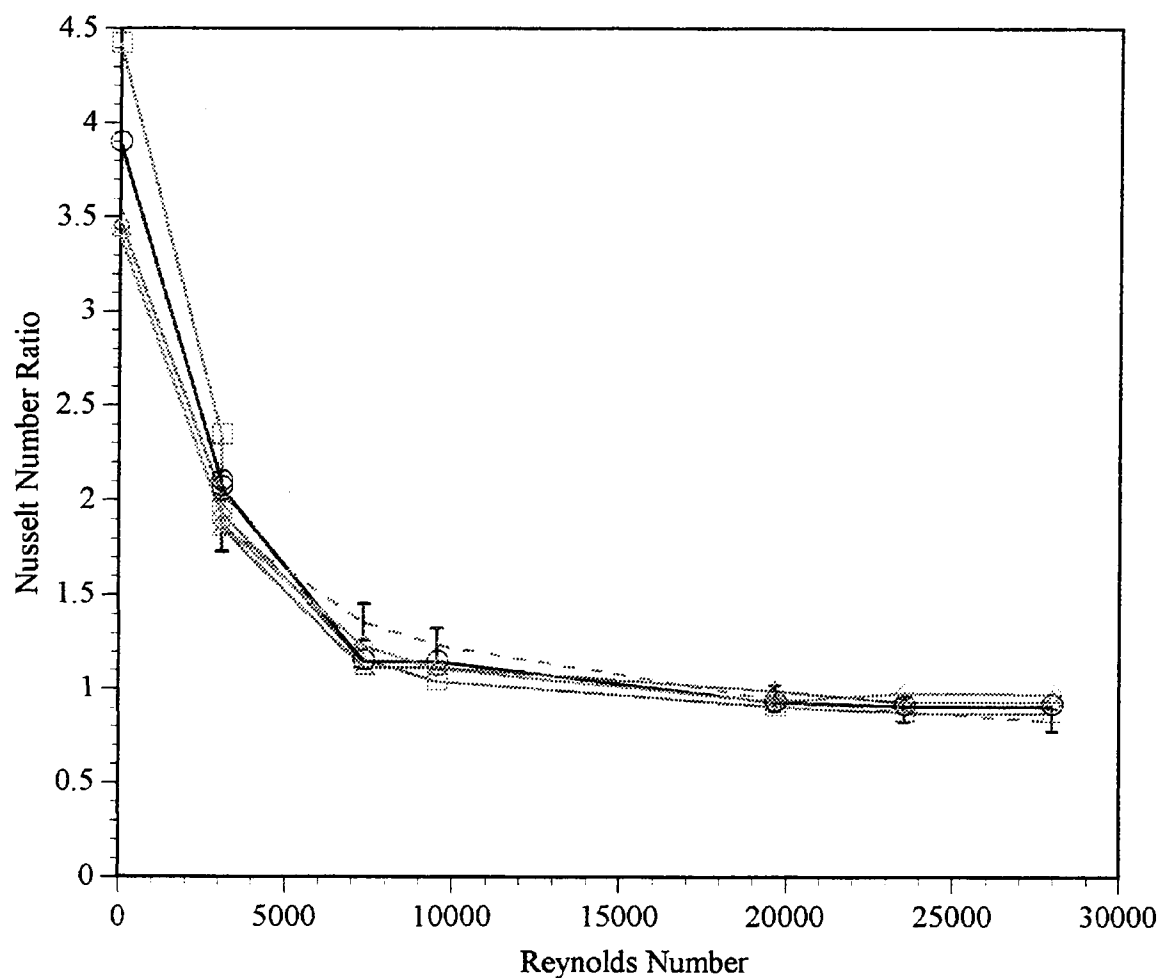


Plate Temperature:

40 °C
 60 °C
 80 °C
 90 °C
 - - - Best Fit

Empirical Relationship: $Nu_{Ratio} = 7.74 Re_D^{-0.189} \quad R^2 = 0.914$

Figure 5.28: Average Nusselt Number Ratio for the First Channel Width.

For the second channel width, 28.70 cm (11.3 in), the rate of heat transfer was increased by the infrasound for Reynolds numbers less than 30,000, Figure 5.29. Since the standard deviation of the Nusselt number ratio as function of plate temperature increased slightly, 0.119, the empirical relationship, Equation 5.4, was significantly less accurate and the uncertainty of the Nusselt number ratio for the second channel width was $\pm 8.47\%$.

$$Nu_{ratio} = 81.63 Re_D^{-0.442} \quad R^2 = 0.746 \quad (5.4)$$

For the third channel width, 43.69 cm (17.2 in), the average Nusselt number ratio was greater than one for Reynolds numbers less than 10,000, Figure 5.30. Even though the standard deviation of the Nusselt number ratio as function of temperature increased to 0.261 and the uncertainty of the Nusselt number ratio for the third channel width was $\pm 8.40\%$, an accurate correlation between the Nusselt number ratio and the Reynolds number, Equation 5.5, was obtained.

$$Nu_{ratio} = 17.83 Re_D^{-0.304} \quad R^2 = 0.993 \quad (5.5)$$

Observing the average Nusselt number ratio as function of the Reynolds number and the channel width at a fixed plate temperature, the Nusselt number ratio was the largest for the second channel width, Figures 5.31 - 5.34. This is most likely directly related to the air velocity distribution within the second channel width. Since the Nusselt number ratio in the second channel width for Reynolds numbers between 0 and 10,000 was not measured because of limitations of the blower, it can not be

Average Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
Channel Width, 28.70 cm (11.3 in)

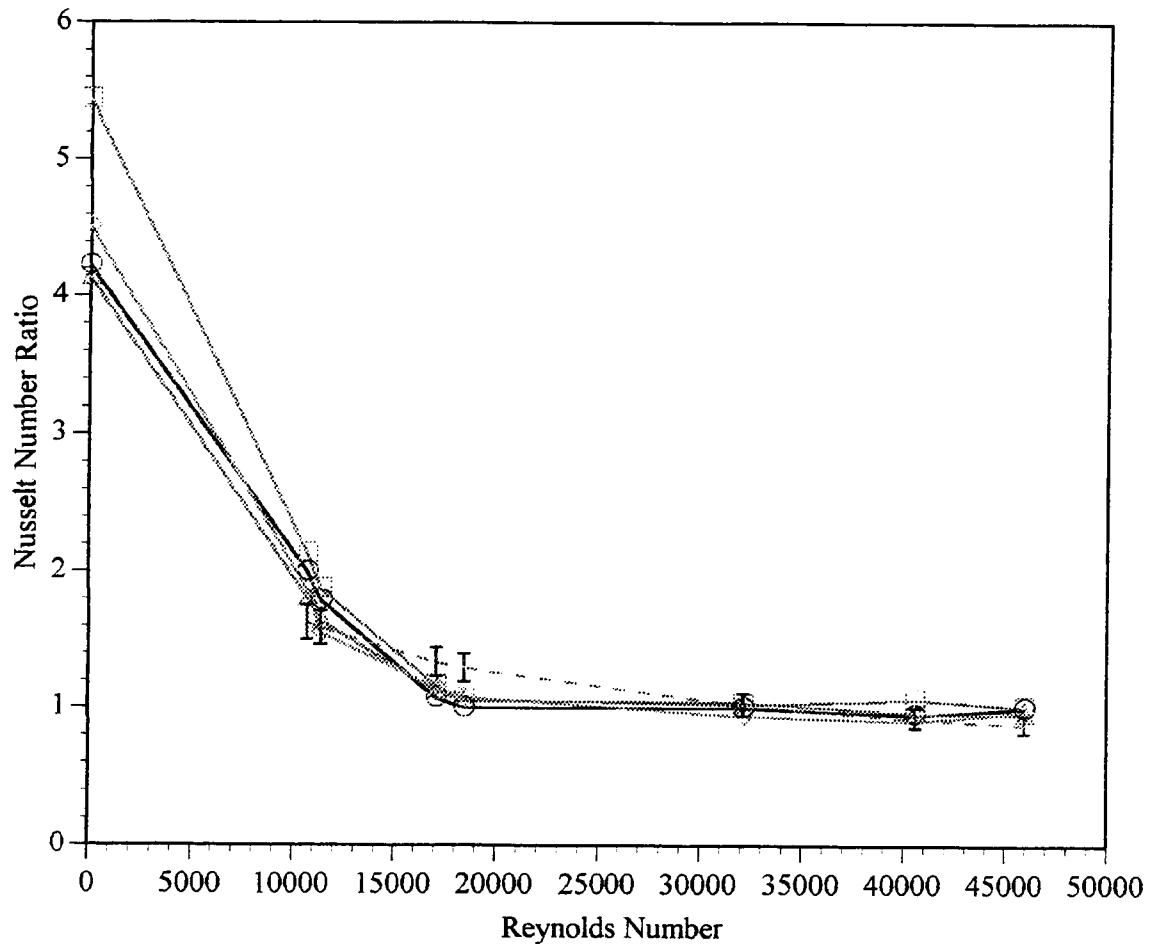


Plate Temperature:

—○— 40 °C
—●— 60 °C
—△— 80 °C
—◇— 90 °C
- - - Best Fit

Empirical Relationship: $Nu_{Ratio} = 81.63 Re_D^{-0.442} \quad R^2 = 0.746$

Figure 5.29: Average Nusselt Number Ratio for the Second Channel Width.

ascertained if the second channel width was an optimum width for acoustic enhancement.

The average Nusselt number ratio exhibited a positive relationship with the sound pressure level inside the channel, Figure 5.35, but did not appear sensitive to the frequency of the infrasound over the frequency range examined, resonant frequency ± 2.75 Hz, Figure 5.36.

Variation of Average Heat Transfer Coefficient

When compared to the average Nusselt number with infrasonic enhancement as a function Reynolds number, the average heat transfer coefficient with infrasonic enhancement exhibited similar trends. The average heat transfer coefficient was not significantly effected by the temperature difference between the copper plate and the air. The average velocity of the air inside the channel had the largest influence on the average heat transfer coefficient; the channel width had only a moderate effect, as shown in Tables 5.1 - 5.3.

Average Ratio of Nusselt Number with Infrasound to Nusselt Number without Infrasound Channel Width, 43.69 cm (17.2 in)

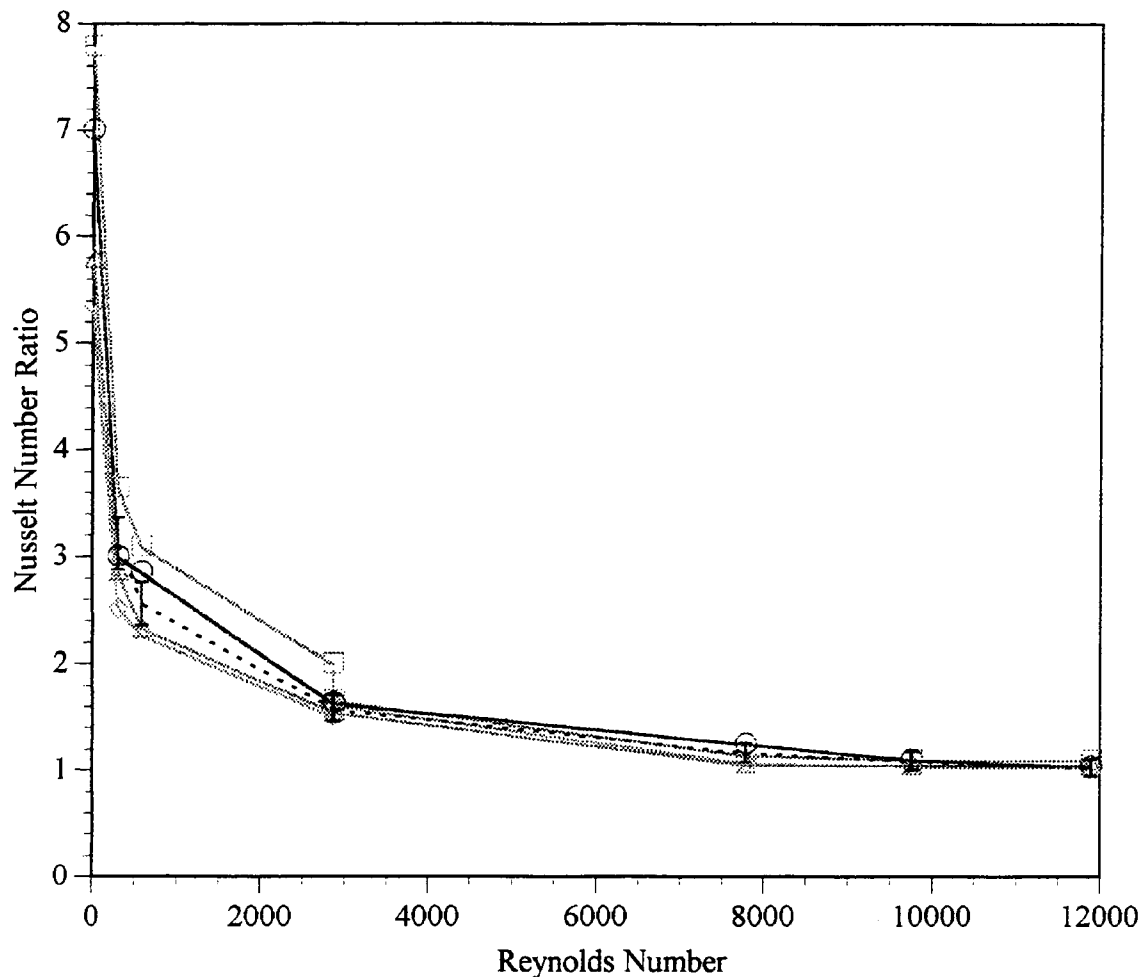


Plate Temperature:

—•— 40 °C
—○— 60 °C
- - - 80 °C
- . - . 90 °C
..... Best Fit

$$\text{Empirical Relationship: } Nu_{\text{Ratio}} = 17.83 Re_D^{-0.304} \quad R^2 = 0.993$$

Figure 5.30: Average Nusselt Number Ratio for the Third Channel Width.

**Average Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
Plate Temperature, 40 °C (104 °F)**

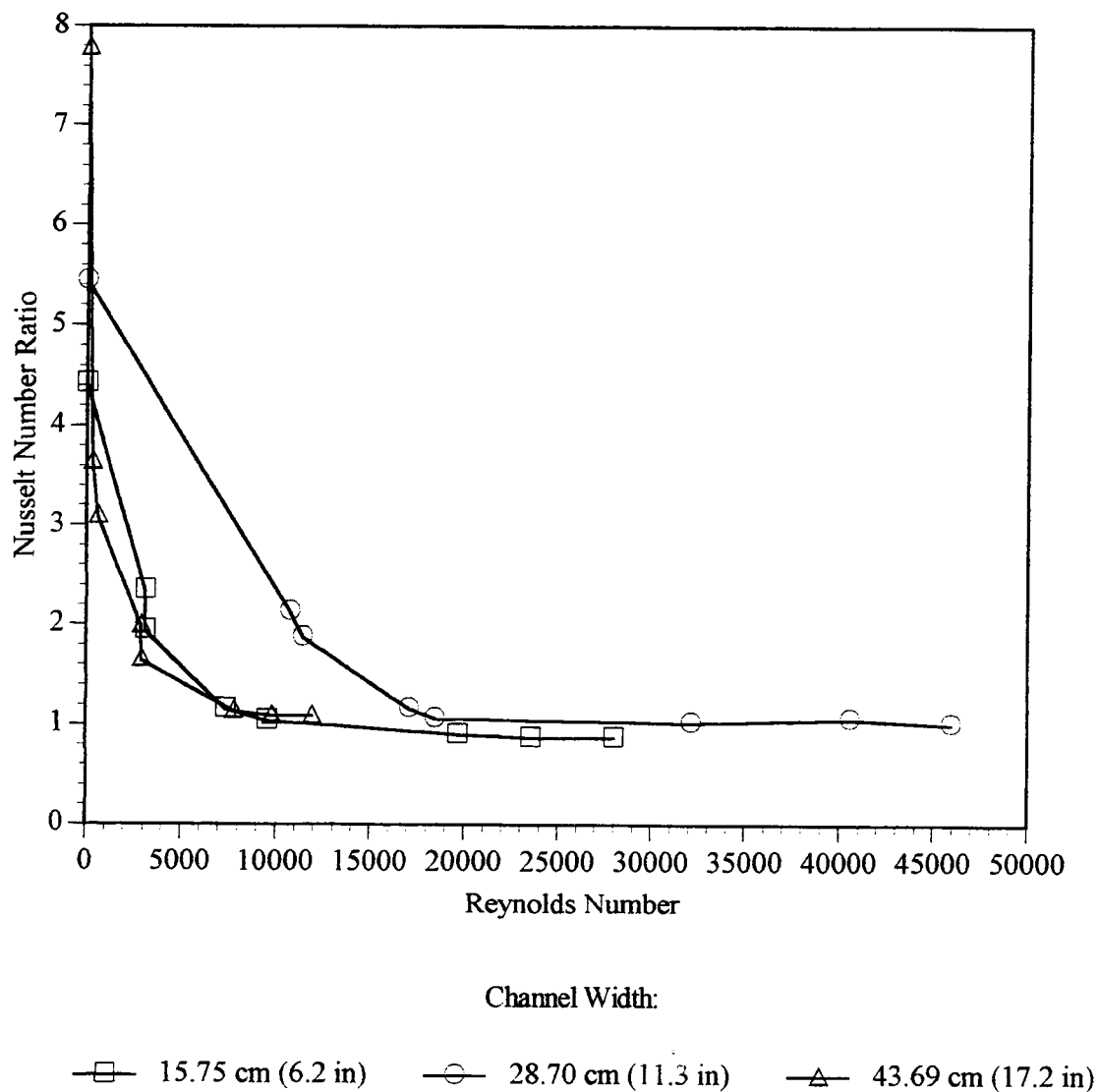


Figure 5.31: Average Nusselt Number Ratio with the Plate Temperature at 40 °C.

Average Ratio of Nusselt Number with Infrasound to Nusselt Number without Infrasound Plate Temperature, 60 °C (140 °F)

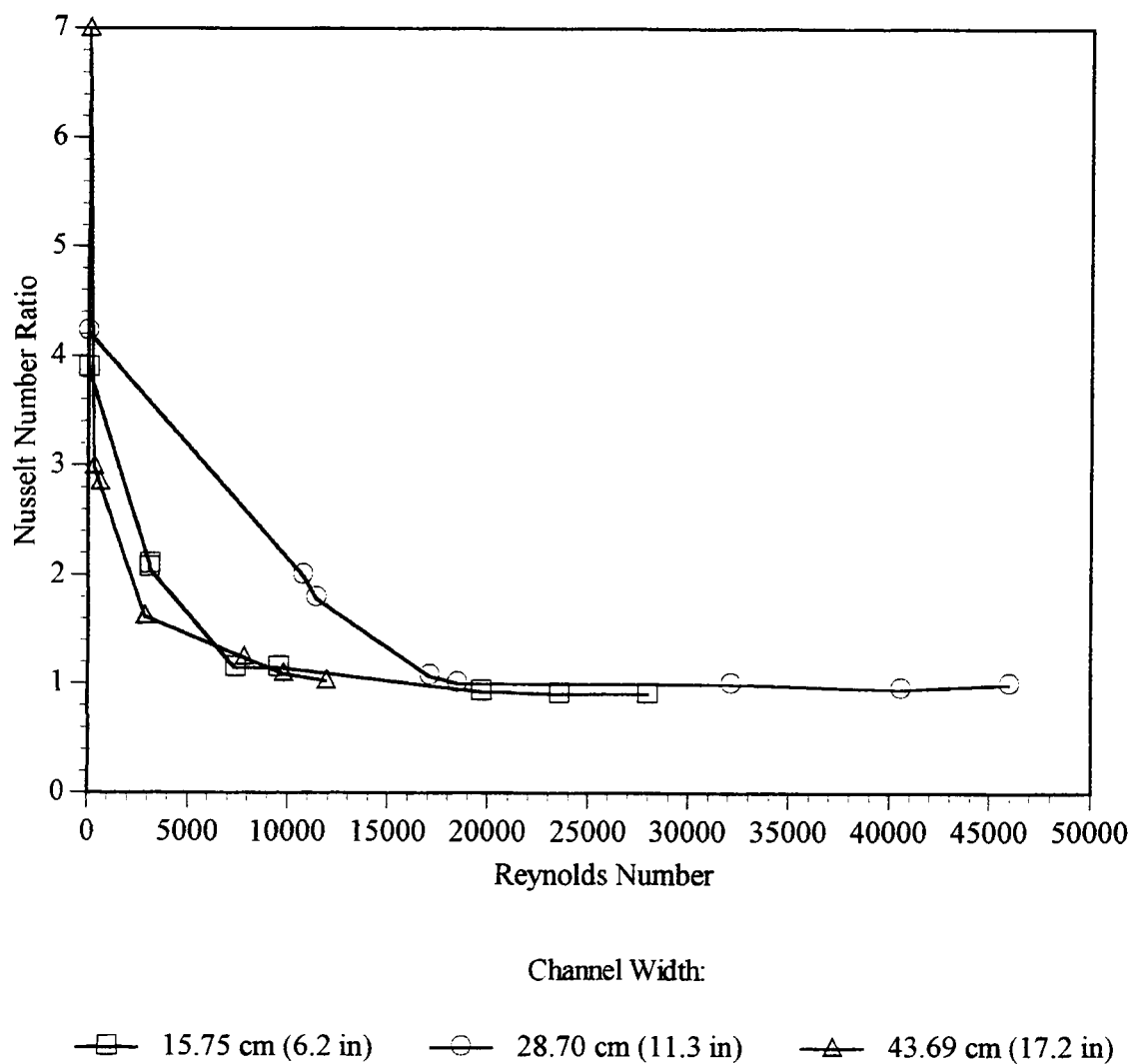


Figure 5.32: Average Nusselt Number Ratio with the Plate Temperature at 60 °C.

Average Ratio of Nusselt Number with Infrasound to Nusselt Number without Infrasound Plate Temperature, 80 °C (176 °F)

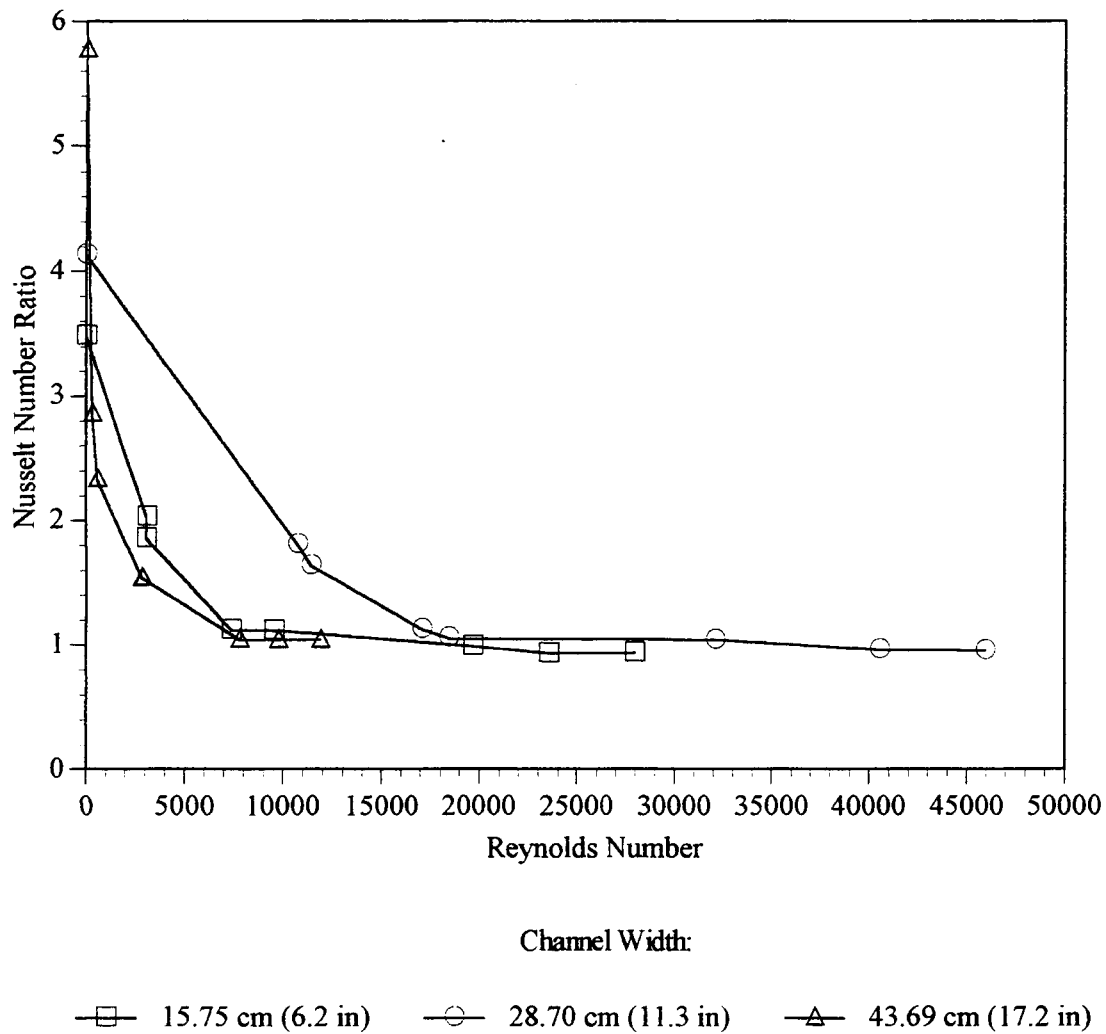


Figure 5.33: Average Nusselt Number Ratio with the Plate Temperature at 80 °C.

Average Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
Plate Temperature, 90 °C (194 °F)

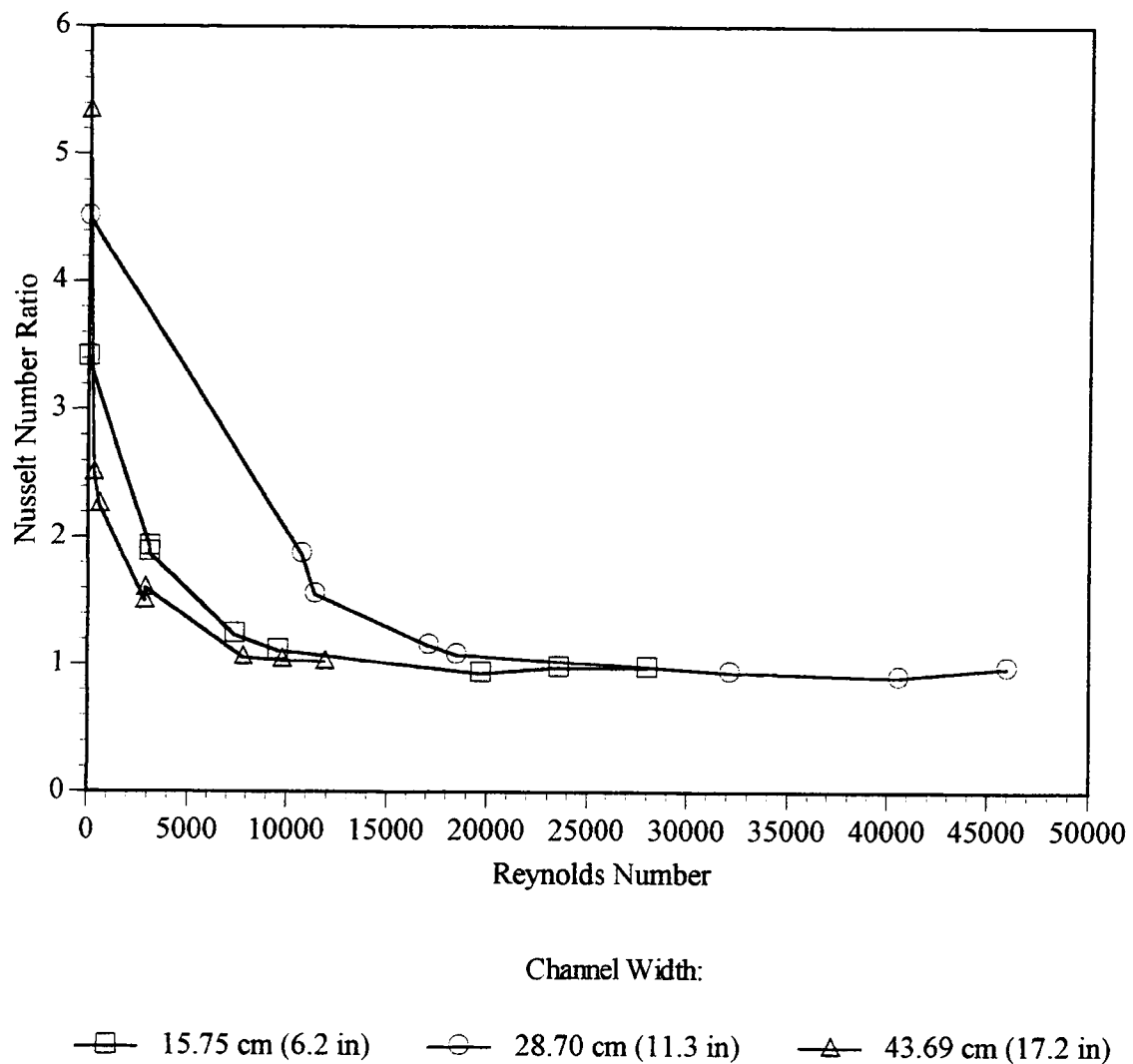


Figure 5.34: Average Nusselt Number Ratio with the Plate Temperature at 90 °C.

**Average Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
as a Function of Sound Pressure Level
Channel Width, 28.70 cm (11.3 in)**

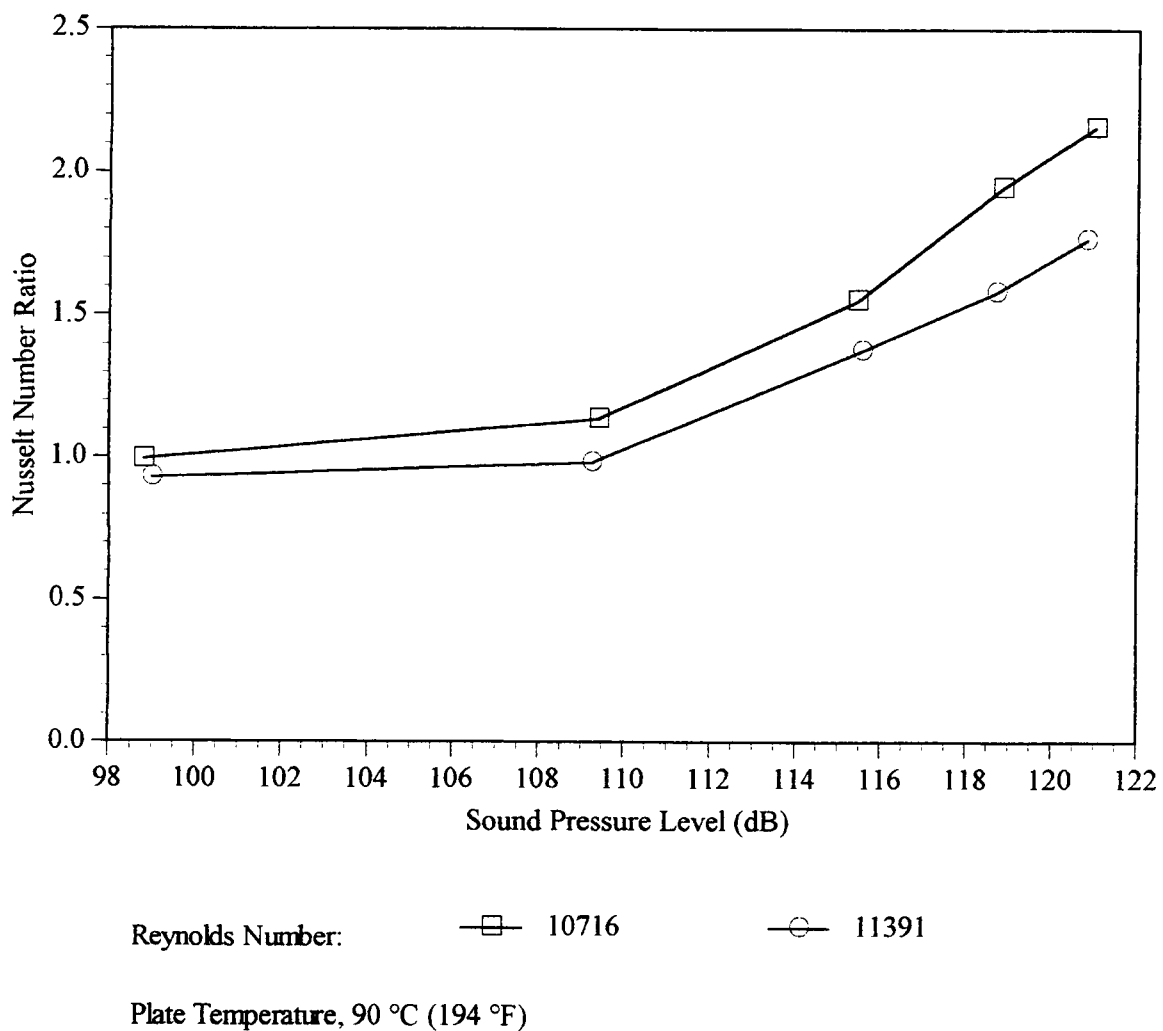
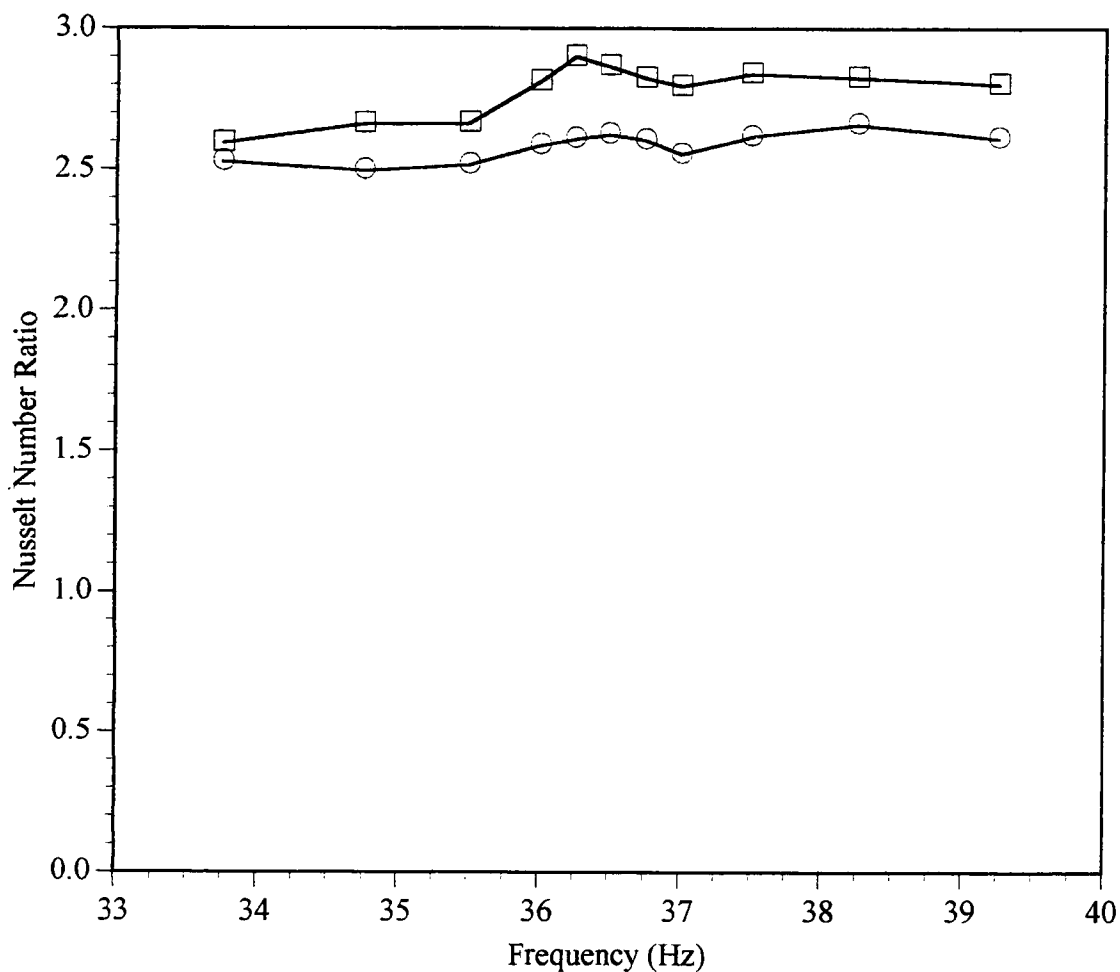


Figure 5.35: Average Nusselt Number Ratio as a Function of Sound Pressure Level.

Average Ratio of Nusselt Number with Infrasound
to Nusselt Number without Infrasound
as a Function of Input Sound Frequency
Channel Width, 15.75 cm (6.2 in)



Reynolds Number:

—□— 3085

—○— 3075

Plate Temperature, 90 °C (194 °F)

Figure 5.36: Average Nusselt Number Ratio as a Function of Sound Frequency.

Table 5.1: Average Heat Transfer Coefficients with Infrasond, First Channel Width.

Average Heat Transfer Coefficients (W/m ² ·K), with Infrasond, Channel Width 15.75 cm				
Reynolds Number	Plate Temperature			
	40 °C	60 °C	80 °C	90 °C
0	25.925	24.387	23.164	24.037
3085	34.008	33.413	33.013	33.402
3075	33.791	34.497	32.647	33.386
7343	32.069	31.847	31.447	33.091
9527	31.802	32.408	33.702	34.247
19666	42.551	44.412	45.384	46.262
23539	47.618	49.906	50.789	52.300
27975	50.401	51.895	53.334	56.050

Table 5.2: Average Heat Transfer Coefficients with Infrasond, Second Channel Width.

Average Heat Transfer Coefficients (W/m ² ·K), with Infrasond, Channel Width 28.70 cm				
Reynolds Number	Plate Temperature			
	40 °C	60 °C	80 °C	90 °C
0	33.019	27.925	29.814	29.836
10716	30.041	26.136	26.059	27.681
11391	29.225	27.797	26.836	26.414
17088	31.919	31.097	31.264	31.569
18465	32.069	31.280	33.536	32.525
32137	39.063	38.357	41.163	40.529
40551	46.612	44.468	45.246	43.996
45961	48.229	48.279	47.862	50.089

Table 5.3: Average Heat Transfer Coefficients with Infrasonud, Third Channel Width.

Average Heat Transfer Coefficients ($\text{W/m}^2\cdot\text{K}$), with Infrasonud, Channel Width 43.69 cm				
Reynolds Number	Plate Temperature			
	40 °C	60 °C	80 °C	90 °C
0	41.318	41.551	37.863	37.524
307	39.135	38.163	35.080	30.814
587	40.585	38.230	32.219	33.252
2863	34.430	33.486	34.924	32.886
2876	35.163	35.363	35.002	35.480
7785	36.946	39.341	37.185	37.774
9758	42.868	42.229	42.890	43.179
11894	43.057	42.529	43.951	43.857

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

Potential for Infrasonic Enhancement of the Rate of Heat Transfer

The rate of heat transfer over a flat plate can be increased by the presence of infrasound. Compared to natural convection, infrasound can increase the rate of heat transfer by approximately an order of magnitude; however, compared to forced convection, the effects of infrasound decrease with increasing air velocity. The primary infrasonic enhancement is concentrated in the laminar flow regime, $Re_D \leq 2300$. As the flow rate is increased, the effects of the standing, sound wave are overpowered. The use of infrasonic enhancement inside aluminum ingot furnaces would require the air velocity to be reduced approximately four orders of magnitude. This reduction of air velocity would reduce the rate of heat transfer approximately two orders of magnitude, which the infrasonic enhancement could not overcome. Based on the level of acoustic energy and air flow rates examined in this study, infrasonic enhancement of rate of heat transfer inside aluminum ingot furnace does not appear practical at this time.

Conclusions

The rate of heat transfer over a flat plate was increased by the presence of a standing sound wave in a channel. Infrasonic effects are primarily a function of the sound pressure level, air velocity inside the channel and the channel width, which can be combined into a single parameter, Reynolds number based on the hydraulic diameter of the channel. An empirical relationship between the ratio of the heat transfer with infrasound to the heat transfer without infrasound, using a Nusselt number ratio, and the Reynolds number; $Nu_{Ratio} = a Re_D^b$ was developed. The thermophysical properties of the heat transfer fluid are embedded in the constant, a . Relative to the rate of heat transfer the sensitivity of experimental apparatus to the frequency of infrasound was insignificant.

For all three channel widths the rate of heat transfer with infrasound was greater than rate of heat transfer without infrasound for Reynolds numbers less than 10,000. In general, the largest increases in the rate of heat transfer occurred at zero air velocity. In a few situations, the maximum increase in the rate of heat transfer occurred for non-zero Reynolds numbers, based on an average air velocity inside the channel. Each of these situations was observed near the channel wall suggesting infrasonic effects were more significant at lower air velocities, as expected. The largest, single increase in the rate of heat transfer occurred at zero air velocity for the third channel width, 43.69 cm (17.2 in). The second channel width, 28.70 cm (11.3 in), experienced the largest increase in the rate of heat transfer for all Reynolds

numbers. These conclusions may be the result of the presence of a stagnation region caused by the air flow entering the channel perpendicular to the flow direction.

Recommendations

The Nusselt number ratio in the second channel width, 28.70 cm (11.3 in), should be measured for Reynolds numbers less than 10,000. Additionally, the rate of heat transfer for all channel widths should be measured at fixed Reynolds numbers. The rate of heat transfer for additional channel widths between 15.75 cm (6.2 in) and 43.79 cm (17.2 in) should be measured to determine the effect of channel width on acoustic enhancement.

An attempt to visualize the flow structure in the channel should be made to determine what type of flow structures exist near the channel walls. If the flow structures near the wall appear to have a significant impact on the rate of heat transfer, the possibility of introducing the air flow through the channel walls or the resonance chamber and speaker box should be investigated. This arrangement may reduce the asymmetric results experienced in the first channel width. Additionally, a flow visualization study should seek to determine if a transition to turbulent flow occurs within the channel, the extent entrance region effects on the results and the extent of flow separation near the entrance or exit to the channel.

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LIST OF REFERENCES

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APPENDICES

APPENDIX A

Equipment List

Table A.1: Equipment List

Equipment	Manufacture	Model Number	Serial Number	UT ID Number
Constant Temperature Bath	Haake	NK 22	N/A	361639
Data Acquisition / Control Unit	Hewlett Packard	3497A	N/A	368049
Function Generator	Gold Star	FG - 2002C	201106	N/A
Hughes Blower	Rotron Manufacturing Co.	929700 - 5, #AO-16555	232157	N/A
Oscilloscope	Tektronix	5103W	N/A	257641
Dual Channel Signal Analyzer	Brüel & Kjær	2034	N/A	368179
Condenser Microphone	Brüel & Kjær	Cartridge Type 4133	414149	N/A
Subwoofers, 15" diameter	Blaupunkt	PC15	N/A	1389 & 1391
Power Amplifier	Crown	460 CSL™	A126785	N/A
Vane Anemometer	AM	5000	N/A	119947
Kiel Probe	United Sensor	KAC -12	N/A	N/A
Inclined Manometer	Dwyer	N/A	N/A	Department number, 116

APPENDIX B

Resonance Frequencies of the Copper Plate and the Plywood Channel Bottom

Density of Copper: $\rho_{Cu} = 8900 \frac{kg}{m^3}$ Modulus of Elasticity: $E_{Cu} = 121 \cdot 10^9 \cdot Pa$

Poisson's Ratio: $\nu_{Cu} = 0.33$ [Juvinall and Marshek, 1991]

Plate Dimensions: $b = 30 \cdot in$ $a = 18.7 \cdot in$ $h = 0.375 \cdot in$

Mass of the Plate: $m_{plate} = \rho_{Cu} \cdot b \cdot a \cdot h$ $m_{plate} = 30.682 \cdot kg$

Distributing the mass of copper tubing over the plate:

Outside Diameter: $d_o = 0.25 \cdot in$ Thickness: $t = 0.03 \cdot in$

Length of tubing: $l_t = 24 \cdot (18.7 \cdot in) + 20 \cdot (0.5) \cdot \pi \cdot (1.25 \cdot in) + 8 \cdot (3 \cdot in)$ $l_t = 13.007 \cdot m$

Mass of tubing: $m_{tube} = \rho_{Cu} \cdot \frac{\pi}{4} \cdot [d_o^2 - (d_o - t)^2] \cdot l_t$ $m_{tube} = 0.827 \cdot kg$

Combined mass: $m_{total} = m_{plate} + m_{tube}$ $m_{total} = 31.509 \cdot kg$

Mass per unit area: $m_- = \frac{m_{total}}{a \cdot b}$ $m_- = 87.058 \cdot kg \cdot m^{-2}$

Flexural Rigidity: $D_{Cu} = \frac{E_{Cu} \cdot h^3}{12 \cdot (1 - \nu_{Cu}^2)}$ $D_{Cu} = 9.779 \cdot 10^3 \cdot kg \cdot m^2 \cdot sec^{-2}$
[Szilard, 1974]

The copper plate was constrained in the direction of the long axis of the channel by the speaker box and the resonance chamber. It rested on the top of the channel walls and was not constrained perpendicular to the long axis of the channel or in the vertical direction. The natural frequency of the copper plate was calculated assuming the edges next to the speaker box and resonance chamber were pinned and the other edges were free.

Natural Frequency for a plate pinned on two edges and free on the other two edges [Szilard, 1974]. $\lambda_{11} = 9.87$

$$\omega_{pin_free} = \frac{\lambda_{11}}{a^2} \cdot \sqrt{\frac{D_{Cu}}{m_-}} \quad f_{pin_free} = \frac{\omega_{pin_free}}{2 \cdot \pi} \quad f_{pin_free} = 73.794 \cdot sec^{-1}$$

Since the mechanical properties of plywood vary and can not be determined precisely, a range of values for the Modulus of Elasticity was used.

Modulus of Elasticity:

Density of plywood: $\rho_{ply} = 540 \cdot \frac{kg}{m^3}$	Douglas Fir	Eastern White Pine
[Incropera and DeWitt, 1991]	$E_{ply} = 1950 \cdot 10^3 \cdot psi$	$E_{ply} = 1240 \cdot 10^3 \cdot psi$
Poisson's Ratio: $\nu = 0.33$	[Mark's Standard Handbook for Mechanical Engineers]	

Plate Dimensions: $b = 30 \cdot in$ $a = 18.7 \cdot in$ $h = 0.50 \cdot in$

Mass of the Plate: $m_{ply} = \rho_{ply} \cdot b \cdot a \cdot h$ $m_{ply} = 2.482 \cdot kg$

Mass per unit area: $m_p = \frac{m_{ply}}{a \cdot b}$ $m_p = 6.858 \cdot kg \cdot m^{-2}$

Flexural Rigidity: $D_{ply} = \frac{E_{ply} \cdot h^3}{12 \cdot (1 - \nu^2)}$ [Szilard, 1974]

(Douglas fir) $D_{ply} = 2.575 \cdot 10^3 \cdot kg \cdot m^2 \cdot sec^{-2}$

(Eastern White Pine) $D_{ply} = 1.638 \cdot 10^3 \cdot kg \cdot m^2 \cdot sec^{-2}$

The plywood channel bottom was anchored to the channel walls with four screws per wall and to the resonance chamber and speaker box with angle brackets. Using the same boundary conditions as the copper plate the natural frequency of the plywood bottom was calculated.

Natural Frequency for a plate pinned on two edges $\lambda_{11} = 9.87$ and free on the other two edges [Szilard, 1974].

$$\omega_{pin_free} = \frac{\lambda_{11}}{a^2} \cdot \sqrt{\frac{D_{ply}}{m_p}} \quad f_{pin_free} = \frac{\omega_{pin_free}}{2 \cdot \pi}$$

(Douglas fir) (Eastern White Pine)

$$f_{pin_free} = 134.932 \cdot sec^{-1} \quad f_{pin_free} = 107.599 \cdot sec^{-1}$$

APPENDIX C

Data Acquisition Program

```
DECLARE SUB HTC (width%, Temp(), Tcal(), Flux(), h(), Lc())
DECLARE SUB TCalculate (Temp!(), Tcal!())
DECLARE SUB TEMPSET (TC!())
DECLARE SUB GAGE12 (n%, TC!(), HFG!(), t%, A%, B%, g%, Troom!, SPL!, freq!)
DECLARE SUB AVERAGE (n%, TC!(), TCs(), t%, Troom1!, Troom2!, Troom!, SPL1!, SPL2!, SPL!,
freq1!, freq2!, freq!, HFG!(), HFGs(), g%)
DECLARE SUB CHANNEL (width%, Dh)
DECLARE SUB CHANNEL1 (n%, TC(), HFG(), t%, g%, Troom, SPL, freq)
DECLARE SUB CHANNEL2 (n%, TC(), HFG(), t%, g%, Troom, SPL, freq)
DECLARE SUB CHANNEL3 (n%, TC(), HFG(), t%, g%, Troom, SPL, freq)
DECLARE SUB PLATE (t%, TC!(), Temp!(), Tavg, STD)
DECLARE SUB NUSSELT (g%, Temp!(), h!(), Dh!, Lc!(), Tair, kair!(), kavg!, NuD!(), NuL!())
DECLARE SUB DISPLAY (test$, Temp!(), Troom!, Tavg!, Tair, kair!(), kavg!, t%, Flux!(), g%, h!(),
NuD!(), NuL!(), SPL, freq, gages%, width%)
DECLARE SUB AIR (x%, Temp!(), Tair, kair!(), kavg!)
DECLARE SUB HEATFLUX (HFG!(), g%, Flux!())
DECLARE SUB TEMPCONV (TC!(), t%, Temp!())
' **** Thermocouple and Heat Flux Gage Reader Program ****
' Connect Thermocouples to Card #1 in slot 3 (Channels 60-79), in order
' shown on diagram. Connect Heat Flux Gages to the first four channels
' of Card #2 in slot 4 (Channels 80-99)
' ***** Ice Bath Thermocouple, Card #1, slot 3, Channel 79 *****
' remember to run uli.com (universal language interface)
' Programmer: John Preston

' Program only calculate local Heat Transfer Coefficients and local Nusselt
' Numbers; the results of several tests must be combined for average values.

SCREEN 12
bkg& = 65536 * 33
PALETTE 0, bkg&

OPEN "GPIB0" FOR OUTPUT AS #1 'INITIALIZE PC CARD
OPEN "GPIB0" FOR INPUT AS #2
PRINT #1, "ABORT"
PRINT #1, "RESET"
PRINT #1, "TIMEOUT 12"
PRINT #1, "REMOTE 09"
PRINT #1, "GPIBEOS OUT CR" ' THIS IS VERY IMPORTANT!!
' PRINT #1, "GPIBEOS IN CR" ' don't use this
PRINT #1, "CLEAR 09"
```

```

DIM SHARED TC(20)      ' Output Voltage from Thermocouples
DIM SHARED HFG(14)     ' Output Voltage from Heat Flux Gages
DIM SHARED Temp(20)    ' Thermocouple Temperatures
DIM SHARED Flux(14)    ' Heat Flux from the plate
DIM SHARED h(14)       ' Heat Transfer Coefficient, (W/m^2 K), local
DIM SHARED Dh          ' Hydraulic Diameter of the channel
DIM SHARED Lc(14)      ' Characteristic Length from air inlet to HFG
DIM SHARED NuD(14)     ' Nusselt Number, at HFG location, based on diameter
DIM SHARED NuL(14)     ' Nusselt Number, at HFG location, based on length
DIM SHARED kair(14)    ' Thermal conductivity of air at HFG location
DIM SHARED TCs(20)     ' Sum of TC(i%)
DIM SHARED HFGs(14)    ' Sum of HFG(i%)
DIM SHARED Tcal(14)    ' Temperature at HFG locations
DIM SHARED t%          ' Number of temperature measurements
DIM SHARED g%          ' Number of heat flux gages

```

begin:

```
CALL TEMPSET(TC())
```

check:

```
PRINT " Check for Uniform Plate Temperature Before Starting Data Collection "
```

```
CALL PLATE(t%, TC(), Temp(), Tavg, STD)
```

```
SOUND 1500, 20
```

```
PRINT
```

```
PRINT " Average Plate Temperature "; Tavg; " degrees Celsius"
```

```
PRINT " Standard Deviation "; STD; " degrees Celsius"
```

```
INPUT " Do you want to recheck the plate temperature (y or n)"; r$
```

```
IF r$ = "y" OR r$ = "Y" THEN GOTO check
```

```
PRINT
```

```
INPUT " Test Number "; test$
```

```
PRINT
```

```
n% = 20          'Number of Data Collection Cycles
```

```
INPUT "Enter the number of Heat Flux Gages"; gages%
```

```
IF gages% = 2 THEN
```

```
INPUT " Enter the location of the Heat Flux Gages, (RdF and Omega)"; A%, B%
```

GAGEPOSITION:

```
IF A% = 1 AND B% = 2 THEN
```

```
CALL GAGE12(n%, TC(), HFG(), t%, A%, B%, g%, Troom, SPL, freq)
```

' Calculations

```
CALL TEMPCONV(TC(), t%, Temp())
```

```
CALL HEATFLUX(HFG(), g%, Flux())
```

```
CALL CHANNEL(width%, Dh)
```

```
Lc(A%) = 11 * .0254 'Characteristic length from air inlet to gage
```

```
Lc(B%) = 11 * .0254
```

```

' Calculate Heat Transfer Coefficient
  h(A%) = Flux(A%) / (Temp(1) - Temp(2))
  h(B%) = Flux(B%) / (Temp(1) - Temp(2))

  CALL NUSSELT(g%, Temp(), h(), Dh, Lc(), Tair, kair(), kavg, NuD(), NuL())

  CALL DISPLAY(test$, Temp(), Troom, Tavg, Tair, kair(), kavg, t%, Flux(), g%, h(), NuD(),
NuL(), SPL, freq, gages%, width%)
  ELSE GOTO GAGEPOSITION
  END IF

ELSEIF gages% = 6 THEN
  CALL CHANNEL(width%, Dh)
  IF width% = 1 THEN
    CALL CHANNEL1(n%, TC(), HFG(), t%, g%, Troom, SPL, freq)
  ELSEIF width% = 2 THEN
    CALL CHANNEL2(n%, TC(), HFG(), t%, g%, Troom, SPL, freq)
  ELSE
    CALL CHANNEL3(n%, TC(), HFG(), t%, g%, Troom, SPL, freq)
  END IF

' Calculations

  CALL TEMPCONV(TC(), t%, Temp())
  CALL HEATFLUX(HFG(), g%, Flux())
  CALL HTC(width%, Temp(), Tcal(), Flux(), h(), Lc())
  CALL NUSSELT(g%, Temp(), h(), Dh, Lc(), Tair, kair(), kavg!, NuD(), NuL())
  CALL DISPLAY(test$, Temp(), Troom, Tavg, Tair, kair(), kavg!, t%, Flux(), g%, h(), NuD(), NuL(),
SPL, freq, gages%, width%)

  END IF

  INPUT " Do you wish to collect more data (y or n) "; r$
  IF r$ = "n" OR r$ = "N" GOTO gotodos 'Exit the program
  INPUT " Do you wish to reset the plate temperature (y or n) "; r$
  IF r$ = "y" OR r$ = "Y" GOTO begin
  GOTO check 'Else check for uniform plate temp
gotodos:
END 'End of program

SUB AIR (x%, Temp(), Tair, kair(), kavg!)
' Subroutine calculates local and average values of thermal conductivity
' of the air in the channel based an equation on page 507 of "Theory and
' Design For Mechanical Measurements" by Figliola and Beasley.

  A = -2.45 'Constants
  B = .113
  c = -6.287E-05
  d = 1.891E-08

' Select the appropriate local air temperature value

```

```

IF x% = 1 OR x% = 2 OR x% = 3 OR x% = 6 OR x% = 9 THEN
    airt% = 2
ELSEIF x% = 4 OR x% = 7 OR x% = 10 THEN
    airt% = 5
ELSEIF x% = 5 OR x% = 8 OR x% = 11 THEN
    airt% = 9
ELSE airt% = 2      'Default value
END IF

Tlocal = Temp(airt%) + 273.15      'Convert to Kelvin

kair(x%) = (A + B * Tlocal + c * Tlocal ^ 2 + d * Tlocal ^ 3) * .001

Tair = ((Temp(2) + Temp(5) + Temp(6) + Temp(9) + Temp(10)) / 5) + 273.15

kavg! = (A + B * Tair + c * Tair ^ 2 + d * Tair ^ 3) * .001

END SUB

SUB AVERAGE (n%, TC(), TCs(), t%, Troom1, Troom2, Troom, SPL1, SPL2, SPL, freq1, freq2, freq,
HFG(), HFGs(), g%)
' Subroutine averages the values from data accquisition

    FOR k% = 1 TO t%
        TC(k%) = TCs(k%) / n%
    NEXT k%
    TC(19) = TCs(19) / n%
    Troom = (Troom1 + Troom2) / 2
    Troom = (Troom - 32) / 1.8      'Convert to Celsius
    SPL = (SPL1 + SPL2) / 2
    freq = (freq1 + freq2) / 2
    FOR k% = 1 TO g%
        HFG(g%) = HFGs(g%) / n%
    NEXT k%

    FOR k% = 1 TO t%                'Reset variables
        TCs(k%) = 0!
    NEXT k%
    TCs(19) = 0!
    Troom1 = 0!
    Troom2 = 0!
    SPL1 = 0!
    SPL2 = 0!
    freq1 = 0!
    freq2 = 0!
    FOR k% = 1 TO g%
        HFGs(k%) = 0!
    NEXT k%

END SUB

```

SUB CHANNEL (width%, Dh)

' Subroutine calculates the hydraulic diameter of the channel
' based on the channel wall position (in meters).

wall:

INPUT "Enter Channel Position, (1, 2, or 3) "; width%

IF width% = 1 THEN

Dh = ((2 * 2.5 * 6.2) / (2.5 + 6.2)) * .0254 'Hydraulic Diameter [m]

ELSEIF width% = 2 THEN

Dh = ((2 * 2.5 * 11.3) / (2.5 + 11.3)) * .0254 'Hydraulic Diameter [m]

ELSEIF width% = 3 THEN

Dh = ((2 * 2.5 * 17.2) / (2.5 + 17.2)) * .0254 'Hydraulic Diameter [m]

ELSE GOTO wall

END IF

END SUB

SUB CHANNEL1 (n%, TC(), HFG(), t%, g%, Troom, SPL, freq)

' Subroutine collects the data for the first channel position

INPUT " Enter room air temperature (degrees Farenheit) "; Troom1

INPUT " Enter the maximum SPL (dB) and the corresponding frequency "; SPL1, freq1

PRINT " Press any key to begin data collection"

ready1: d\$ = INKEY\$: IF d\$ = "" GOTO ready1

PRINT " Data Collection In Progress"

start! = TIMER

FOR i% = 1 TO n%

PRINT #1, "OUTPUT 09; AF69 AL67"

PRINT #1, "OUTPUT 09; AC69"

FOR j% = 10 TO 8 STEP -1

PRINT #1, "ENTER 09"

PRINT #1, "OUTPUT 09;AS"

INPUT #2, TC(j%) 'Thermcouples #10 - #8

NEXT j%

PRINT #1, "OUTPUT 09; AC80"

PRINT #1, "ENTER 09"

INPUT #2, HFG(1) 'Gage Position #1

PRINT #1, "OUTPUT 09; AC81"

PRINT #1, "ENTER 09"

INPUT #2, HFG(2) 'Gage Position #2

PRINT #1, "OUTPUT 09; AC66"

PRINT #1, "ENTER 09"

INPUT #2, TC(7) 'Thermcouple #7

PRINT #1, "OUTPUT 09; AC71"

PRINT #1, "ENTER 09"

INPUT #2, TC(12) 'Thermocouple #12

PRINT #1, "OUTPUT 09; AC83"

PRINT #1, "ENTER 09"

INPUT #2, HFG(4) 'Gage Position #7

PRINT #1, "OUTPUT 09; AC82"

```

PRINT #1, "ENTER 09"
INPUT #2, HFG(3)          'Gage Position #6
PRINT #1, "OUTPUT 09; AF60 AL62"
PRINT #1, "OUTPUT 09; AC60"
FOR j% = 1 TO 3
    PRINT #1, "ENTER 09"
    PRINT #1, "OUTPUT 09;AS"
    INPUT #2, TC(j%)      'Thermocouples #1 - #3
NEXT j%
PRINT #1, "OUTPUT 09; AC70"
PRINT #1, "ENTER 09"
INPUT #2, TC(11)         'Thermocouple #11
PRINT #1, "OUTPUT 09; AC84"
PRINT #1, "ENTER 09"
INPUT #2, HFG(5)         'Gage Position #11
PRINT #1, "OUTPUT 09; AC85"
PRINT #1, "ENTER 09"
INPUT #2, HFG(6)         'Gage Position #10
PRINT #1, "OUTPUT 09; AF63 AL65"
PRINT #1, "OUTPUT 09; AC63"
FOR j% = 4 TO 6
    PRINT #1, "ENTER 09"
    PRINT #1, "OUTPUT 09;AS"
    INPUT #2, TC(j%)      'Thermocouples #4 - #6
NEXT j%
PRINT #1, "OUTPUT 09; AC79"
PRINT #1, "ENTER 09"
INPUT #2, TC(19)         'Ice Bath Reference

FOR k% = 1 TO 12         'Sum measurements, used for avg value
    TCs(k%) = TCs(k%) + TC(k%)
NEXT k%
    TCs(19) = TCs(19) + TC(19)
FOR k% = 1 TO 6
    HFGs(k%) = HFGs(k%) + HFG(k%)
NEXT k%

NEXT i%
finish! = TIMER
SOUND 1500, 20
INPUT " Enter room air temperature (degrees Farenheit) "; Troom2
INPUT " Enter the maximum SPL (dB) and the corresponding frequency "; SPL2, freq2
PRINT " Data Collection Finished"
PRINT "Data Collection took ", (finish! - start!) / 60, " minutes"

t% = 12                  'Temperatures
g% = 6                   'Heat flux gages
CALL AVERAGE(n%, TC(), TCs(), t%, Troom1, Troom2, Troom, SPL1, SPL2, SPL, freq1, freq2,
freq, HFG(), HFGs(), g%)

END SUB

```

SUB CHANNEL2 (n%, TC(), HFG(), t%, g%, Troom, SPL, freq)

' Subroutine collects the data for second channel position

INPUT " Enter room air temperature (degrees Farenheit) "; Troom1

INPUT " Enter the maximum SPL (dB) and the corresponding frequency "; SPL1, freq1

PRINT " Press any key to begin data collection"

ready2: d\$ = INKEY\$: IF d\$ = "" GOTO ready2

PRINT " Data Collection In Progress"

start! = TIMER

FOR i% = 1 TO n%

PRINT #1, "OUTPUT 09; AF69 AL67"

PRINT #1, "OUTPUT 09; AC69"

FOR j% = 10 TO 8 STEP -1

PRINT #1, "ENTER 09"

PRINT #1, "OUTPUT 09;AS"

INPUT #2, TC(j%) 'Thermcouples #10 - #8

NEXT j%

PRINT #1, "OUTPUT 09; AC80"

PRINT #1, "ENTER 09"

INPUT #2, HFG(1) 'Gage Position #2

PRINT #1, "OUTPUT 09; AC81"

PRINT #1, "ENTER 09"

INPUT #2, HFG(2) 'Gage Position #3

PRINT #1, "OUTPUT 09; AC66"

PRINT #1, "ENTER 09"

INPUT #2, TC(7) 'Thermcouple #7

PRINT #1, "OUTPUT 09; AC71"

PRINT #1, "ENTER 09"

INPUT #2, TC(12) 'Thermocouple #12

PRINT #1, "OUTPUT 09; AC73"

PRINT #1, "ENTER 09"

INPUT #2, TC(14) 'Thermocouple #14

PRINT #1, "OUTPUT 09; AC83"

PRINT #1, "ENTER 09"

INPUT #2, HFG(4) 'Gage Position #8

PRINT #1, "OUTPUT 09; AC82"

PRINT #1, "ENTER 09"

INPUT #2, HFG(3) 'Gage Position #6

PRINT #1, "OUTPUT 09; AF60 AL62"

PRINT #1, "OUTPUT 09; AC60"

FOR j% = 1 TO 3

PRINT #1, "ENTER 09"

PRINT #1, "OUTPUT 09;AS"

INPUT #2, TC(j%) 'Thermcouples #1 - #3

NEXT j%

PRINT #1, "OUTPUT 09; AC70"

PRINT #1, "ENTER 09"

INPUT #2, TC(11) 'Thermocouple #11

PRINT #1, "OUTPUT 09; AC72"

PRINT #1, "ENTER 09"

```

    INPUT #2, TC(13)          'Thermocouple #13
    PRINT #1, "OUTPUT 09; AC84"
    PRINT #1, "ENTER 09"
    INPUT #2, HFG(5)          'Gage Position #12
    PRINT #1, "OUTPUT 09; AC85"
    PRINT #1, "ENTER 09"
    INPUT #2, HFG(6)          'Gage Position #11
    PRINT #1, "OUTPUT 09; AF63 AL65"
    PRINT #1, "OUTPUT 09; AC63"
    FOR j% = 4 TO 6
        PRINT #1, "ENTER 09"
        PRINT #1, "OUTPUT 09;AS"
        INPUT #2, TC(j%)      'Thermocouples #4 - #6
    NEXT j%
    PRINT #1, "OUTPUT 09; AC79"
    PRINT #1, "ENTER 09"
    INPUT #2, TC(19)          'Ice Bath Reference

    FOR k% = 1 TO 14          'Sum measurements, used for avg value
        TCs(k%) = TCs(k%) + TC(k%)
    NEXT k%
    TCs(19) = TCs(19) + TC(19)
    FOR k% = 1 TO 6
        HFGs(k%) = HFGs(k%) + HFG(k%)
    NEXT k%

    NEXT i%
    finish! = TIMER
    SOUND 1500, 20
    INPUT " Enter room air temperature (degrees Farenheit) "; Troom2
    INPUT " Enter the maximum SPL (dB) and the corresponding frequency "; SPL2, freq2
    PRINT " Data Collection Finished"
    PRINT "Data Collection took ", (finish! - start!) / 60, " minutes"

    t% = 14                   'Temperatures
    g% = 6                     'Heat flux gages
    CALL AVERAGE(n%, TC(), TCs(), t%, Troom1, Troom2, Troom, SPL1, SPL2, SPL, freq1, freq2,
    freq, HFG(), HFGs(), g%)

END SUB

SUB CHANNEL3 (n%, TC(), HFG(), t%, g%, Troom, SPL, freq)
' Subroutine collects the data for the third channel position

    INPUT " Enter room air temperature (degrees Farenheit) "; Troom1
    INPUT " Enter the maximum SPL (dB) and the corresponding frequency "; SPL1, freq1
    PRINT " Press any key to begin data collection"
    ready3: d$ = INKEY$: IF d$ = "" GOTO ready3

    PRINT " Data Collection In Progress"
    start! = TIMER

```

```

FOR i% = 1 TO n%
  PRINT #1, "OUTPUT 09; AF69 AL67"
  PRINT #1, "OUTPUT 09; AC69"
  FOR j% = 10 TO 8 STEP -1
    PRINT #1, "ENTER 09"
    PRINT #1, "OUTPUT 09;AS"
    INPUT #2, TC(j%)      'Thermocouples #10 - #8
  NEXT j%
  PRINT #1, "OUTPUT 09; AC80"
  PRINT #1, "ENTER 09"
  INPUT #2, HFG(1)        'Gage Position #3
  PRINT #1, "OUTPUT 09; AC81"
  PRINT #1, "ENTER 09"
  INPUT #2, HFG(2)        'Gage Position #4
  PRINT #1, "OUTPUT 09; AC66"
  PRINT #1, "ENTER 09"
  INPUT #2, TC(7)         'Thermocouple #7
  PRINT #1, "OUTPUT 09; AC71"
  PRINT #1, "ENTER 09"
  INPUT #2, TC(12)        'Thermocouple #12
  PRINT #1, "OUTPUT 09; AC73"
  PRINT #1, "ENTER 09"
  INPUT #2, TC(14)        'Thermocouple #14
  PRINT #1, "OUTPUT 09; AC75"
  PRINT #1, "ENTER 09"
  INPUT #2, TC(16)        'Thermocouple #16
  PRINT #1, "OUTPUT 09; AC83"
  PRINT #1, "ENTER 09"
  INPUT #2, HFG(4)        'Gage Position #9
  PRINT #1, "OUTPUT 09; AC82"
  PRINT #1, "ENTER 09"
  INPUT #2, HFG(3)        'Gage Position #6
  PRINT #1, "OUTPUT 09; AF60 AL62"
  PRINT #1, "OUTPUT 09; AC60"
  FOR j% = 1 TO 3
    PRINT #1, "ENTER 09"
    PRINT #1, "OUTPUT 09;AS"
    INPUT #2, TC(j%)      'Thermocouples #1 - #3
  NEXT j%
  PRINT #1, "OUTPUT 09; AC70"
  PRINT #1, "ENTER 09"
  INPUT #2, TC(11)        'Thermocouple #11
  PRINT #1, "OUTPUT 09; AC72"
  PRINT #1, "ENTER 09"
  INPUT #2, TC(13)        'Thermocouple #13
  PRINT #1, "OUTPUT 09; AC74"
  PRINT #1, "ENTER 09"
  INPUT #2, TC(15)        'Thermocouple #15
  PRINT #1, "OUTPUT 09; AC84"
  PRINT #1, "ENTER 09"
  INPUT #2, HFG(5)        'Gage Position #13

```

```

PRINT #1, "OUTPUT 09; AC85"
PRINT #1, "ENTER 09"
INPUT #2, HFG(6)          'Gage Position #12
PRINT #1, "OUTPUT 09; AF63 AL65"
PRINT #1, "OUTPUT 09; AC63"
FOR j% = 4 TO 6
    PRINT #1, "ENTER 09"
    PRINT #1, "OUTPUT 09;AS"
    INPUT #2, TC(j%)      'Thermcouple #4 - #6
NEXT j%
PRINT #1, "OUTPUT 09; AC79"
PRINT #1, "ENTER 09"
INPUT #2, TC(19)         'Ice Bath Reference

FOR k% = 1 TO 16         'Sum measurements, used for avg value
    TCs(k%) = TCs(k%) + TC(k%)
NEXT k%
TCs(19) = TCs(19) + TC(19)
FOR k% = 1 TO 6
    HFGs(k%) = HFGs(k%) + HFG(k%)
NEXT k%

NEXT i%
finish! = TIMER
SOUND 1500, 20
INPUT " Enter room air temperature (degrees Farenheit) "; Troom2
INPUT " Enter the maximum SPL (dB) and the corresponding frequency "; SPL2, freq2
PRINT " Data Collection Finished"
PRINT "Data Collection took ", (finish! - start!) / 60, " minutes"

t% = 16                  'Temperatures
g% = 6                   'Heat flux gages
CALL AVERAGE(n%, TC(), TCs(), t%, Troom1, Troom2, Troom, SPL1, SPL2, SPL, freq1, freq2,
freq, HFG(), HFGs(), g%)

END SUB

SUB DISPLAY (test$, Temp(), Troom, Tav, Tair, kair(), kav, t%, Flux(), g%, h(), NuD(), NuL(),
SPL, freq, gages%, width%)
' Subroutine prints calculated results to the screen and to a data file

blower:
INPUT " Damper or Plate (d or p) "; restrict$
IF restrict$ = "d" OR restrict$ = "D" THEN GOTO damper
INPUT " Enter Plate Used (A or B) "; p$
IF p$ = "A" OR p$ = "a" THEN
    vel = 177.5 / 60
ELSEIF p$ = "B" OR p$ = "b" THEN
    vel = 350 / 60
ELSE GOTO blower
END IF

```

```

GOTO printing
damper:
INPUT " Enter Blower Damper Position (0, .125, 1, 2, 3, or 4) "; blow
IF blow = 0 THEN
    vel = 0
ELSEIF blow = .125 THEN
    vel = 835 / 60
ELSEIF blow = 1 THEN
    vel = 1029 / 60 'Air Velocities based on
ELSEIF blow = 2 THEN
    vel = 3024 / 60 'previous average values
ELSEIF blow = 3 THEN
    vel = 3905 / 60 'from calibration
ELSEIF blow = 4 THEN
    vel = 5405 / 60
ELSE GOTO blower
END IF
printing:
PRINT "Test Number", test$, DATE$, TIME$
PRINT
PRINT "Thermocouples: Position      Temperature"
FOR i% = 1 TO t%
    PRINT "              "; i%, Temp(i%)
NEXT i%

PRINT
PRINT "Average Plate Temperature "; Tavg; " degrees Celsius"
PRINT "Average Air Temperature in the Channel "; Tair - 273.15; " degrees Celsius"
PRINT "Average Air Thermal Conductivity, k, "; kavg!; " W/m K"
PRINT "Average Room Temperature during the test "; Troom; " degrees Celsius"
PRINT "Average Air Velocity in the channel "; vel; " ft/s"
PRINT "Average Sound Pressure Level, SPL, "; SPL; " dB"
PRINT "Average Frequency, "; freq; " Hz"
PRINT
PRINT "Position Flux   HTC   NuD   NuL   kair"
IF gages% = 2 THEN
    PRINT A%, Flux(A%), h(A%), NuD(A%), NuL(A%), kair(A%)
    PRINT B%, Flux(B%), h(B%), NuD(B%), NuL(B%), kair(B%)
ELSEIF gages% = 6 THEN
    IF width% = 1 THEN
        PRINT 1, Flux(1), h(1), NuD(1), NuL(1), kair(1)
        PRINT 2, Flux(2), h(2), NuD(2), NuL(2), kair(2)
        PRINT 6, Flux(3), h(3), NuD(3), NuL(3), kair(3)
        PRINT 7, Flux(4), h(4), NuD(4), NuL(4), kair(4)
        PRINT 10, Flux(5), h(5), NuD(5), NuL(5), kair(5)
        PRINT 11, Flux(6), h(6), NuD(6), NuL(6), kair(6)
    ELSEIF width% = 2 THEN
        PRINT 2, Flux(1), h(1), NuD(1), NuL(1), kair(1)
        PRINT 3, Flux(2), h(2), NuD(2), NuL(2), kair(2)
        PRINT 6, Flux(3), h(3), NuD(3), NuL(3), kair(3)
        PRINT 8, Flux(4), h(4), NuD(4), NuL(4), kair(4)

```

```

    PRINT 11, Flux(5), h(5), NuD(5), NuL(5), kair(5)
    PRINT 12, Flux(6), h(6), NuD(6), NuL(6), kair(6)
ELSE
    'width% = 3
    PRINT 3, Flux(1), h(1), NuD(1), NuL(1), kair(1)
    PRINT 4, Flux(2), h(2), NuD(2), NuL(2), kair(2)
    PRINT 6, Flux(3), h(3), NuD(3), NuL(3), kair(3)
    PRINT 9, Flux(4), h(4), NuD(4), NuL(4), kair(4)
    PRINT 12, Flux(5), h(5), NuD(5), NuL(5), kair(5)
    PRINT 13, Flux(6), h(6), NuD(6), NuL(6), kair(6)
END IF
END IF
PRINT

INPUT " Do you want to save the data to a file (y or n)"; r$
IF r$ = "n" OR r$ = "N" THEN GOTO leave
saveit: PRINT : INPUT " Enter drive:file name for data ", n$
        PRINT " Insert Formated Disk In Drive - Press Any Key When Ready"

save: d$ = INKEY$: IF d$ = "" GOTO save
OPEN n$ FOR OUTPUT AS #3 'OPEN DATA PATH

PRINT #3, "Test Number", test$, DATE$, TIMES
PRINT #3,
PRINT #3, "Thermocouples: Position Temperature"
FOR i% = 1 TO t%
    PRINT #3, i%, Temp(i%)
NEXT i%

PRINT #3,
PRINT #3, Tavg
PRINT #3, Tair - 273.15
PRINT #3, kavg!
PRINT #3, Troom
PRINT #3, vel
PRINT #3, SPL
PRINT #3, freq
PRINT #3,
PRINT #3, "Position Flux HTC NuD NuL kair"
IF gages% = 2 THEN
    PRINT #3, A%, Flux(A%), h(A%), NuD(A%), NuL(A%), kair(A%)
    PRINT #3, B%, Flux(B%), h(B%), NuD(B%), NuL(B%), kair(B%)
ELSEIF gages% = 6 THEN
    IF width% = 1 THEN
        PRINT #3, 1, Flux(1), h(1), NuD(1), NuL(1), kair(1)
        PRINT #3, 2, Flux(2), h(2), NuD(2), NuL(2), kair(2)
        PRINT #3, 6, Flux(3), h(3), NuD(3), NuL(3), kair(3)
        PRINT #3, 7, Flux(4), h(4), NuD(4), NuL(4), kair(4)
        PRINT #3, 10, Flux(5), h(5), NuD(5), NuL(5), kair(5)
        PRINT #3, 11, Flux(6), h(6), NuD(6), NuL(6), kair(6)
    ELSEIF width% = 2 THEN
        PRINT #3, 2, Flux(1), h(1), NuD(1), NuL(1), kair(1)

```

```

PRINT #3, 3, Flux(2), h(2), NuD(2), NuL(2), kair(2)
PRINT #3, 6, Flux(3), h(3), NuD(3), NuL(3), kair(3)
PRINT #3, 8, Flux(4), h(4), NuD(4), NuL(4), kair(4)
PRINT #3, 11, Flux(5), h(5), NuD(5), NuL(5), kair(5)
PRINT #3, 12, Flux(6), h(6), NuD(6), NuL(6), kair(6)
ELSE 'width% = 3
PRINT #3, 3, Flux(1), h(1), NuD(1), NuL(1), kair(1)
PRINT #3, 4, Flux(2), h(2), NuD(2), NuL(2), kair(2)
PRINT #3, 6, Flux(3), h(3), NuD(3), NuL(3), kair(3)
PRINT #3, 9, Flux(4), h(4), NuD(4), NuL(4), kair(4)
PRINT #3, 12, Flux(5), h(5), NuD(5), NuL(5), kair(5)
PRINT #3, 13, Flux(6), h(6), NuD(6), NuL(6), kair(6)
END IF
END IF

CLOSE #3
ON ERROR GOTO 0
GOTO leave
wrong: PRINT "ERROR": GOTO saveit

leave:

END SUB

SUB GAGE12 (n%, TC(), HFG(), t%, A%, B%, g%, Troom, SPL, freq)
' Subroutine collects the data for gage positions 1 & 2

INPUT " Enter room air temperature (degrees Farenheit) "; Troom1
INPUT " Enter the maximum SPL (dB) and the corresponding frequency "; SPL1, freq1
PRINT " Press any key to begin data collection"
ready12: d$ = INKEY$: IF d$ = "" GOTO ready12

PRINT " Data Collection In Progress"
start! = TIMER
FOR i% = 1 TO n%
PRINT #1, "OUTPUT 09;AC69"
PRINT #1, "ENTER 09"
INPUT #2, TC(10)
PRINT #1, "OUTPUT 09;AC68"
PRINT #1, "ENTER 09"
INPUT #2, TC(9)
PRINT #1, "OUTPUT 09;AC67"
PRINT #1, "ENTER 09"
INPUT #2, TC(8)
PRINT #1, "OUTPUT 09;AC66"
PRINT #1, "ENTER 09"
INPUT #2, TC(7)
PRINT #1, "OUTPUT 09;AC60"
PRINT #1, "ENTER 09"
INPUT #2, TC(1)
PRINT #1, "OUTPUT 09;AC80"

```

```

        PRINT #1, "ENTER 09"
        INPUT #2, HFG(A%)
    PRINT #1, "OUTPUT 09;AC81"
        PRINT #1, "ENTER 09"
        INPUT #2, HFG(B%)
    PRINT #1, "OUTPUT 09;AC61"
        PRINT #1, "ENTER 09"
        INPUT #2, TC(2)
    PRINT #1, "OUTPUT 09;AF62 AL65"
    PRINT #1, "OUTPUT 09;AC62"
    FOR j% = 3 TO 6
        PRINT #1, "ENTER 09"
        PRINT #1, "OUTPUT 09;AS"
        INPUT #2, TC(j%)
    NEXT j%
    PRINT #1, "OUTPUT 09;AC79"
        PRINT #1, "ENTER 09"
        INPUT #2, TC(19)

    FOR k% = 1 TO 10      'Sum measurements, used for avg value
        TCs(k%) = TCs(k%) + TC(k%)
    NEXT k%
    TCs(19) = TCs(19) + TC(19)
    HFGs(A%) = HFGs(A%) + HFG(A%)
    HFGs(B%) = HFGs(B%) + HFG(B%)

    NEXT i%
    SOUND 1500, 20
    INPUT " Enter room air temperature (degrees Farenheit) "; Troom2
    INPUT " Enter the maximum SPL (dB) and the corresponding frequency "; SPL2, freq2
    PRINT " Data Collection Finished"
    finish! = TIMER
    PRINT "Data Collection took ", (finish! - start!) / 60, " minutes"

    t% = 10      'Temperatures
    g% = 2      'Heat flux gages
    CALL AVERAGE(n%, TC(), TCs(), t%, Troom1, Troom2, Troom, SPL1, SPL2, SPL, freq1, freq2,
    freq, HFG(), HFGs(), g%)

END SUB

SUB HEATFLUX (HFG(), g%, Flux())
' Subroutine converts heat flux gage voltages (uV) to heat flux values
' based on calibration data supplied by the gage manufacture.

    FOR i% = 1 TO g%
        HFG(i%) = HFG(i%) / .000001 'Convert to microvolts
    NEXT i%

    Flux(1) = HFG(1) / 3.15 'Flux in Btu/ft^2 hr, RdF #1 gage
    Flux(2) = HFG(2) / 3.24 'Flux in Btu/ft^2 hr, Omega gage

```

```

Flux(3) = HFG(3) / 3.23      'Flux in Btu/ft^2 hr, RdF #2 gage
Flux(4) = HFG(4) / 3.23      'Flux in Btu/ft^2 hr, RdF #3 gage
Flux(5) = HFG(5) / 3.2       'Flux in Btu/ft^2 hr, RdF #4 gage
Flux(6) = HFG(6) / 3.22      'Flux in Btu/ft^2 hr, RdF #5 gage

```

```

FOR i% = 1 TO g%
Flux(i%) = Flux(i%) * 3.1524  'Convert to SI units, W/m^2
HFG(i%) = 0!                  'Reset values
NEXT i%

```

END SUB

SUB HTC (width%, Temp(), Tcal(), Flux(), h(), Lc())

' Subroutine calculates the local heat transfer coefficient for each
' heat flux gage based on the plate temperature at the gage and the air
' temperature above the gage. Additionally, it calculates the characteristic
' length from air inlet to each gage

CALL TCALCULATE(Temp(), Tcal())

IF width% = 1 THEN

```

h(1) = Flux(1) / (Tcal(2) - Temp(9))  'Gage Pos. #1
Lc(1) = 1 * .0254
h(2) = Flux(2) / (Tcal(2) - Temp(9))  'Gage Pos. #2
Lc(2) = 1 * .0254
h(3) = Flux(3) / (Temp(1) - Temp(2))  'Gage Pos. #6
Lc(3) = 11 * .0254
h(4) = Flux(4) / (.5 * (Temp(11) + Temp(12)) - Temp(2)) 'Gage Pos. #7
Lc(4) = 11 * .0254
h(5) = Flux(5) / (Tcal(11) - Temp(5)) 'Gage Pos. #10
Lc(5) = 21 * .0254
h(6) = Flux(6) / (Tcal(11) - Temp(5)) 'Gage Pos. #11
Lc(6) = 21 * .0254

```

ELSEIF width% = 2 THEN

```

h(1) = Flux(1) / (Tcal(2) - Temp(9))  'Gage Pos. #2
Lc(1) = 1 * .0254
h(2) = Flux(2) / (Tcal(3) - Temp(9))  'Gage Pos. #3
Lc(2) = 1 * .0254
h(3) = Flux(3) / (Temp(1) - Temp(2))  'Gage Pos. #6
Lc(3) = 11 * .0254
h(4) = Flux(4) / (.5 * (Temp(13) + Temp(14)) - Temp(2)) 'Gage Pos. #8
Lc(4) = 11 * .0254
h(5) = Flux(5) / (Tcal(11) - Temp(5)) 'Gage Pos. #11
Lc(5) = 21 * .0254
h(6) = Flux(6) / (Tcal(12) - Temp(5)) 'Gage Pos. #12
Lc(6) = 21 * .0254

```

ELSE 'width% = 3 by default

```

h(1) = Flux(1) / (Tcal(3) - Temp(9)) 'Gage Pos. #3
Lc(1) = 1 * .0254
h(2) = Flux(2) / (Tcal(4) - Temp(9)) 'Gage Pos. #4
Lc(2) = 5 * .0254
h(3) = Flux(3) / (Temp(1) - Temp(2)) 'Gage Pos. #6
Lc(3) = 11 * .0254
h(4) = Flux(4) / (.5 * (Temp(15) + Temp(16)) - Temp(2)) 'Gage Pos. #9
Lc(4) = 11 * .0254
h(5) = Flux(5) / (Tcal(12) - Temp(5)) 'Gage Pos. #12
Lc(5) = 21 * .0254
h(6) = Flux(6) / (Tcal(13) - Temp(5)) 'Gage Pos. #13
Lc(6) = 21 * .0254

```

END IF

END SUB

SUB NUSSELT (g%, Temp(), h(), Dh, Lc(), Tair, kair(), kavg!, NuD(), NuL())

' Subroutine calculates Nusselt numbers based on the hydraulic diameter
' of the channel (constant for a given given channel position) and a
' characteristic length. The characteristic length is taken to be the
' distance along the long axis of the channel from the center of the
' air inlet to the center of the gage.

```

FOR i% = 1 TO g%
  CALL AIR(i%, Temp(), Tair, kair(), kavg!)
  NuD(i%) = h(i%) * Dh / kair(i%) 'Diameter
  NuL(i%) = h(i%) * Lc(i%) / kair(i%) 'Length
NEXT i%

```

END SUB

SUB PLATE (t%, TC(), Temp(), Tavg, STD)

' Subroutine checks for uniform plate temperature. After uniform
' temperature is determined, the program waits 30 seconds and rechecks
' to eliminate any flucuations.

```

Told = 0! 'Old Average Plate Temperature
STDold = 0! 'Old Standard Deviation
count% = 0

```

repeat:

start! = TIMER

platetemp:

```

PRINT #1, "OUTPUT 09;AC60"
PRINT #1, "ENTER 09"
INPUT #2, TC(1)
PRINT #1, "OUTPUT 09;AC62"
PRINT #1, "ENTER 09"
INPUT #2, TC(2) 'Thermocouple #3
PRINT #1, "OUTPUT 09;AC63"

```

```

PRINT #1, "ENTER 09"
INPUT #2, TC(3)      'Thermocouple #4
PRINT #1, "OUTPUT 09;AC66"
PRINT #1, "ENTER 09"
INPUT #2, TC(4)      'Thermocouple #7
PRINT #1, "OUTPUT 09;AC67"
PRINT #1, "ENTER 09"
INPUT #2, TC(5)      'Thermocouple #8
PRINT #1, "OUTPUT 09;AF70 AL75"      'First and Last Channel
PRINT #1, "OUTPUT 09;AC70"
FOR i% = 6 TO 11
PRINT #1, "ENTER 09"
PRINT #1, "OUTPUT 09;AS"      'Step through channels
INPUT #2, TC(i%)      'Thermocouples #11 - #16
NEXT i%
PRINT #1, "OUTPUT 09;AC79"      'Thermocouple Reference Voltage, Ice Bath
PRINT #1, "ENTER 09"
INPUT #2, TC(19)

t% = 11
CALL TEMPCONV(TC(), t%, Temp())
Tsum# = 0#
Tssum# = 0#

FOR i% = 1 TO t%
Tsum# = Tsum# + Temp(i%)
Tssum# = Tssum# + Temp(i%) ^ 2      'Sum of Squares
NEXT i%

Tavg = Tsum# / t%
var = (t% * Tssum# - Tsum# ^ 2) / (t% * (t% - 1))      'Sample Variance
IF var < 1E-08 THEN var = 1E-10      'Truncation
STD = var ^ .5      'Standard Deviation
IF STD > 2! GOTO platetemp

DiffT = ABS(Tavg - Told)
DiffSD = ABS(STD - STDold)
Told = Tavg
STDold = STD
IF DiffT > 1! OR DiffSD > 1! GOTO repeat
count% = count% + 1
IF count% < 2 THEN PRINT "Checked once, waiting 30 seconds to confirm"
pause:
finish! = TIMER
IF finish! - start! < 30 GOTO pause
IF count% < 2 THEN
PRINT " Rechecking"
GOTO repeat
END IF
Tsum# = 0!      'Rezero variables
Tssum# = 0!

```

```

    FOR i% = 1 TO t%
        TC(i%) = 0!
        Temp(i%) = 0!
    NEXT i%

END SUB

SUB TCALCULATE (Temp(), Tcal())
' Subroutine calculates the plate temperature at each heat flux gage
' location based on surrounding temperatures, either measured or
' previously calculated.

    Tcal(1) = .5 * (Temp(8) + Temp(12))
    Tcal(2) = .5 * (Temp(8) + Temp(12))
    Tcal(3) = .5 * (Temp(14) + Tcal(2))
    Tcal(4) = .5 * (Temp(16) + Tcal(3))
    Tcal(10) = .5 * (Temp(4) + Temp(11))
    Tcal(11) = .5 * (Temp(4) + Temp(11))
    Tcal(12) = .5 * (Temp(13) + Tcal(11))
    Tcal(13) = .5 * (Temp(15) + Tcal(12))

END SUB

SUB TEMPCONV (TC(), t%, Temp())
' Subroutine converts thermoelectric voltage (uV) from Type T
' thermocouples to degrees Celsius, using the Thermocouple
' Reference Tables based on the ITS-90, Omega Thermocouple
' Tables, Table A9.1, page A15. Only applicable for
' temperatures between 0 and 400 degrees Celsius.

' Constants
c0 = 0!
c1 = .025928
c2 = -7.602961E-07
c3 = 4.637791E-11
c4 = -2.165394E-15
c5 = 6.048144E-20
c6 = -7.293422E-25

TC(19) = TC(19) / .000001 'Convert to microvolts

FOR i% = 1 TO t%
    TC(i%) = TC(i%) / .000001 'Convert to microvolts
    TC(i%) = TC(i%) - TC(19) 'Subtract Ice Bath Reference
    Temp(i%) = c0 + c1 * TC(i%) + c2 * (TC(i%) ^ 2) + c3 * (TC(i%) ^ 3) + c4 * (TC(i%) ^ 4)
    Temp(i%) = Temp(i%) + c5 * (TC(i%) ^ 5) + c6 * (TC(i%) ^ 6)
NEXT i%

END SUB

SUB TEMPSET (TC())

```

' Subroutine allows the user to input a desired plate temperature
' and alerts the user when that temperature has been achieved

```

INPUT " Enter the desired plate temperature (degrees Celsius) "; Td
readplate:
PRINT #1, "OUTPUT 09;AF70 AL75"      'First and Last Channel
PRINT #1, "OUTPUT 09;AC70"
FOR i% = 1 TO 6
PRINT #1, "ENTER 09"
PRINT #1, "OUTPUT 09;AS"      'Step through channels
INPUT #2, TC(i%)      'Thermocouples #11 - #16
NEXT i%
PRINT #1, "OUTPUT 09;AC79"      'Thermocouple Reference Voltage, Ice Bath
PRINT #1, "ENTER 09"
INPUT #2, TC(19)
t% = 6      'Temperatures
Tsum# = 0!      'Sum of Temperatures
CALL TEMPCONV(TC(), t%, Temp())
FOR i% = 1 TO t%
d = Td - Temp(i%)
IF d > 1.5 OR d < -1.5 THEN
IF d < 2 THEN
PRINT "Difference between desired and actual temperature, "; d
PRINT "Adjust temperature, press any key to continue"
adjust:    d$ = INKEY$: IF d$ = "" GOTO adjust
start! = TIMER
PRINT "Sixty second delay "
delay:    finish! = TIMER: IF finish! - start! < 60 GOTO delay
END IF
GOTO readplate
END IF
Tsum# = Tsum# + Temp(i%)
NEXT i%
SOUND 1500, 10
SOUND 1700, 10
PRINT "Plate Temperature, "; Tsum# / t%; " degrees Celsius "
Tsum# = 0!      'Rezero variables
FOR i% = 1 TO t%
TC(i%) = 0!
Temp(i%) = 0!
NEXT i%
END SUB

```

APPENDIX D

Projected Average Air Velocities Inside Each Channel

Table D.1: Measured and Projected Average Air Velocities for each Channel Width.

Damper Position	Kiel Probe Average Velocity (ft/sec)	Vane Anemometer Average Velocity (ft/sec)	Projected Channel Average Velocity (ft/sec)
Channel Width, 15.75 cm (6.2 in)			$R^2 = 0.777$
Open	24.06	37.92	15.85
3/4 Open	13.52	31.91	13.33
1/2 Open	10.87	26.66	11.14
1/4 Open	6.52	12.92	5.40
1/8 Open		9.96	4.16
Plate "B"		4.17	1.74
Plate "A"		4.18	1.75
Channel Width, 28.70 cm (11.3 in)			$R^2 = 0.996$
Open	51.80	39.12	22.66
3/4 Open	44.35	33.85	19.99
1/2 Open	34.84	25.66	15.84
1/4 Open	20.97	12.34	9.10
1/8 Open		10.99	8.42
Plate "B"		5.44	5.62
Plate "A"		4.79	5.28
Channel Width, 43.69 cm (17.2 in)			$R^2 = 0.916$
Open	20.51	45.63	5.50
3/4 Open	14.90	37.90	4.51
1/2 Open	9.96	30.76	3.60
1/4 Open	5.75	13.00	1.33
1/8 Open		12.95	1.32
Plate "B"		4.71	0.27
Plate "A"		3.70	0.14

APPENDIX E

Comparasion of Theoretical and Experimental Natural Convection Nusselt Numbers

Nusselt Number as a function of Plate Temperature on the
Lower Surface of a Heated Plate for Natural Convection
Channel Width 28.70 cm (11.3 in)

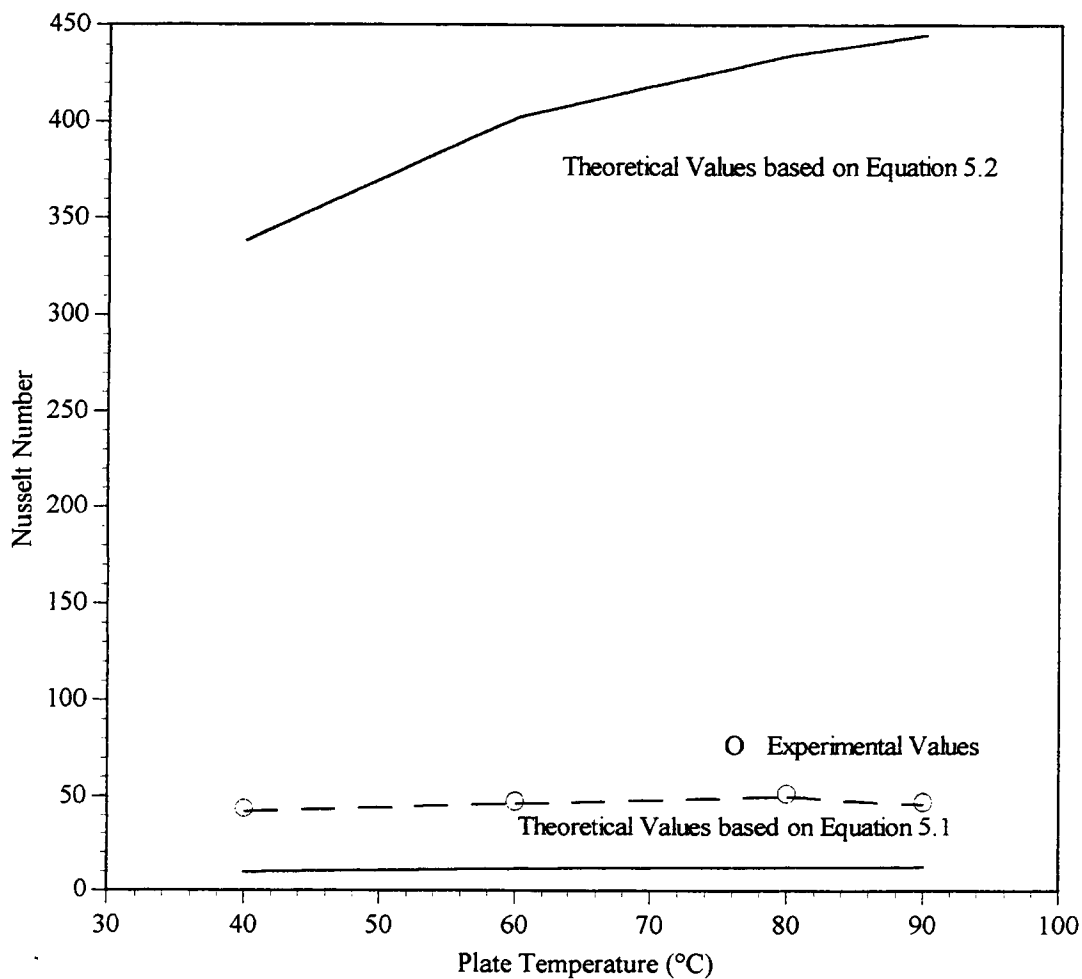


Figure E.1: Nusselt Number based on Natural Convection as a Function of the Plate Temperature for the Second Channel Width.

**Nusselt Number as a function of Plate Temperature on the
Lower Surface of a Heated Plate for Natural Convection
Channel Width 43.69 cm (17.2 in)**

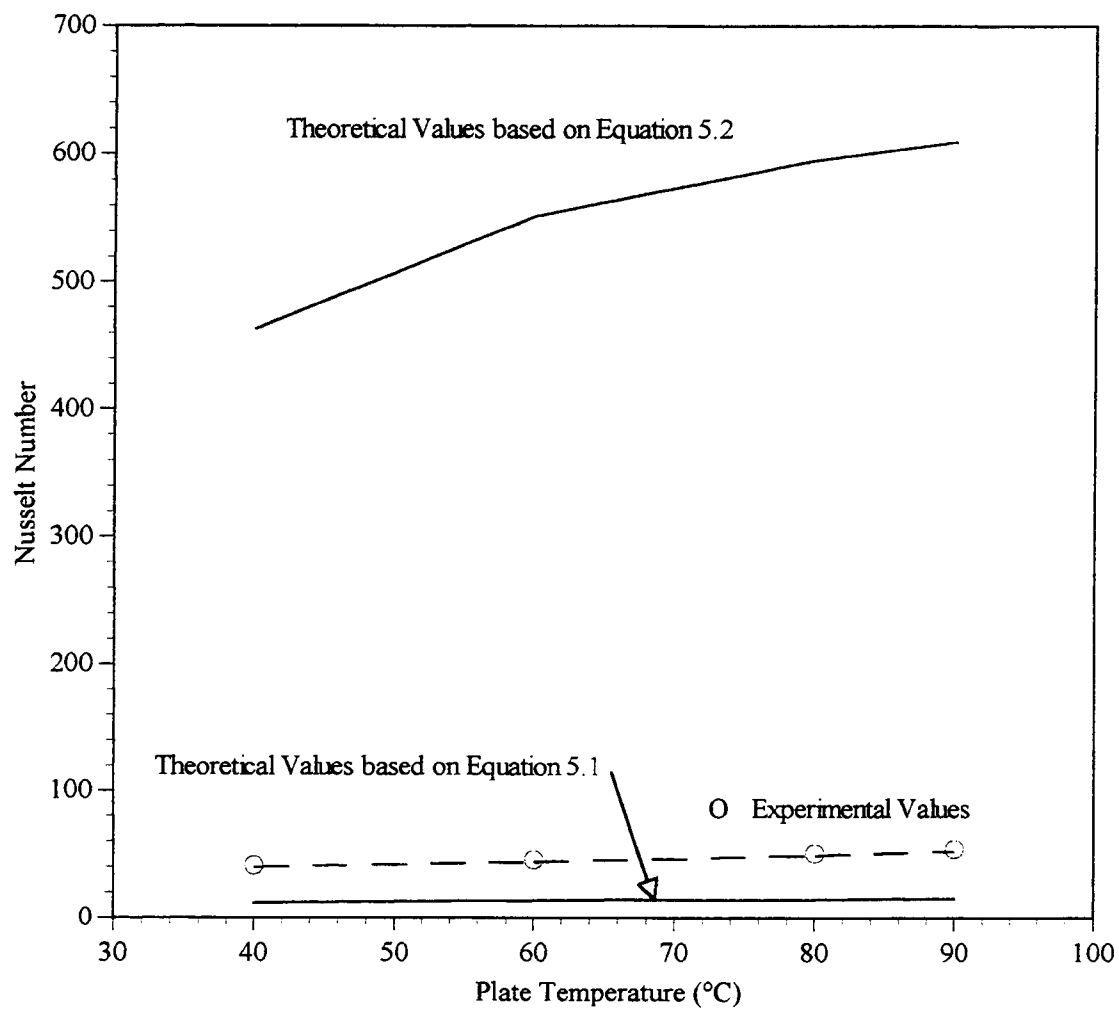


Figure E.2: Nusselt Number based on Natural Convection as a Function of the Plate Temperature for the Third Channel Width.

APPENDIX F

Uncertainty Analysis

The uncertainty of the Nusselt number ratio was based on the uncertainty of heat transfer coefficient, the hydraulic diameter of the channel and the thermal conductivity of the air. The uncertainty of the heat transfer coefficient was based on the uncertainty of the heat flux measurement and the uncertainty of the temperature of the copper plate and air using the root-sum-squares method of uncertainty analysis [Figliola and Beasley, 1991].

Dimensions of the first channel: Height: $a = 2.5 \cdot \text{in}$ Width: $b = 6.2 \cdot \text{in}$

Uncertainty in dimensions: $u_a = 0.125 \cdot \text{in}$ $u_b = 0.125 \cdot \text{in}$

Hydraulic Diameter: $D_h = \frac{2 \cdot a \cdot b}{a + b}$ $D_h = 0.091 \cdot \text{m}$

Uncertainty of the
Hydraulic Diameter: $u_{D,h} = \sqrt{\left[\frac{2 \cdot b^2}{(a + b)^2} \cdot u_a \right]^2 + \left[\frac{2 \cdot a^2}{(a + b)^2} \cdot u_b \right]^2}$ $u_{D,h} = 0.003 \cdot \text{m}$

Uncertainty in the temperature measurements was based on a bias error from the data acquisition unit, fifty measurements, a maximum standard deviation of 2 °C and ± 0.3 °C error range of the approximate inverse function for Type T thermocouples (0 °C to 400 °C) [NIST Monograph 175].

The bias error was based on the accuracy of the voltmeter and a thermal offset voltage.

Thermocouple voltage at
plate temperature of 90 °C. $E = 3814 \mu\text{V}$

Accuracy Error (0.007%): $E_{\text{accuracy}} = E \cdot 0.00007 + 5$ $E_{\text{accuracy}} = 5.267 \mu\text{V}$

Thermal Offset Voltage: $E_{\text{to}} = 1$

Voltage Error: $E_{\text{error}} = \sqrt{E_{\text{accuracy}}^2 + E_{\text{to}}^2}$ $E_{\text{error}} = 5.361 \mu\text{V}$

Constants for the approximate inverse function for Type T Thermocouples:

$$c_0 = 0.000 \quad c_1 = 2.592800 \cdot 10^{-2} \quad c_2 = -7.602961 \cdot 10^{-7} \quad c_3 = 4.637791 \cdot 10^{-11}$$

$$c_4 = -2.165394 \cdot 10^{-15} \quad c_5 = 6.048144 \cdot 10^{-20} \quad c_6 = -7.293422 \cdot 10^{-25}$$

Uncertainty in temperature based on voltage error (in microvolts):

$$T_{ve} = (c_0 + c_1 E_{error} + c_2 E_{error}^2 + c_3 E_{error}^3 + c_4 E_{error}^4 + c_5 E_{error}^5 + c_6 E_{error}^6) \cdot K$$

$$T_{ve} = 0.139 \cdot K$$

Variability of Ice Bath: $T_{ice} = 0.5 \cdot K$

Bias Error: $e_{bias} = \sqrt{T_{ve}^2 + T_{ice}^2} \quad e_{bias} = 0.519 \cdot K$

Number of measurements: $n = 50$ Maximum standard deviation: $S_T = 2 \cdot K$

Error Range: $e_r = 0.3 \cdot K$ Student t estimator (95% confidence level): $t_{49.95} = 2.0111$
[Figliola and Beasley, 1991]

Standard Deviation of the mean: $S_{t-} = \frac{S_T}{n^{0.5}} \quad S_{t-} = 0.283 \cdot K$

Uncertainty of temperature: $u_T = \sqrt{e_{bias}^2 + e_r^2 + (t_{49.95} \cdot S_{t-})^2}$

Plate Temperature (90 °C): $u_{Tplate} = 0.826 \cdot K$

Air Temperature (45 °C): $u_{Tair} = 0.826 \cdot K$

The thermal conductivity of air was calculated based on the average air temperature in the channel using an empirical formula [Figliola and Beasley, 1991].

Average air temperature in the channel: $T_{air} = 318 \cdot K$

Constants: $A = -2.450 \cdot \frac{W}{m \cdot K}$ $B = 0.1130 \cdot \frac{W}{m \cdot K^2}$ $C = -6.287 \cdot 10^{-5} \cdot \frac{W}{m \cdot K^3}$ $D = 1.891 \cdot 10^{-8} \cdot \frac{W}{m \cdot K^4}$

$$k_{air} = (A + B \cdot T_{air} + C \cdot T_{air}^2 + D \cdot T_{air}^3) \cdot 10^{-3} \quad k_{air} = 0.028 \cdot kg \cdot m \cdot sec^{-3} \cdot K^{-1}$$

Using a 95% confidence level (two times the standard deviation of the formula) and the uncertainty of the temperature.

Standard Deviation: $\sigma = 0.1198 \cdot 10^{-3} \cdot \frac{W}{m \cdot K}$

Uncertainty of the thermal conductivity of air.

$$u_{k,air} = \sqrt{(2 \cdot \sigma)^2 + \left[(B + 2 \cdot C \cdot T_{air} + 3 \cdot D \cdot T_{air}^2) \cdot 10^{-3} \cdot u_{Tair} \right]^2}$$

$$u_{k,air} = 2.483 \cdot 10^{-4} \cdot \text{kg} \cdot \text{m} \cdot \text{sec}^{-3} \cdot \text{K}^{-1}$$

The uncertainty of the heat flux measurements was estimated from the results of the repeatability tests performed at low air velocities and the maximum temperature difference (90 °C, plate temperature). At each air velocity four tests without infrasonic enhancement and four test with infrasonic enhancement were performed. An average heat flux value was obtained from the local heat flux at each of the six gage locations, 1200 total measurements (4 tests x 6 locations x 50 measurements per test).

Channel width: $b = 6.2 \cdot \text{in}$ Average Air Velocity: $v_{air} = 1.75 \cdot \frac{\text{ft}}{\text{sec}}$

Average Heat Flux Gage voltage without infrasound: $E_{hfg} = 866 \mu\text{V}$

Accuracy Error: $E_{accuracy} = E_{hfg} \cdot 0.00007 + 5$

Heat flux bias error assuming 4% calibration accuracy [Ortolano and Hines, 1983].

$$e_{hf,bias} = \sqrt{\left(\frac{3.1524}{3.212} \cdot E_{accuracy} \right)^2 + \left(\frac{E_{hfg}}{3.212} \cdot 0.04 \right)^2} \cdot \frac{W}{m^2} \quad e_{hf,bias} = 11.873 \cdot \text{kg} \cdot \text{sec}^{-3}$$

Student t estimator (95% confidence level): $t_{inf,95} = 1.960$ $N = 1200$
[Figliola and Beasley, 1991]

Average Air Temperature: $T_{air} = 318 \cdot \text{K}$ Plate Temperature: $T_{plate} = 363 \cdot \text{K}$

Heat Flux without Infrasound $q = 850 \cdot \frac{W}{m^2}$ $std = 370 \cdot \frac{W}{m^2}$ $S = \frac{std}{N^{0.5}} = 10.681 \cdot \text{kg} \cdot \text{sec}^{-3}$

The reproducibility (repeatability) of the heat flux gages was on the order of 1% [Ortolano and Hines, 1983]. $r = 0.01$

Uncertainty of heat flux measurement: $u_q = \sqrt{(e_{hf,bias})^2 + (r \cdot q)^2 + (t_{inf,95} \cdot S)^2}$

Heat flux without infrasound: $u_{qwo} = 25.524 \cdot \frac{W}{m^2}$

Heat flux with infrasound: $u_{qw} = 50.827 \cdot \frac{W}{m^2}$

The uncertainty of the heat transfer coefficient was estimated from Equation 4.1 and the uncertainty of the heat flux and temperature measurements.

Equation 4.1:
$$h := \frac{q}{T_{\text{plate}} - T_{\text{air}}}$$

Uncertainty of heat transfer coefficient:

$$u_h = \sqrt{\left[\frac{1}{(T_{\text{plate}} - T_{\text{air}})} \cdot u_q \right]^2 + \left[\frac{-q}{(T_{\text{plate}} - T_{\text{air}})^2} \cdot u_{T_{\text{plate}}} \right]^2 + \left[\frac{q}{(T_{\text{plate}} - T_{\text{air}})^2} \cdot u_{T_{\text{air}}} \right]^2}$$

Without infrasound:
$$u_{hwo} = 0.750 \cdot \frac{W}{m^2 \cdot K}$$

With infrasound:
$$u_{hw} = 1.433 \cdot \frac{W}{m^2 \cdot K}$$

Uncertainty of Nusselt number using Equation 4.2:
$$Nu = \frac{h \cdot D_h}{k_{\text{air}}}$$

Average Nusselt number without infrasound: $Nu_{wo} = 61.64$

Average Nusselt number with infrasound: $Nu_w = 110.952$

$$u_{Nu} = \sqrt{\left(\frac{D_h}{k_{\text{air}}} \cdot u_h \right)^2 + \left(\frac{h}{k_{\text{air}}} \cdot u_{D_h} \right)^2 + \left(h \cdot \frac{D_h}{k_{\text{air}}^2} \cdot u_{k_{\text{air}}} \right)^2}$$

Uncertainty of average Nusselt number without infrasound: $u_{Nu,wo} = 3.353$

Uncertainty of average Nusselt number with infrasound: $u_{Nu,w} = 6.238$

Nusselt number ratio defined as:
$$Nu_r = \frac{Nu_w}{Nu_{wo}}$$

Uncertainty of Nusselt number ratio:
$$u_{Nu,r} = \sqrt{\left(\frac{1}{Nu_{wo}} \cdot u_{Nu,w} \right)^2 + \left(\frac{Nu_w}{Nu_{wo}^2} \cdot u_{Nu,wo} \right)^2}$$

Percent Uncertainty of Nusselt number Ratio
$$u\%_{Nu,r} = \frac{u_{Nu,r}}{Nu_r} \quad u\%_{Nu,r} = 7.82\%$$

Similarly, the uncertainty of the Nusselt number ratio for the second and third channel widths was $\pm 8.47\%$ and $\pm 8.40\%$, respectively.

VITA

John M. Preston was born in Chattanooga, TN on December 15, 1970 and lived there until 1981, when he, along with his family moved to Knoxville, TN. In 1989 he graduated from Bearden Senior High School and enrolled in the mechanical engineering department at Clemson University in Clemson, SC. At Clemson he was selected to be Kodak Scholar during his freshman year and during his junior year he was elected president of his *Pi Tau Sigma* collegiate chapter. Mr. Preston graduated in May 1993, *cum laude* with Senior Departmental Honors.

Mr. Preston was awarded a teaching assistantship in Mechanical and Aerospace Engineering Department at the University of Tennessee, Knoxville and enrolled in August 1993 to pursue the M.S. degree. Following a short break after graduation he plans to work for Astec Industries, Inc. in Chattanooga, TN.