

BİLİM VE TEKNIĞE KATKILAR

EXPERIMENTAL INVESTIGATION OF UNSTEADY FORCED CONVECTION IN A CHANNEL WITH AND WITHOUT ARRAYS OF RECTANGULAR PROTRUDING SURFACES

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Abstract: An experimental study has been carried out to investigate the axial variation of inlet temperature and the impact of inlet frequency on decay indices in the thermal entrance region of a parallel plate channel, with and without arrays of rectangular modules attached to the lower channel wall. The modules (when used) are positioned in an inlined fashion to simulate the configuration most generally encountered in the circuit boards of electronic components. The investigation was conducted with both laminar and turbulent forced flows for Reynolds numbers ranging from 1,120 to 22,300. The geometric parameters were fixed at $H/2b=0.1875$, $S/2b=2.96$ for blocked duct, while the inlet heat input frequency varied from 0.02 Hz to 0.24 Hz. The results revealed that the thermal oscillations at different axial stations are periodic variations with approximately the imposed inlet frequency and further characterized with small, but increasing phase lags as downstream distance from the inlet increased.

(Key words: Unsteady forced convection, electronic cooling)

İÇİNDE DİKDÖRTGEN ÇIKINTILARI OLAN VEYA OLMAYAN KANALLARDA DEĞİŞKEN ŞARTLARDA ISI TAŞINIMININ DENEYSEL İNCELENMESİ

Özet: Paralel plakalı kanalın termal giriş bölgesinde giriş sıcaklığının aksel değişimi ve giriş frekansının etkisini araştırmak için önce pürüzsüz daha sonra da kanal içine dikdörtgen prizması şeklinde modüller yerleştirilerek deneyler yapılmıştır. Bu modüller, bilgisayarların devre boardlarında en çok karşılaşılan sıra şeklinde dizilmiş ve simüle edilmeye çalışılmıştır. Bu çalışmada Reynolds sayısı laminar ve türbülanslı akış için 1120-22300 arasında değiştirilmiştir. Giriş ısı frekansı 0.02 Hz - 0.24 Hz arasında değiştirilmektedir. Bloklü kanal için geometrik parametreler $H/2b=0.1875$, $S/2b=2.96$ 'dır. Farklı aksel noktalardaki sıcaklık osilasyonu giriş ısı frekansı gibi periyodik bir değişim şeklindedir. Girişte sıcaklık osilasyonu büyük olmakta ve daha sonraki noktalarda gittikçe küçülmektedir.

(Anahtar kelimeler: Daimi olmayan zorlanmış konveksiyon, elektronik elemanların soğutulması)

NOMENCLATURE

a	The width of the rectangular channel	[m]	D_e	Equivalent diameter of rectangular test section or hydraulic diameter	[m]
A_n	Calibration coefficients, defined by Eq. (7)		D_p	Diameter of PVC pipe	[m]
b	Half height of the rectangular channel	[m]	H	Height of blocks	[m]
C	Discharge coefficient, defined by Eq. (2)		m	Mass flow rate	[kg/s]
d	Diameter of the orifice plate	[m]	Pr	Prandtl number	

*Yazarlar daha önceki sayılarımızda tanıtılmıştır.

Re_p	Reynolds number in the PVC pipe, defined by Eq. (5)	
Re_t	Reynolds number in the test section, defined by Eq. (6)	
S	Space between blocks	[m]
T	Temperature	[°C]
V	Output voltage of thermocouple	[mV]
x	Axial distance	[m]
α	Dimensionless decay index, defined by Eq. (9)	
β	Inlet frequency	[Hz]
ε_1	Expansion factor, defined by Eq. (1)	
ϕ	Orifice plate diameter ratio, d/D_p	
ρ	Fluid density	[kg/m ³]
μ	Dynamic viscosity	[Pa s]
ΔP	Pressure drop across the orifice plate	[Pa]
ΔT	Centerline temperature amplitude, defined by Eq. (8)	[°C]
ΔT_i	Temperature amplitude at the center of the inlet	[°C]
θ_{amp}	Dimensionless temperature amplitude, defined by Eq. (9)	

1. INTRODUCTION

This paper describes an experimental study of temperature distribution in the thermal entrance region of a rectangular channel with arrays of inline rectangular blocks mounted on the lower channel wall. The main focus of the work is to present authentic experimental data on the nature of the temperature distribution along the centerline for a periodic inlet heat input at various frequencies and Reynolds numbers in this region. The investigated geometry bears similar characteristics to those employed in circuit boards of electronic equipment and the computer industry.

In practical engineering applications, the temperature distribution for both solid and fluid varies with time and the heat transfer inside the duct, for example, a general passage between two boards in the computer, may be exposed to a number of planned or unplanned transients during normal operation. The unsteady state can produce undesirable effects resulting in reduce thermal performance and increased thermal stresses, (with eventual mechanical failure). This phenomenon when present will affect the working condition of the computer chips placed on the circuit boards. Since these devices never attain steady state operation because of their nature to operate periodically in time, it is essential to know their transient response in order to provide an effective

control system.

A search of the literature revealed a number of studies related to this investigation. Few simplified solutions to applications involving unsteady forced convection in ducts had been solved by different researchers [1-4]. These solutions just provide a basic understanding of unsteady problems. Kakaç et al. [5-9] have reported a series of experimental and theoretical studies of unsteady forced convection inside smooth ducts. Li et al. [10] made an experimental study of unsteady forced convection in a duct with and without arrays of block-like electronic components, simulating printed electronic circuit boards. Recently, Sözbir et al. [11] made an analysis of unsteady forced flow in parallel plates channel with and without arrays of rectangular protruding surfaces. These studies were limited, based on the inlet heat input, block height and range of periodic inlet frequency.

In this paper, unsteady forced convection in a rectangular duct resulting from a timewise variation of the inlet temperature is experimentally studied. For the cases considered, the thermal responses of the system to the periodic variations has been evaluated after the initial transients have disappeared. Under such conditions, a general time dependent inlet condition can be expanded in terms of sine and cosine functions by use of Fourier series expansion. Based on this assumption, it is believed that results of sinusoidal variation of inlet temperature will provide a basic study with fundamental results for further research (theoretical) of unsteady forced convection under the general time dependent inlet condition.

In this study, the experiments are carried out in two phases. The first phase is performed with the channel as a smooth parallel-plates channel, while in the second phase, rectangular blocks of uniform size are placed on the base of the channel in the thermal entrance region, thereby simulating the electronic circuit board. In both cases, experiments covered laminar and turbulent flow regimes. The presentation of results is structured to show the effects of inlet frequency (β), the approach Reynolds number (Re), and the block dimensions on the temperature responses and the decay of the temperature amplitude along the duct.

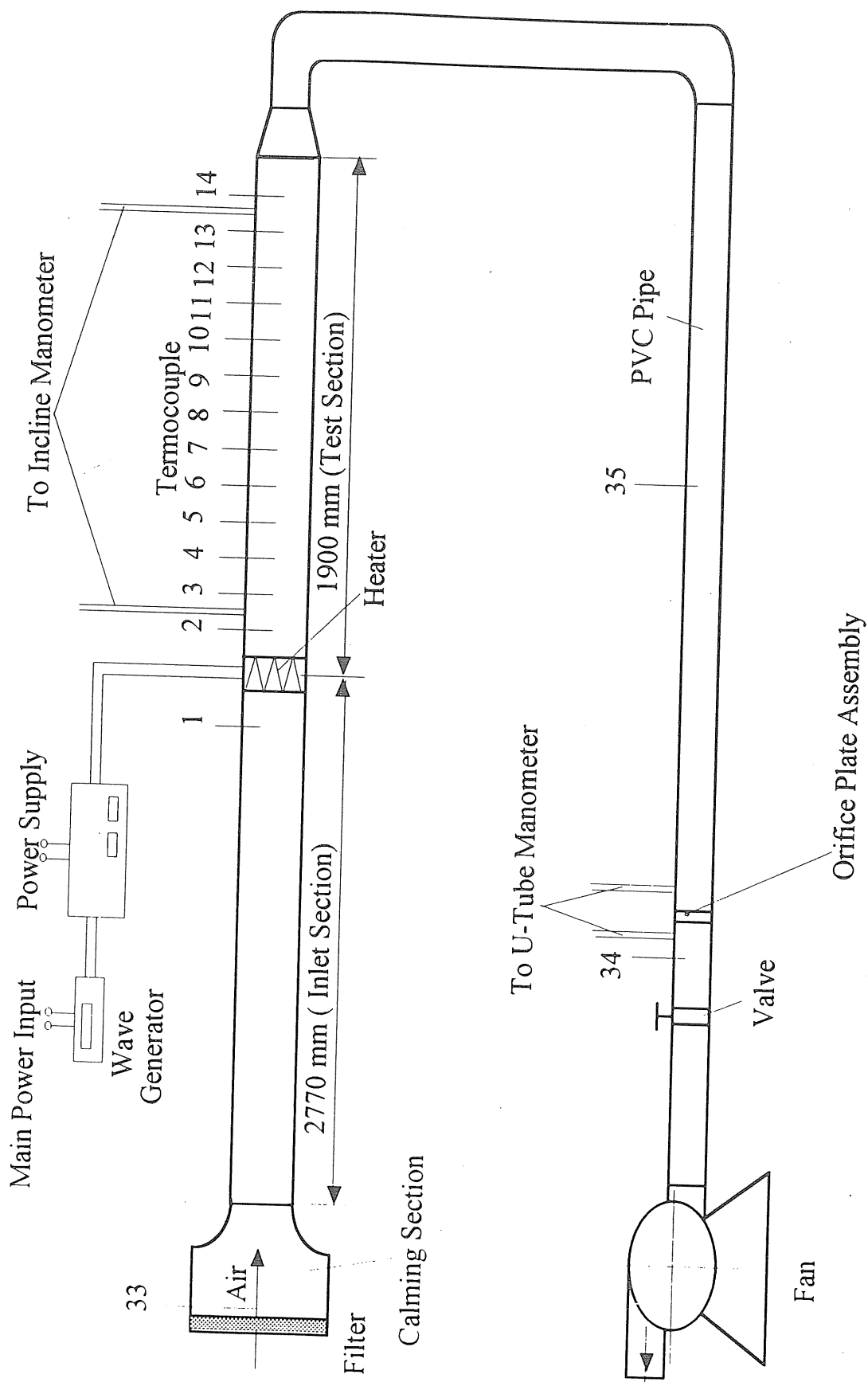


Figure 1. Schematic diagram of the experimental set-up.

2. EXPERIMENTS

The experiments were conducted in a system originally designed and built by Ding [12]. For this investigation, the basic geometry of the channel is a cross-sectional area of $254 \times 25.4 \text{ mm}^2$ ($10 \times 1 \text{ in.}^2$), while the blocks are $63.5 \times 38.1 \text{ mm}^2$ ($2.5 \times 1.5 \text{ in.}^2$) and 4.76 mm ($3/16 \text{ in.}$) in height. With the available set-up, it was possible to cover inlet frequencies from 0.02 Hz to 0.24 Hz and Reynolds numbers based on the mean inlet fluid velocity and maximum flow area of the duct, from 1,120 to 2,225 in laminar flow, and 11,240 to 22,286 in turbulent flow. A schematic diagram of the apparatus with the basic components of the experimental system is shown in Fig.1 while the dimensions of the test section and the arrangement of the blocks is shown in Fig.2.

The experimental apparatus was operated in the suction mode. The apparatus consisted of the rectangular duct, with different sections for filter, calming, inlet, test and convergence. The instrumentation included a wave generator, power supply, electric heater, inclined manometer, voltmeters, thermocouples, the circular section with an orifice plate to measure the mass flow rate and a centrifugal fan.

2.1. Experimental Apparatus

When air entered the apparatus, it first encountered an air filter, then flows into the calming section which was designed to provide a smooth air flow into the inlet section. The filter prevented dust and dirt particles from entering the test section. Air flows from the calming section to the inlet section of the duct ($2,770 \text{ mm}$ in length) where the velocity was allowed to become fully developed. Then the air flow then traveled through the heater and into the test section where the thermal characteristics were measured and recorded.

The duct was constructed with an outer casing made from 6.35 mm ($1/4 \text{ in.}$) plywood with outer dimensions of $114.3 \times 381 \times 4,670 \text{ mm}$ which incorporates the inlet and test sections. The inside of the casing was lined with 25.4 mm extruded styrofoam, leaving a cross-sectional flow area of $254 \times 25.4 \text{ mm}^2$ ($10 \times 1 \text{ in.}^2$). This part constitutes the heart of the experiment, since it is where the blocks are placed and the temperature variations are measured. The dimensions of the test section and the arrangement of the blocks are shown in Fig 2. The styrofoam was cemented to the inside of the casing to prevent air leakage or infiltration.

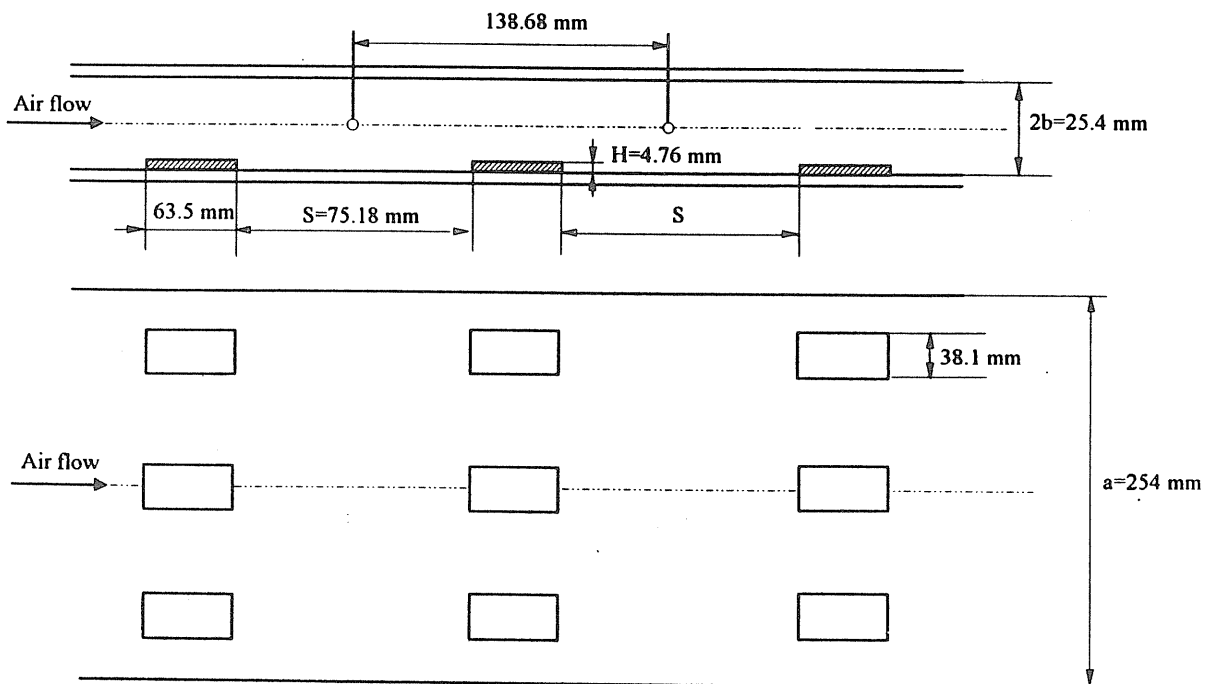


Figure 2. Geometric parameters for duct, array of blocks and thermocouple locations.

Throughout the study, periodic variations of heat input was provided by an electric heater constructed to fit into the duct, and to induce minimum disturbance to the air stream. For the assembly of the heater, 0.254 mm diameter nichrome resistance wire was used as the heating element, and was threaded between two plates of insulating material (transit board). The heater was composed of two sections each of which had 20.5 Ω of resistance at room temperature, and powered by a wave function generator and power supply.

At the end of the test section, the air flow entered a convergent section which changes from a rectangular duct upstream to a circular duct downstream. The circular duct was made from PVC pipe. The pipe (2,040 mm in length) was connected to the convergent section and a set of PVC companion flanges were used to house the orifice plate. Two different sizes of PVC pipe were used in order to obtain different flow regimes during the investigation. The one with a 101.6 mm (4 in.) nominal diameter was used for turbulent flow, and the other with 50.8 mm (2 in.) nominal diameter for laminar flow. On passing the companion flanges, another section of PVC pipe (840 mm or 33 in. in length) was used as the feed into the fan. This section also contains an adjustable damper which could be used for varying the flow rate.

2.2. Experimental Instrumentation

The key instruments of the sine variation of heat input to the heater at various frequencies are the wave function generator and the power supply. The function generator generated various wave forms, such as sine, triangle, square, ramp up, ramp down, haversine, havertriangle and DC. Sine wave form with DC offset was used in the present experiments. In the sine wave mode, the range of output voltage from the generator was from 0 to 1 volt, and by using the DC offset at the front panel, any output value within this range could be selected. The output from the generator was used as the control signal to the power supply. The sine wave was fed through a power supply, making it possible to generate the desired sine wave output to the electric heater. The output from the power supply was then fed to the electric heater, therefore, the current passing through the nichrome wire of the heater was controlled, while the voltage applied to the heater was dynamically determined by the heater impedance.

To measure the variation of temperature along the duct resulting from the sinusoidal heat input, thirteen thermocouples were placed at equal intervals along the test section, starting at the exit of electric heater. The thermocouples were made from 0.01 in. (0.254 mm) in diameter, teflon-coated chromel/constantan E-type thermocouples wires. Thermocouples were also placed at 5 pipe diameters downstream of the orifice plate and at the inlet of the duct to measure the outlet and inlet temperatures of the air stream, as the reference temperatures for the air density.

The mass flow rate was calculated from the pressure drop across the orifice plate constructed from brass, which was machined according to standard specifications given by ASME Standard [13]. The pressure tapings were located at D_p and $D_p/2$ from the faces of the orifice plate and connect through rubber tubing to an inclined manometer. Just downstream of the orifice plate is the location of the fan which provided air flow through the duct. The fan was equipped with a 0.54 kW-115 volts AC motor with a six-inch inlet and outlet diameters. Therefore, the piping just upstream containing the orifice plate was upsized by a connecting flange which changed the flow cross-section diameter to six inches. Once it passed through the fan, the air stream was routed out of the building by the use of flexible piping.

2.3. Experimental Procedure

During the course of the experimental investigations the controlled parameters were the duct geometry, inlet temperature, and Reynolds number. Duct geometry was changed from smooth duct to a channel with the arrays of blocks. For each duct geometry, a series of experiments were performed by changing the Reynolds number and inlet temperature frequency. For a specified inlet frequency, and Reynolds number, the variation of the temperature amplitudes at different locations along the duct were obtained at thirteen measurement points along with, temperatures at the entrance and downstream of the orifice plate. The data were recorded on a data acquisition while the dynamic thermocouple responses were recorded by an eight channel strip-chart recorder.

Prior to the start of each run, the positions of the thermocouples were checked while the voltage output of all thermocouples are examined for open circuitry. The pressure taps and tygon

tubing were also inspected to ensure proper connections. Steps of a typical experimental procedure can be stated as follows:

- (1) After the initial preparations mentioned above, the fan is turned on. The pressure drop across the orifice plate is adjusted to the desired range. The exact value of the mass flow rate and Reynolds number would also depend on the average temperature; therefore, they could be determined only after the data reduction.
- (2) The wave generator is then turned on and the output frequency adjusted to a desired value and stabilized. The sinusoidal variation is observed on the oscilloscope connected to the function generator.
- (3) After ensuring adequate air flow through the system, the power supply is turned on and its output voltage and current monitored. If periodic variation is not observed on the front panel, the potentiometer is readjusted.
- (4) After the heater is turned on, the temperatures at different thermocouple locations, especially the first thermocouple after the heater, are monitored since styrofoam plate cemented on the wall do not withstand temperatures above 85-95 °C for extended time.
- (5) Temperature amplitudes at various locations are checked to ensure that stability in the variation is reached. It is observed that usually after a period of 120 minutes, sustained periodic variation of temperature, could be reached.
- (6) After the steady state (the stabilization of temperature amplitude) is reached, all measurements from the thermocouples are recorded by a millivoltmeter through a rotary selection switch and the data acquisition system.
- (7) After recording the values from the voltmeter, the strip-chart recorder is turned on and traces the dynamic temperature responses along the duct.
- (8) The pressure drop across the orifice plate is rechecked and recorded on the acquisition form.

- (9) After the completion of data recordings (temperatures and pressure drops), the procedure is repeated, starting with step 2 to carry out experiments at the next desired inlet frequency. By releasing the stabilizer button of the wave generator, the frequency could be readjusted.
- (10) After experiments have been conducted for all desired values of the inlet frequency at a fixed Reynolds number, the experiments are repeated for another mass flow rate and then for different duct geometries.

3. DATA REDUCTION

After each experimental run, raw data consisting of millivolt readings from the thermocouples and pressure drop readings in mm of water column from the manometer, are recorded. Generally, raw data includes: (i) the pressure drop across the orifice plate, (ii) the static gage pressure at the upstream pressure tapping location; (iii) the average thermocouple emf's (in millivolt) for the air flow 5Dp downstream of the orifice plate; (iv) the maximum and minimum emf's of the thermocouples at all measuring points along the test section, and (v) the test section pressure drop. From this raw data, mass flow rate and Reynolds numbers are calculated, then the temperature amplitudes along the test section and corresponding decay indices in the test section are also determined.

3.1 Reynolds Numbers

Mass flow rate through the PVC pipe ($\dot{m}$) is calculated by the following expressions given by ASME Standards [13]:

$$\dot{m} = C \cdot \epsilon_1 \cdot \frac{\pi \cdot d^2}{4} \cdot \left(\frac{2 \cdot \rho_a \cdot \Delta P}{1 - \phi^4} \right)^{1/2} \quad (1)$$

where ϵ_1 is the expansion factor, which can be assumed as unity under our experimental conditions, d the orifice plate diameter, ϕ the orifice plate diameter ratio, ρ_a the density of air, ΔP the pressure drop across the orifice plate, and C the discharge coefficient expressed according to ASME Standards [13] as:

$$C = 0.5959 + 0.0312\phi^{2.1} - 0.1840\phi^8 + 0.039\phi^4(1-\phi^4)^{-1} - 0.01584\phi^3 + 91.71\phi^{2.5}Re_p^{-3/2} \quad (2)$$

The Reynolds number, Re_p , in the PVC pipe is then calculated as:

$$Re_p = \frac{4\dot{m}}{\pi \cdot \mu \cdot D_p} \quad (3)$$

where μ is the viscosity of air.

Once the experimental conditions and set-up geometries are specified, the Reynolds number, Re_p , in the PVC pipe can be determined since it appeared as a function of the mass flow rate and also a function of the discharge coefficient, C , as indicated by Eqs. (1) and (3). The discharge coefficient also appeared as a function of the Reynolds number, Re_p , as stated in Eq. (2). Combining Eqs. (1) and (3), the discharge coefficient may be stated as:

$$C = \frac{\mu \cdot D_p}{\epsilon_1 \cdot d^2} \left(\frac{2 \cdot \rho_a \cdot \Delta P}{1 - \phi^4} \right)^{-1/2} \cdot Re_p \quad (4)$$

Equation (4) represents a straight line while Eq. (2) represents an exponential curve. Thus, finding the solution of Eq. (2) and (4) is equivalent to finding the intersection (Re_p° , C°) for both curves. By utilizing an iteration procedure to obtain the solution of Re_p , it is then taken as the Reynolds number in the redevelopment section, and the mass flow rate, $\dot{m}$, is then determined from Eq. (3), which is rearranged and expressed as:

$$\dot{m} = \frac{\pi}{4} \cdot \mu \cdot D_p \cdot Re_p \quad (5)$$

while the Reynolds number in the test section, Re_t , is determined by employing the following relation:

$$Re_t = \frac{\pi \cdot D_p}{2(a+2b)} \cdot Re_p \quad (6)$$

where a and b are the width and half height of the rectangular channel, respectively.

3.2 Temperature Conversion

Temperature measurements are obtained from the thermal characteristics of standard beaded-type chromel-constantan (E-type) thermocouples. The relationship between the measured

thermocouple output in millivolts and the temperature in degrees Celsius is given by

$$T = \sum_{n=0}^9 A_n \cdot V^n \quad (7)$$

where V and T are, respectively, the output voltage of the thermocouple and the corresponding temperature in $^\circ\text{C}$, while A_n 's are the coefficients of the calibration curve fitting polynomial obtained, respectively, as [14]:

$$A_0 = -0.13077, A_1 = 20.04789, A_2 = -4.48241, A_3 = 1.91747, A_4 = -3.92654 \cdot 10^{-2}, A_5 = -0.17116, A_6 = 0.33795, A_7 = 1.25672 \cdot 10^{-3}, A_8 = -8.26621 \cdot 10^{-4} \text{ and } A_9 = 5.62857 \cdot 10^{-5}$$

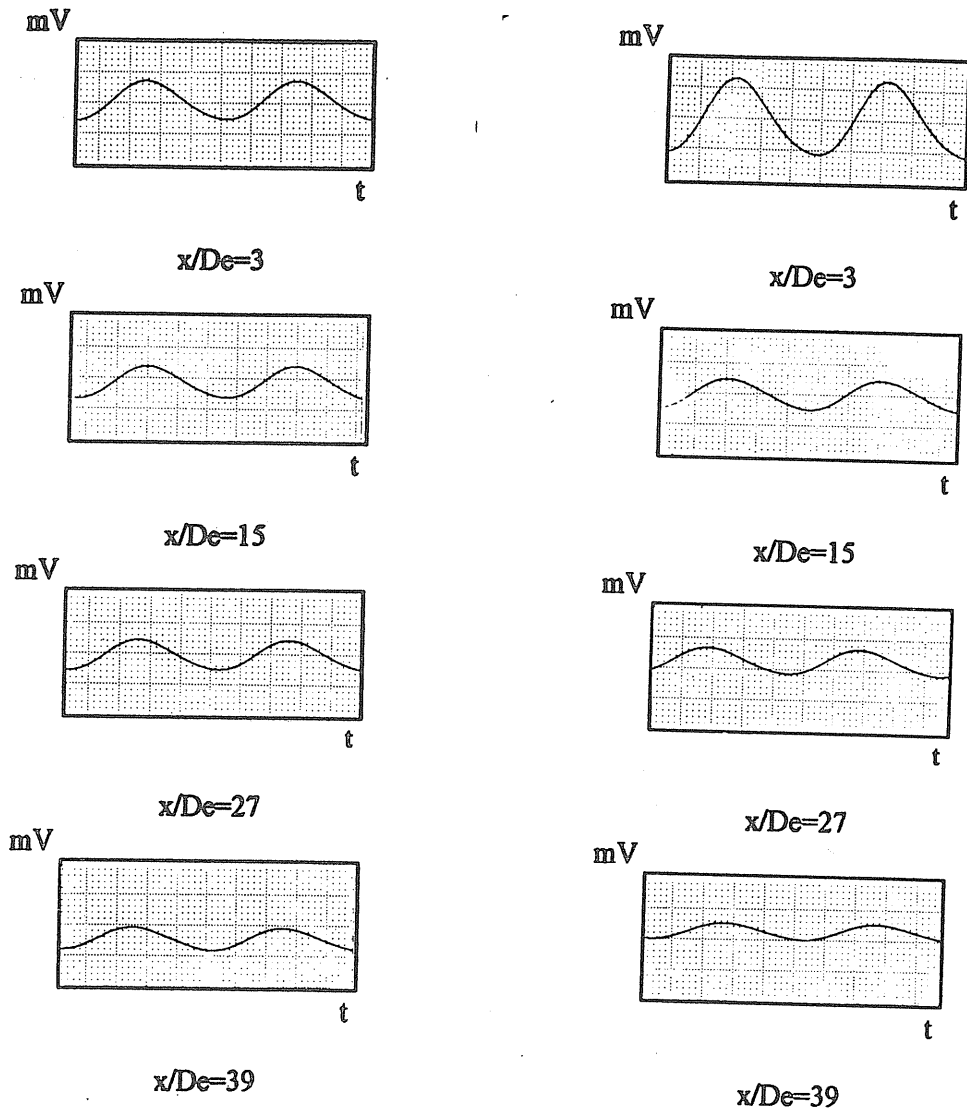
3.3. Experimental Uncertainties

The measurements made with E-type thermocouples usually have a maximum uncertainty of $\pm 0.2^\circ\text{C}$ within the temperature range of 0°C to 100°C . The uncertainty associated with the mass flow rate was affected by all of the parameters in the experiments, such as the pressure drop across the orifice plate, the inlet and outlet temperatures, the dimensionless of the experimental set-up, the calculations of all factors and coefficients and the thermal properties of the working fluid. The maximum value of the uncertainty in the mass flow rate was evaluated as $\pm 1.03\%$ and $\pm 2.63\%$ for laminar and turbulent flows, respectively, by Sözbir [14].

4. RESULTS AND DISCUSSIONS

4.1. Smooth Duct Experimental Results

In the first phase of the investigation, experiments were conducted in the smooth rectangular duct to compliment the main objectives of the studies. The temperature variation with time at predetermined locations were measured and recorded, a sample output of which is shown in Fig. 3. The thermal response is shown to be periodic in nature, fluctuating with a frequency approximately equivalent to that impose at the inlet and propagating with small but increasing phase lags between successive points as the downstream distance from the inlet increased. Amplitudes of minimum values of the thermocouple outputs, as



a) $Re \approx 22,280$, $\beta = 0.04$ Hz
 paper speed = 1 mm/sec.
 time scale: 1 sec./mm
 amplitude scale: $40 \mu V/\text{div}$.

b) $Re \approx 1,120$, $\beta = 0.04$ Hz,
 paper speed: 1 mm/sec.
 time scale: 1 sec./mm
 amplitude scale: $100 \mu V/\text{div}$.

Figure 3. Variation of centerline at different locations along the smooth duct.

$$\Delta T = \frac{T_{\max} - T_{\min}}{2} \quad (8)$$

In Figs. 4 and 7, the temperature amplitude values for thirteen thermocouple locations are plotted against dimensionless distance, x/D_e , for different inlet frequencies, β , and Reynolds numbers, Re . Figure 4 and 5 show the experimental

results for laminar flow, whereas Fig. 6 and 7 are the results for turbulent flow. The experimental data follow a straight line on these semilog plots, wherein regression lines of the experiment result are shown solid. Linear appearance of these plots is due to the exponential decay of temperature amplitude along the duct. The slopes directly gives the decay index, α , defined by as

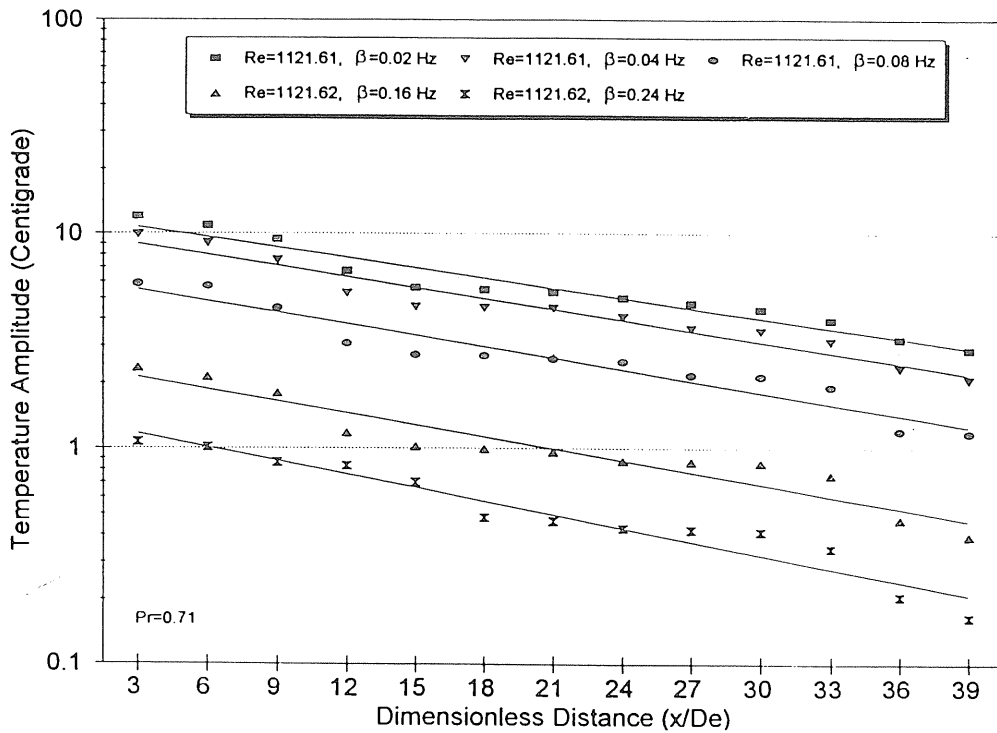


Figure 4. Decay of centerline temperature amplitudes along the smooth duct in laminar flow, $0.02 \text{ Hz} \leq \beta \leq 0.24 \text{ Hz}$, $Re \approx 1,120$ and $Pr = 0.71$.

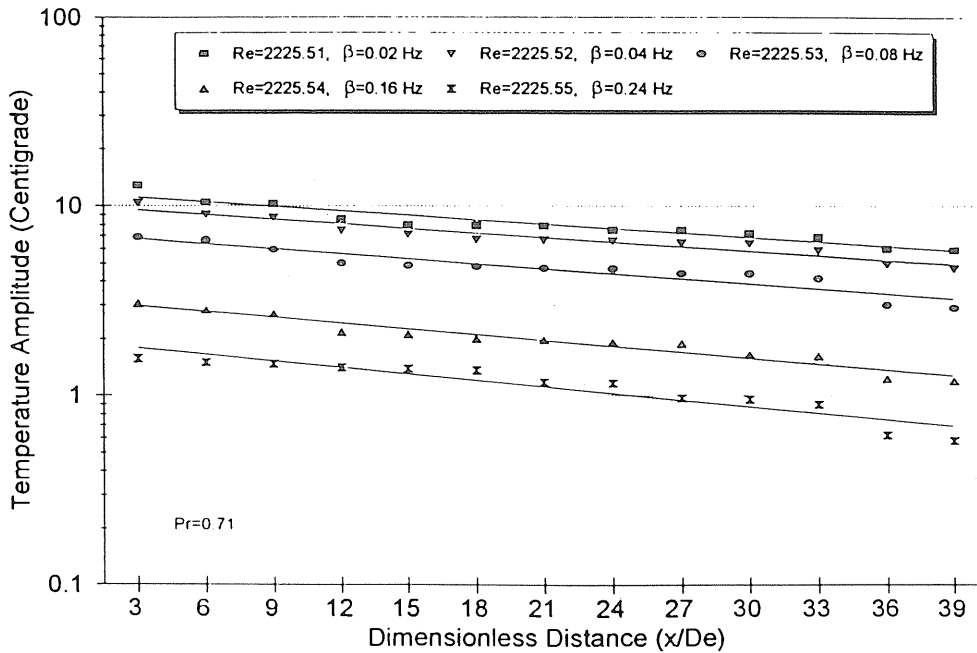


Figure 5. Decay of centerline temperature amplitudes along the smooth duct in laminar flow, $0.02 \text{ Hz} \leq \beta \leq 0.24 \text{ Hz}$, $Re \approx 2,120$ and $Pr = 0.71$.

$$\theta_{amp} = \frac{\Delta T}{\Delta T_i} = e^{-\alpha \cdot x/D_e} \quad (9)$$

values of α . This effect of inlet frequency is typically stronger in high Reynolds number flow as illustrated in Table 1.

As observed, higher inlet frequencies yield larger

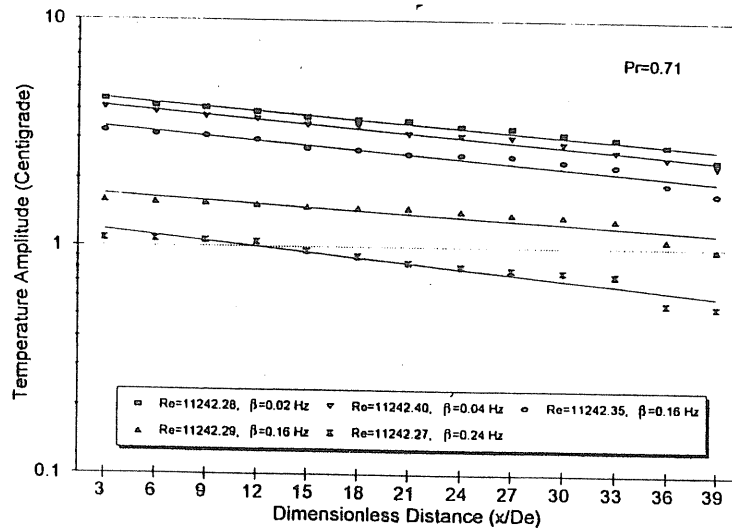


Figure 6. Decay of centerline temperature amplitudes along the smooth duct in turbulent flow, $0.02 \text{ Hz} \leq \beta \leq 0.24 \text{ Hz}$, $Re \approx 11,240$ and $Pr = 0.71$.

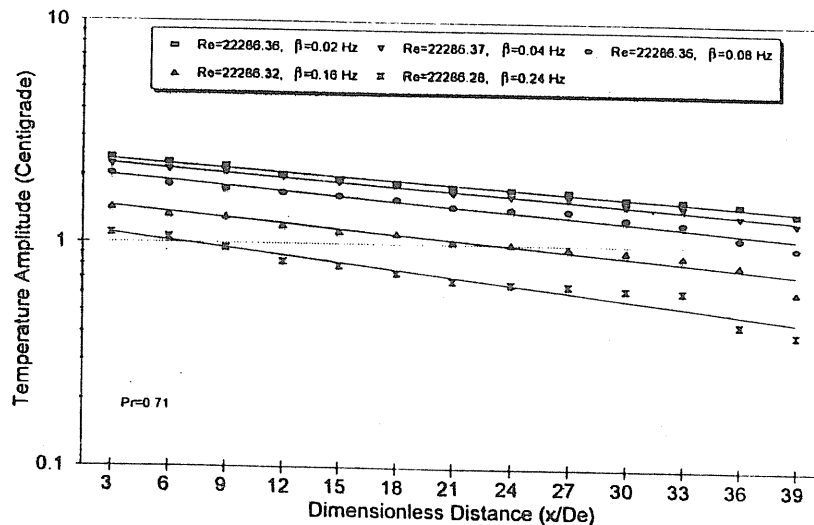


Figure 7. Decay of centerline temperature amplitudes along the smooth duct in turbulent flow, $0.02 \text{ Hz} \leq \beta \leq 0.24 \text{ Hz}$, $Re \approx 22,280$ and $Pr = 0.71$.

4.2. Experiments with Blocks

In this phase, inline arrays of rectangular blocks are placed on the bottom surface of the duct. To compare with the result for smooth duct, the air temperature in the maximum flow area sections, (in the center space between each row) are measured (see Fig.2).

The temperature variation obtained from the recordings is similar to those obtain in the case of smooth duct flows. A sample of the temperature variation with time for blocked duct is shown in Fig. 8. The axial temperature variation for duct

geometry ($H/2b=0.1875$, $S/2b=2.96$) at different inlet frequencies and Reynolds numbers are plotted in Figs. 9 and 12. The regression lines of the experimental results are shown solid in the figures. The linear appearance of these plots is shown in the semilog graphics illustrating a similar exponential decay of temperature amplitude along the duct as observed for the smooth duct cases.

The effects of inlet frequency on the decay index and Reynolds number are shown to exist in the same qualitative way as in the smooth duct cases. It is observed from Figs. 9 and 12 that for higher inlet frequency the temperature amplitudes

Table 1. Experimental decay indices for smooth duct.

Re	β (Hz)	α	Re	β (Hz)	α
1121.61	0.02	0.0365	11242.28	0.02	0.0140
1121.61	0.04	0.0392	11242.40	0.04	0.0149
1121.61	0.08	0.0410	11242.35	0.08	0.0150
1121.62	0.16	0.0424	11242.29	0.16	0.0108
1121.62	0.24	0.0475	11242.27	0.24	0.0184
2225.51	0.02	0.0178	22286.36	0.02	0.0137
2225.52	0.04	0.0184	22286.37	0.04	0.0150
2225.53	0.08	0.0203	22286.36	0.08	0.0171
2225.54	0.16	0.0235	22286.32	0.16	0.0183
2225.55	0.24	0.0263	22286.28	0.24	0.0242

decay faster along the duct, This can be interpreted as, the higher the inlet frequency, the larger the decay index. The decay in the centerline temperature amplitudes for all cases of the blocked duct are found to be faster when compared with the smooth duct cases, at similar Reynolds numbers and inlet frequencies. Therefore, the temperature oscillation decays faster within the blocked duct, because the augmented surface enhanced the heat transfer between the fluid and wall. Further comparisons are presented in Table 2.

5. CONCLUSION

Temperatures in ducts with and without blocks oscillate at approximately the same frequency as the inlet temperature oscillations. The temperature amplitude decays along the duct exponentially. Away from the thermal entrance, the decay of the amplitude is dominant by one term, since the experimental results are almost linear in the semilog plots.

Under the same conditions, the decay index for the blocked ducts appeared larger than that for the smooth duct. Thus, the amplitude of temperature oscillation in blocked ducts decays faster than that in the duct without block. The temperature oscillation decays faster at high inlet frequency than that at lower inlet frequency for both smooth and blocked ducts.

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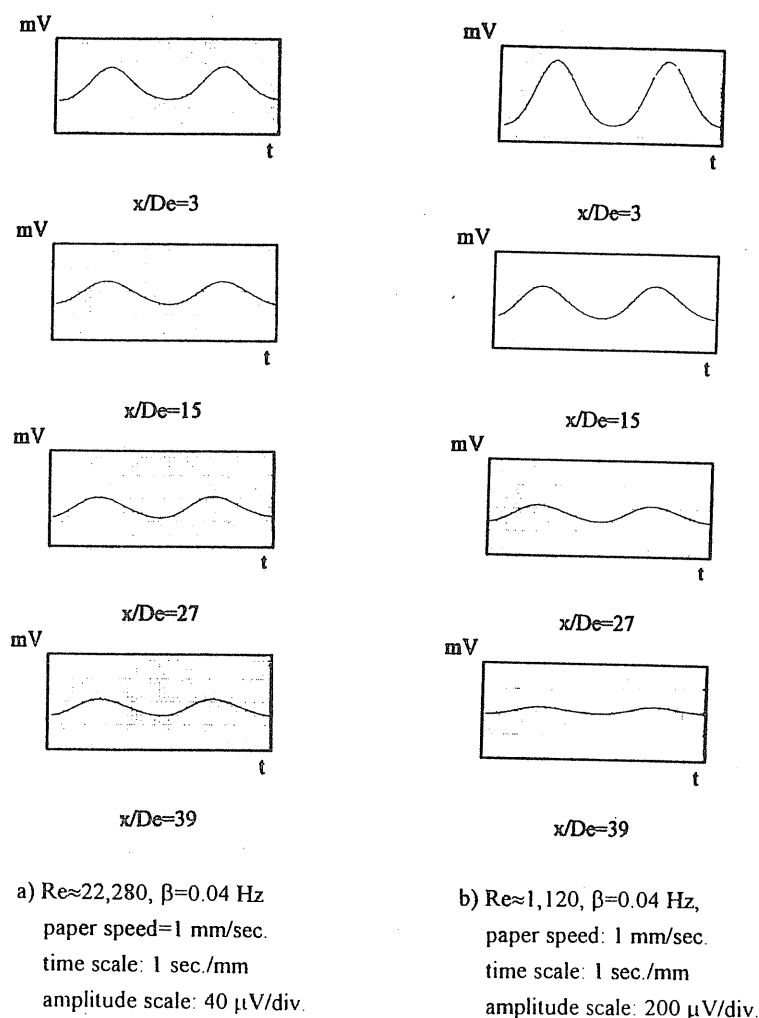


Figure 8. Variation of centerline temperature at different locations along the blocked duct.

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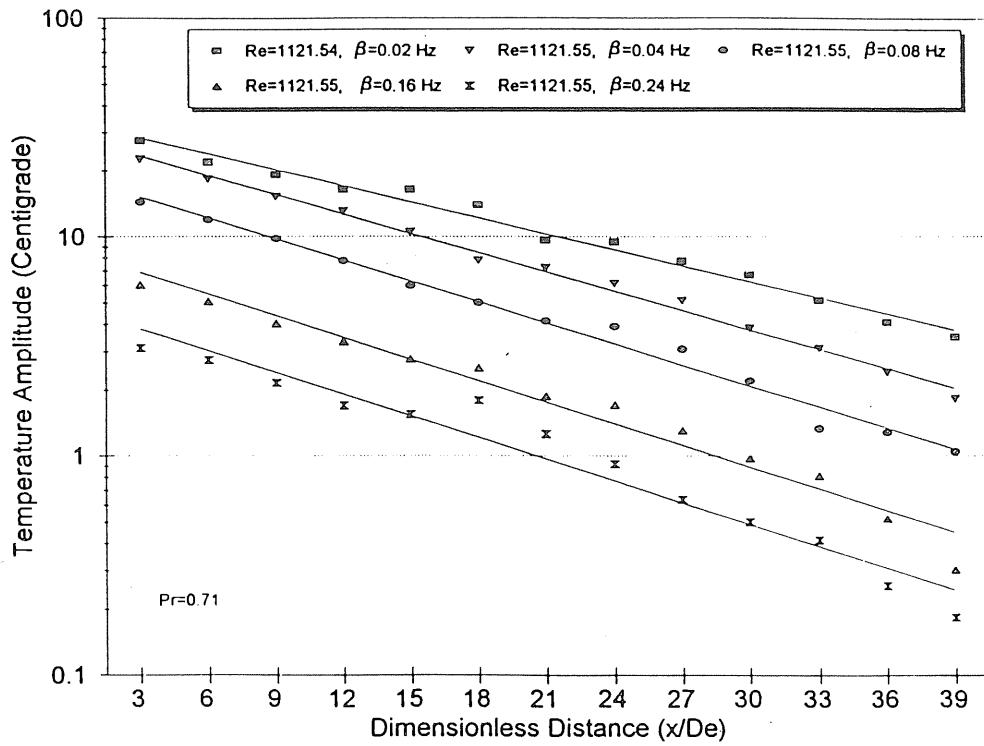


Figure 9. Decay of centerline temperature amplitudes along the blocked duct in laminar flow, $0.02 \text{ Hz} \leq \beta \leq 0.24 \text{ Hz}$, $Re \approx 1,120$ and $Pr = 0.71$, $H/2b = 0.1875$, $S/2b = 2.96$.

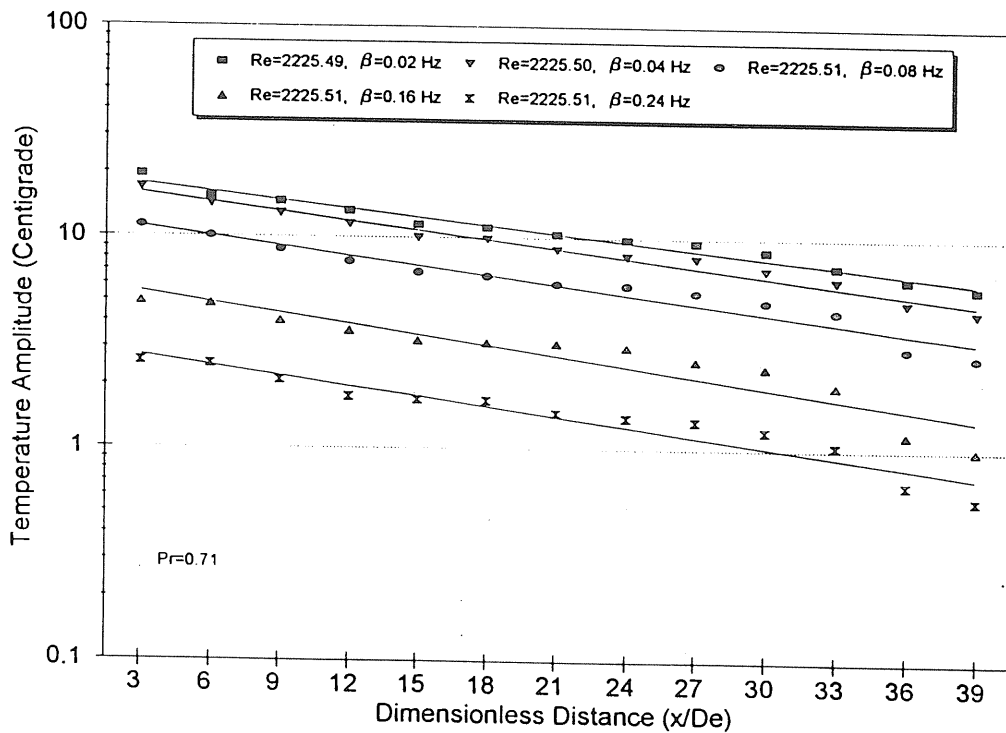


Figure 10. Decay of centerline temperature amplitudes along the blocked duct in laminar flow, $0.02 \text{ Hz} \leq \beta \leq 0.24 \text{ Hz}$, $Re \approx 2,120$ and $Pr = 0.71$, $H/2b = 0.1875$, $S/2b = 2.96$.

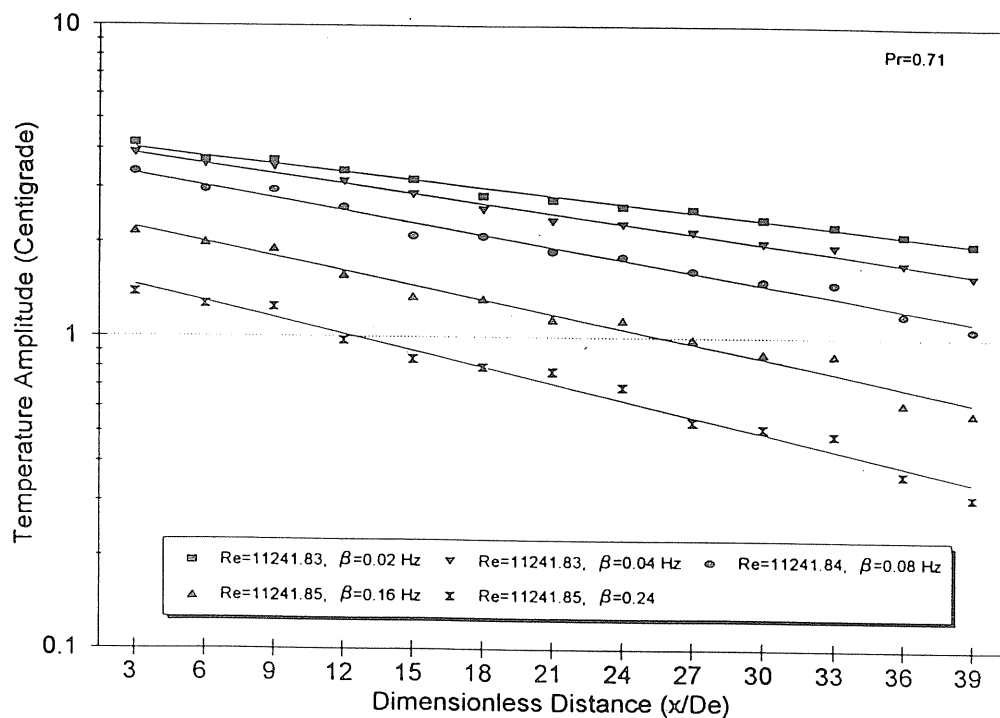


Figure 11. Decay of centerline temperature amplitudes along the blocked duct in turbulent flow, $0.02 \text{ Hz} \leq \beta \leq 0.24 \text{ Hz}$, $Re \approx 11,240$ and $Pr=0.71$, $H/2b=0.1875$, $S/2b=2.96$.

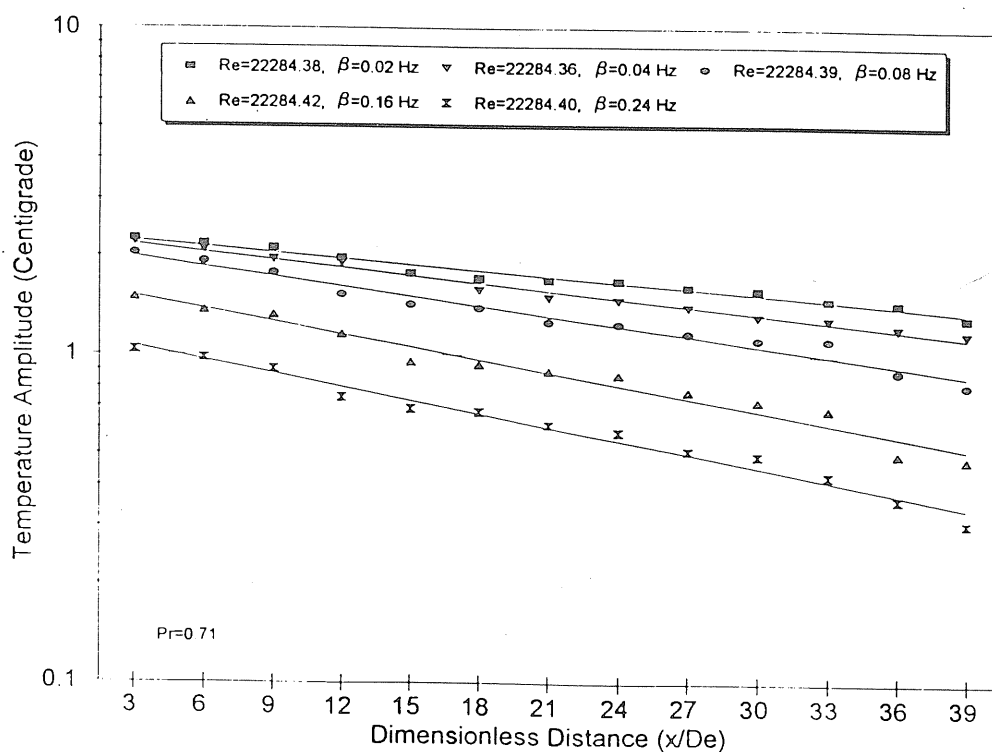


Figure 12. Decay of centerline temperature amplitudes along the blocked duct in turbulent flow, $0.02 \text{ Hz} \leq \beta \leq 0.24 \text{ Hz}$, $Re \approx 22,280$ and $Pr=0.71$, $H/2b=0.1875$, $S/2b=2.96$.

Table 2. Experimental decay indices for blocked duct with $H/2b=0.1875$, $S/2b=2.96$.

Re	β (Hz)	α	Re	β (Hz)	α
1121.54	0.02	0.0563	11241.83	0.02	0.0195
1121.55	0.04	0.0677	11241.83	0.04	0.0245
1121.55	0.08	0.0736	11241.84	0.08	0.0302
1121.55	0.16	0.0756	11241.85	0.16	0.0354
1121.55	0.24	0.0761	11241.85	0.24	0.0400
2225.49	0.02	0.0293	22284.38	0.02	0.0139
2225.50	0.04	0.0329	22284.36	0.04	0.0180
2225.51	0.08	0.0343	22284.39	0.08	0.0231
2225.51	0.16	0.0382	22284.42	0.16	0.0298
2225.51	0.24	0.0363	22284.40	0.24	0.0315

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