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AN EXPERIMENTAL STUDY OF FORCED VIBRATION ON NATURAL CONVECTION BETWEEN CLOSED ENDED CONCENTRIC AND ECCENTRIC ANNULAR OF HORIZONTAL CYLINDER

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Abstract

An experimental study has been done into the effects of vertical mechanical vibrating, vertical eccentricity, and the Rayleigh number on natural convection heat transferring out of a horizontally enclosed, ending cylindrical annulus with a radius rate of 2.6 and an aspect ratio of (2:1). The annulus produced between two concentric and vertically eccentric circular cylinders is positioned horizontally, and its internal wall is uniformly heated while isothermally cooling the external wall. The range of present conditions for Rayleigh number is 5×10^4 ≤ Ra≤ 6.48×10^6, and Pr = 0.703, the frequency of vibration is 0 ≤ f ≤ 20Hz; and the amplitude is b mm), with possible exclusion of the highest positive and negative eccentricities. Plots of the average Nusselt number variation against the Rayleigh number showed a significant increase in negative vertical eccentricity. It was found that the average Nusselt decreased as the internal cylinder changed its location vertically from negative to positive through the center, which is normally a desirable effect, but has no advantage over the concentric on the positive side. The Rayleigh number was found to be relatively sensitive to eccentricity. However, an increase of Rayleigh number leads to a nearly proportional increase in the average Nusselt number and a smaller yet still substantial increase in positive eccentricity. This study concluded that the vibration under the current experimental setup significantly affects the concentric position of the internal cylinder, whether the effect is positive or negative. The vibrational average Nusselt number increased in varying proportions, depending on the location of the heated inner cylinder.

Keywords: cylindrical annulus, concentric, vertically eccentric, vibration inner cylinder

List of Symbols/Acronyms

Ai – surface area of the heated vibrational inner cylinder $[m^2]$
b – amplitude [mm]
Di, Do – inner and outer cylinder diameter [m]
e – eccentricity [m]
ϵ – dimensionless eccentricity = $\frac{e}{L}$
g – gravitational acceleration $[\frac{m}{s^2}]$
h ,$\bar{h}$ – local and average heat transfer coefficient
K – thermal conductivity $[\frac{W}{m^2.℃}]$
l – axial length of the cylinder [m]
L – the gap width = (Ro-Ri) [m]
Nu, $\overline{Nu}$ – local and average Nusselt number
Pr – prandtle number $[\frac{\vartheta}{\alpha}]$
q`` – heat flux $[\frac{W}{m^2}]$
Ro, Ri – outer and the inner cylinder radius [m]
T – temperature [℃]
α – thermal diffusivity $[\frac{m^2}{s}]$
β – thermal expansion coefficient $[\frac{1}{℃}]$
ρ – density of the fluid $[\frac{kg}{m^3}]$
ω – natural frequency [rad/sec]

1. INTRODUCTION

The transfer of heat from a heated body to its boundary layer, accompanied by fluid motion produced by alterations in fluid density in the boundary layer, is known as natural convection. The boundary layer acts as an effective insulator to the transmission of thermal heat from a heated body to its surroundings. Therefore, the boundary layer must be thinned in order to augment the rate of heat transfer. The most popular techniques used to disturb the boundary layer are to decrease the boundary layer thickness, enhance the transverse fluid motion in the boundary layer, or both.

One of the active methods for disturbing the boundary layer is vibration, which thins the layer's thickness by mechanically generating turbulence. In recent years, the possibilities of tapping the vibration to augment the rate of heat transfer have concerned a great attention. Generally, vibration can be classified into two kinds of methos in that the heat transfer of the body or surface is vibrated and subdivided into two techniques. In the first method, the body is kept still while the oscillations are established in the flowing fluid around the boy. An oscillatory motion is subjected to the heated body itself, which represents the second technique. Both of these techniques aim to produce an oscillating relative velocity vector between a heated body and its fluid surrounding.

In many physical systems, the heat transferring by convection within fluid-filled is an importance mechanism. Many technical systems, such as heat, ventilation, air - conditioners, and many types of power production facilities, depend on heating exchangers. Many cutting-edge nuclear reactor ideas employ natural convection as the primary drive forces for coolant circulating, which has the added benefit of acting as emergent cooling systems in the case of a power outage (natural convection is exploited to avoid the demand of pumps). So, these applications along with the well-known design of modest dual pipe heating exchangers comprise of an annular duct, which will perfectly be centric, However, practically, several have some eccentricities brought on by producing tolerances, construction flaws, or thermic strains. Eccentricities all along the ducts might not be consistent due to pipe bending, end misaligned, or creep, but the current study is more interested in uniform eccentricity.

The literature concerning heat transfer natural convection involves several researches about the eccentric influence in concentric and vertical and horizontal eccentric annuli. Most of these studies are concerned with two dimensional, horizontal cylindrical annuli at either isothermal or constant heat flux boundary condition with either opened or closed ends. The number of researches concerning natural convection amid couple eccentric cylinders is comparatively low when compared to the concentric case. The annulus amid two vertical and horizontal eccentric cylinders has received the majority of attention within several experimentations as well as numerical researches about natural convection. Kuehn and Goldstein [1,2] provided a detailed overview of the pertinent literatures in 1976 and 1978. They used a constant radius rate of 2.6 and computed the results numerically that use the finite difference technique and empirically employ a Mach-Zehnder interferometers. They investigated how several factors, including the Rayleigh number, eccentricities, as well as fluid characteristics, affected heat convection in annuli. In the case of a vertical eccentric annulus, the natural convection has been examined by Chakrabarti et al. [3], Prusa and Yao [4], Habibi and Pop [5], to name a few, while the thermic domain within a horizontal eccentric annulus has been explored by Guj and Stella [6] and Guj et al. [7]. Hosseini and others. 2005[8] discovered a spontaneous airflow ratio induction for a vertically eccentric annulus. According to their experimental findings, the heat transmission is at its greatest for the investigation radius ratio at an ideal eccentricity of around 0.5. The researchers discovered that the tiny gap's heat transfer coefficient is significantly higher than the heat transferring factor of the nearby broader gap.

According to an experimental study on natural convection conducted by Eid El [9] in 2010, the eccentricity not only enhances the heat transferring ratio within two cylinders, but also boosts this ratio of heat transferring inside an annulus. Inside the concentric instance, the vertical eccentricity could increase free convection for around 15%, while the horizontal eccentricity could increase natural convection for around 10%.

It recognized that the physic modellings introduced within the present reports made simple to become a static system within a fixed wall. As far as the author is aware, the prior reports did not account for the influences of surface or body vibrating. Regarding actual applications, though, outside forces may cause the container or any other body to vibrate. Any thermal device will experience some vibration when it is operating normally. So, the vibration may be generated from the flow itself or from other components during its motion. For instance, inside an electronic system that has fan-cooled modules, the fan motor might induce wall vibrating, so this wall vibrating causes a considerable changing within the characteristics of the heat transfer coefficients along these surfaces. Unfortunately, the influences of the body vibrating upon natural convection heat transferring inside the containers have not adequately be studied, consequently the relevant data are still missing.

The influence of vibration upon the convective heat transfer has been investigated for flat plates, cylinders, different geometries of cavities, for varied vibration orientation vector relative to these surfaces, various vibration intensities “the amplitude and frequency”, and for various surface heating conditions. The observations from these results varied from a large and significant effect to small or even decrease the rate of heat transfer. No analytical results are yet available for predicting all the experimentally observed effects.

A considerable body of literature has been written about the general issue of the interaction between vibration and convective heat transferring. The survey results have been reported on free convection from various geometries. The previous investigation about the vibrational influences upon heat transferring of natural convection was implemented vi Lemlich [10](1955), that conducted an experimentation investigation on the electrical heated cables. The outcomes revealed that there is a great influence for amplitudes and frequencies compared to the influence of different elements, while no significant effect was noticed when changing the vibrational orientation. Forbes [11] (1970) carried out an empirical research pertaining the effect of mechanical vibrations on heats’ transferring free convection in a rectangular section enclosure which filled with water as working fluid. The outcomes showed that the fluid column embedded within the enclosure has significant effects on the heat transferring features at frequencies close to the resonant natural frequency. About 60% of heat transfer, which represents the maximum augmentation, was obtained. Dawood et al. (1981)[12] and Kimoto et al.(1983) [13] experimentally investigated the influence of vertical vibration on natural heat convection from a horizontal cylinder in the still air. The conclusion from the results observed that the vibration augments the heat transferring rate with about three times by the comparison with the stationary case. In an experimental investigation about the improvement of heat transferring ratio amid two coaxial cylinders, Ivanova and Kozlov[14],(1988) discovered that up to 600 non-dimensional frequencies result in an increase within the ratio of heat transferring. There is a slight reduction in heat transferring rate after a non-dimensional frequency of 600, which the authors noted this. Murphy and Lambert [15] (2000) experimentally investigated the influence of transverse forced vibration on the rate of natural heat convection in the horizontal heated cylinder. The conclusion drawn from these experiments was that the cylinder under vibration and at natural shedding frequency was able to achieve its maximum Nusselt number. Park and Gharib[16](2001) experimentally scrutinized the heat transfer from a stationary and forced vibrating cylinder in cross flow. They observed that at a frequency reached to the natural vortex shedding frequency, the high rate of heat transfer was achieved at these frequencies. Zhang et al[17] .'s (2004) numerical research of laminar natural convection was conducted upon a vertical flatten plate that was vibrating regularly and being heated uniformly. The findings demonstrated that the oscillating plate's ability to transmit heat relied heavily on the ratio among oscillating velocity and flowing velocity at the plate's boundary layers, the greater this ratio, the higher heat transfer. But the heat transfer decreases as Grashof number increases. Sarhan [18] (2019) experimentally investigated the influence of sinusoidal vertical vibration and orientation (a horizon as well as an incline from horizon within angles 30°,45°,and 60°) of flatten plate heat sink underneath natural convection. The amplitude of vibration was varied from (1.5 to 7.5 mm) and the frequency was varied (0 to 16 Hz) at constant heat flux. The oscillations of the fins were controlled by a tiny actuator to reduce drag in turbulent flows and their contribution to heat transfer. It was observed that as the oscillation frequencies increased, this leads to increase the rate of transferring heat and the maximum increase in the horizontal position and (16Hz).

The overview of the above introduction shows that the vibrational effect on the natural convection heat transfer was evident in studies that examined flat plates in different orientations and inclination angles, wires, enclosures with different geometries, aspect ratios, and boundary conditions of heating, as well as various orientations of shedding vibration compared to such surfaces with different ranging of vibration parameters (amplitude and frequency). Further, there are currently no experiments available for learning more about how vertical sinusoidal vibration interacts with steady, laminar natural convection inside of concentric, vertically eccentric cylinders that are bounded by two horizontal cylinders, accordingly, this is the objective of the current work. Heating the annulus internal wall has been at uniform heat flux, whereas cooling the external wall that was kept at fixed temperature. Besides the eccentricity (the effect of the inner cylinder position within the outer cylinder), heat flux, the influence of vibration parameters (the amplitude and frequency) on the air at annulus between the two cylinders were utilized.

2. THE EXPERIMENTATION SETUP DESCRIPTION

2.1. The suggested analytical solution

The experimentation setup comprises essentially of annular space formed by two concentric and vertically eccentric between two horizontal cylinders and closed by two rubber sheet at each end, and mounted on iron frame. The experimental setup can be divided into six main parts which consist of an annular test section formed between two long horizontal cylinders, heater circuit, closed water circuit, vibration excitation system, temperature thermistor system, and measuring system. Figure 1 illustrates the specimens used in experimental rig system.Solid copper has been used to machine the heated and cold cylinders. The diameter of the outer cylinder is Do=10.4 cm, while the diameter of internal cylinder is Di=4 cm, so that these cylinders form an annular gap that has a radius ratio $R = ro/ri = 2.6$, aspect ratio of $l/Di = 2:1$ with the gap length (radius difference) to the inner diameter $\frac{L=(ro-ri)}{Di} = 0.8$, and the axial length $l = 20\ cm$ for two cylinders. Teflon rods with the same diameter of the inner cylinder are used to insulate the ends of heater and prevent heat loss from the faces of heater.

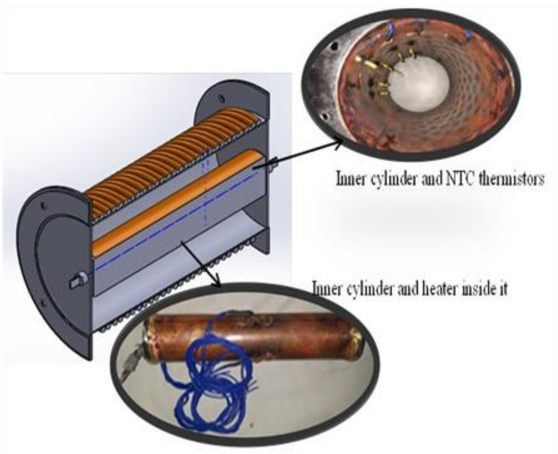

Fig. 1. Specimens used in the experimental tests

The annular space between the cylinders is closed at two ends by flanges, made from insulating material by rubber sheet to prevent the heat losses and to prevent air leakage. Each insulating rubber sheet consists of one hole at each end in the center which is used to change the inner cylinder position and change its position within the outer cylinder and achieve the vertical eccentricity by using stud bolt. The vibration rig is designed and built, where forms from two iron arms welded on the iron plate with dimension (27*3 cm) length and width. The arms hold the heated inner cylinder and used to transmit the vibration to it.

The iron plate is attached to the holder with U form (from iron) using a bolt and shaft from the concave. The U form, consequently, grips the two arms by the centered bolt at the heated inner cylinder, so as to permit the inner cylinder to vibrate vertically. At the same time, the two arms have grooves like the channel at the top end at each of these arms, which tapped to change the inner cylinder positions. The U form is installed on vibrator exciter by screw M10, through it the vibration is transmitted and passed to the heated inner cylinder through the U shape. The iron frame is used to hold and fix the two copper cylinders and their components and tighten them together by using bolts, it also used to limit the on the side movements of the testing section and to permit any up-down movements so as to merely insure the application of a vertical vibrating upon the plate with no side vibrating. For the purpose of absorbing any backlash vibration from the bottom of apparatus vibration, four rubber washers are placed between the wooden layer and ground by using stand bolts. The vibrational system has a high sensitivity due to its precise components and different demands for generating and measuring the vibrating factors. The vibration system consists of a functional generator, a power amplifying device, vibrating exciter, as well as a vibrating measure. To match such components with the testing section and to be fitted with work demands and circumstances, the vibrating rig is produced to satisfy such demands and to solve some difficulties in the work. Figure 2 illustrates the vibration rig system.

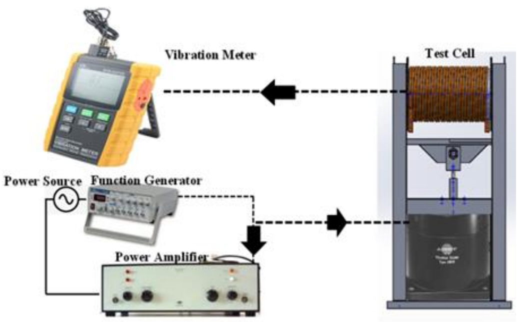

Fig. 2. Test cell with vibration instruments connection scheme

The inner cylinder was heated uniformly by a cartridge heater that was inserted along the axial length from right to left. Voltmeter-ammeter and a variance (transformer) can be used to adjust and test the power of a heater.

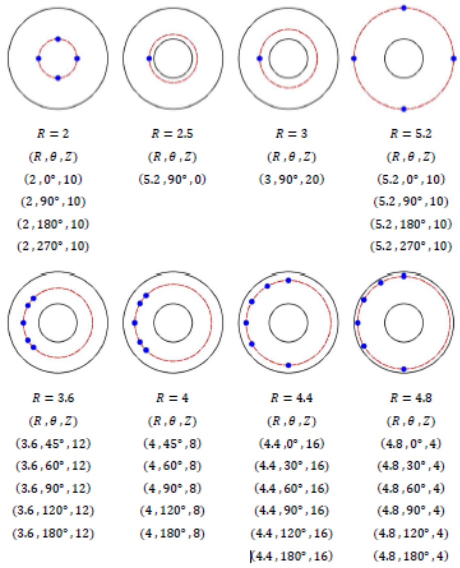

Fig. 3. Schematic representation for NTC thermistors and their fixing on the inner and outer cylinders

A 32NTC thermistor with a temperature range from (-40˚ to 130 ˚C) was positioned. The NTC thermistor was arranged to measure the temperature along the radial from the heated inner cylinder to the colder outer cylinder along the gap between two cylinders. Four NTC thermistors were spaced 90° in the mid axial distance and got placed in about 2 mm from the inner cylinder external surface. Each NTC along with heater wire was passed through a hole nearby one end. Four NTC thermistors got positioned within 1mm inside the surface; they were situated in the same manner of those inside the internal cylinder. The right half of the cylinder was tapped to fix each half since it was assumed that, due to symmetry, the distribution of wall temperature for each half of the cylinder will be equal, disregarding the gap's influence and placing no thermistor. At radius equal to 4.8 and 4.4 cm, six thermistors were positioned at different angles. Whereas at (R=4, and 3.6 cm), five thermistors were placed at each radius, four were placed at (R=5.2 cm), and one was placed at (R=3, and 2.5 cm). Figure (3): Schematic representation for NTC thermistors and their fixing on the inner and outer cylinders. All NTC thermistors were calibrated at room temperature using Distilled water and ethanol alcohol, where the thermistors are put in melting ice, boiling water, and boiling alcohol. The melting and boiling point of distilled water are 0 and 100℃ respectively, and the boiling point of ethanol alcohol is 78.4℃. the thermistors are connected to the data acquisition and computer.

A data acquisition board from National Instruments, model NIUSB-6259, with a 16-bit

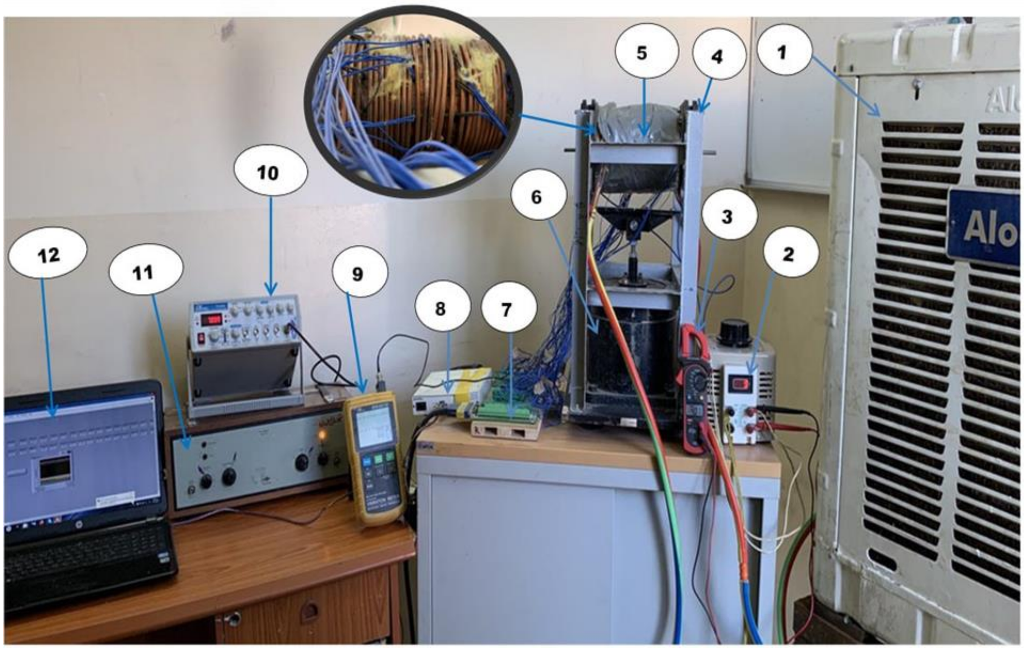

Fig. 4. The experimental test rig. 1:Closed cooling water circuit, 2:Variac, 3:Digital clamp meter, 4:Iron structure, 5:Test section, 6: Vibration exciter, 7:Electric card and its connection wires, 8:National instrument data acquisition, 9: Vibration meter, 10: Function generator, 11: power amplifier, 12: Personal computer (PC)

resolution, 32 ports, and input electric power of 11–30 V DC and 50 W, was used to record NTC thermistor readings at the selected spots in the test cylinder section. Coaxial wire, electronic cards, and a control measuring program built in the LabVIEW software were used to record temperature information. The readings were recorded each 1 min. A closed water circuit included a unit of evaporating air cooling system EACs 3750 CMH, two copper flow channels, and flexible hose. The water was driven by a circulation pump (water pump) (H=6-10m), Q flow rate=2-4m^3/hr , N angular velocity=2850 rpm, connected with the feed line of the water reservoir in EACs. A feeding water reservoir $1m^3$ was provided with a floater to adjust the water level. The large size of the reservoir (thermo-regulated mixing tank) was used to offer an adjustable temperature level of the water jacket within ± 0.01℃ during the experiments. Figure 4 shows the experimental rig.

3. MEASUREMENT PROCEDURE

The experimental procedure included two main parts:

a. Natural convection at the absence of vibration which conducted for natural convection without vibration.

1. Initially, the inner cylinder is placed and fixed in a specified location relative to the outer cylinder.
2. The steady state condition was reached after left more than four hours.
3. When NTC thermistors gave two sequential readings that were comparable, the thermistor's output reading was taken.
4. As previously stated, 32 NTC thermistors were arranged at a symmetry points on the surface of the inner, outer cylinders, and in annulus between the two cylinders.
5. To indicate the average temperature of the heated inner cylinder, the average temperature of the fluid trapped in annulus between the two hot inner cylinder and the cold outer cylinder was determined.

This method was performed for concentric and all eccentric positions of the inner cylinder employed in this experiment in order to collect comprehensive information concerning the natural convection by way of vibrating.

b. The second testing was performed for the influence of the vibration of the heated inner cylinder with natural convection:

1. The electrical heater and a cooling circuit are switched on.
2. A digital oscilloscope was used to calibrate the generator frequency initially. It had then been accustomed to the appropriate level. After that, a power amplifier was used to obtain the produced signal and amplified it.
3. The heated inner cylinder was left at each frequency for at least 15 minutes before moving on to the next frequency until all frequency ranges in the experiments were applied.
4. The heater's electrical power was increased to cover a second run, to achieve steady state conditions for the the next heat flux, and for the same set of frequency that were previously employed in studies to demonstrate its influence on temperature.
5. During each experiment, the following readings were recorded:

a. The heater current in Ampere.
b. The voltage in volt.
c. The amplitudes in (mm), and frequency in (Hz).

d. Rerun the procedure outlined above using a different position for the heated inner cylinder and the same heat flux and frequency settings.

4. DATA ANALYSIS

The mean temperature rising of the heated inner gets calculated by the equation below:

$$\Delta T_{ave} = \frac{\sum_{j=1}^{N} Tj}{N} - T_f = \overline{T_s} - \overline{T_f} \quad (1)$$

Where, N=4

$$\overline{T_f} = \frac{\overline{T_b}+To}{2} \quad (2)$$

Where, $\overline{T_b}$ refers to the mean bulk fluid temperature assessed at the outlet and the bulk mean fluid (air) evaluated at outlet inside the annuli, as the procedure outlined in [19].

Grashof and Rayleigh numbers are calculated from the following equation at a constant heat flux

$$Gr = \frac{\rho^2 g_o \beta q" L^4}{K_f\, \mu^2} = \frac{g_o \beta q" L^4}{k_{cond}\, \vartheta^2} \quad (3)$$

$$Ra = \frac{\rho^2 g_o \beta q" L^4}{K_f\, \mu^2} Pr = \frac{g_o \beta q" L^4}{k_{cond}\, \vartheta^2} \quad (4)$$

The vibrational Rayleigh number

$$Ra_{vib} = \frac{1}{2}\left[\frac{\rho^2 (b\Omega) \beta q" L^2}{K_f\, \mu^2}\right]^2 Pr = \frac{1}{2}\left[\frac{(b\Omega) \beta q" L^2}{K_f\, \vartheta^2}\right]^2 \quad (5)$$

The average heat transfer coefficients can be calculated from the following equation:

$$P = V \times I \quad (6)$$

The internal cylinder's convection and radiation transport heat are:

$$Qcr = P - Qcond \quad (7)$$

Wherein, Qcond is the heat by conduction, as determined by the equation given below:

$$Q_{conduction} = \frac{\Delta Toi}{ln\frac{roi}{rii}/2\pi kl} \quad (8)$$

representation of the convection-radiating heat flux is:

$$q_{cr} = \frac{Qcr}{Ai} \quad (9)$$

where: Ai is the surface area of the internal cylinder, which ($Ai = \pi Dil$)

the radiation heat flux from the following equation:

$$Q_{Radiation} = \frac{\sigma * Ai((\overline{T_{hs}}+273)^4 - (\overline{T_{os}}+273)^4)}{\frac{1}{\varepsilon 1}+\frac{1-\varepsilon 2}{\varepsilon 2}(\frac{Di}{Do})} \quad (10)$$

where $\overline{T_{hs}}, \overline{T_{os}}$, is the inner and outer surface mean temperature.

σ = Stefan Boltzmann constant $= 5.67 \times 10^{-8}\, W/m^2.°C$,

$\varepsilon 1, \varepsilon 2$: emissivity of inner and outer surfaces, respectively, $\varepsilon = 0.65$

$$Q_{convection} = P - Q_{conduction} - Q_{Radiation} \quad (11)$$

$$q``_{conv} = \frac{Q_{convection}}{Ai} \quad (12)$$

$$\bar{h} = \frac{Q_{convection}}{Ai * \Delta T_{ave}} = \frac{q``_{conv}}{\Delta T_{ave}} \quad (13)$$

Hence, the average values of the Nusselt number may be computed from:

$$\overline{Nu} = \frac{\bar{h}\, L}{k} \quad (14)$$

5. UNCERTAINTY ANALYSIS

The uncertainty within the assessed parameters is calculated using the technique proposed by reference [20], [21], and [22]. The computation for the uncertainty in the assessed parameters of (heat transferring ratio q``, average heat transferring factor, average Nusselt value, Rayleigh number and vibrational Rayleigh number) used the uncertainty in the independent information, including geometric dimension, surface temperatures of the heated internal cylinder, fluid temperatures tapped in annular, voltages, currents, and vibrational frequency and amplitude of the vibrational heated internal cylinder .The problem solution for this random fluctuation in the measured values lies in repeating the measurements many times, such uncertainty depends upon the technique proposed by [20].

The maximal measuring uncertainties are as follows: heat flux (±4.27262%) , the transferring factor of average heat (±7.546986%), the Nusselt value (±12.6584%) , Rayleigh number (±3.509%) , finally, the vibration Rayleigh number (±12.68%).

6. RESULTS AND DISCUSSION

The experimental results for natural convection for concentric annuli concentric annuli$\in = 0, Pr = 0.7, Ra = 5 \times 10^4$, and radius ratio $Ro/Ri = 2.6$ were used as the test run and validation part of this work. Figure 5 shows the measured thermic conductivity of local equivalent as being compared to the experimentation data inferred from reference [1]. The two sets of the results for both the external and internal cylinders excellently agree, the maximal inconsistency takes place at the upper part of the external cylinder.

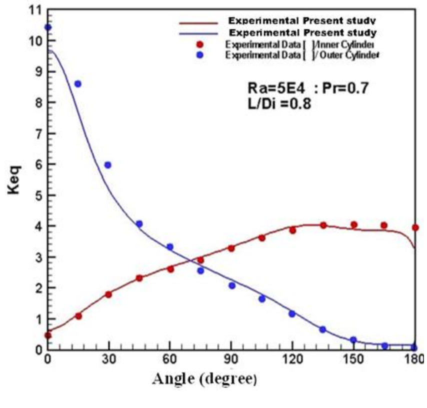

Fig. 5. Comparison of the concentric local heat transfer coefficients at ∈= 0 from [1]

As stated in the previous section of the measurement procedure, there are two groups of results that are given and discussed in this section. The first group of them is the natural convection heat transfer in horizontal annuli in concentric and vertically eccentric long cylinders with a stationary and vibrated heated inner cylinder. Air was used in an annulus as working fluid and was spontaneously heated via its inner cylinder while maintaining a steady heat flux CHF $25 \leq q`` \leq 1500$), $5 \times 10^4 \leq Ra \leq 6.48 \times 10^6$, and $Pr = 0.703$. When the center of the internal cylinder is above that of the external cylinder, so the eccentricity amid the two cylinders can be considered as positive, and can be considered as negative if the center below it.

Figures 6 illustrated the influence of the Rayleigh number upon the average Nusselt number for five various eccentricities at f=0Hz. In the present experiments, the results of a heated inner cylinder that was stationary and placed at five different eccentric positions were used as a reference to measure the enhancement resulted at vibrational conditions.

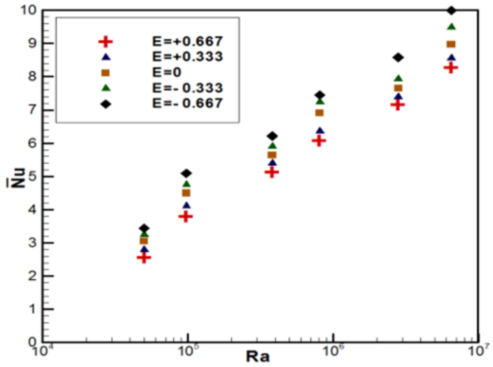

Fig. 6. Effect of Rayleigh number on the average Nusselt for different eccentricities for five different eccentricities at f=0 Hz.

Table 1 displays the ratio of the average Nusselt number's increase to the position of the inner cylinder within the outer cylinder, using the centric position as a reference for positive and negative eccentricity of the heated inner cylinder. It can be concluded that when the heated inner cylinder is positioned in negative eccentric locations, the average Nusselt number becomes significantly larger. The average Nusselt number decreases as the eccentric position of the heated inner cylinder moves vertically as of the negative to the positive through the centre of the outer cylinder. Such behaviour may be explained by the fact that for the negative position of eccentricity the heat transferring as of the cylinder and the influence of buoyant force are all directed toward similar direction, this causes an increasing within loading currents and enhances the heat transferring procedure. The Average Nusselt number in negative eccentric position is higher than that of the centric and in turn higher than that of positive eccentric position. This may be explained as follows: By transferring heat from the heated internal cylinder to the air wall through conduction and then convection inside the negative eccentricity, the air wall may be heated once more, that also impacts the density of the air wall and consequently influences the buoyant force.

6.1. The Effect of Vibration Intensity on $\overline{\mathbf{Nuv}}$

The vibrational intensity is defined as the product of the vibration amplitude by a frequency (b.f) with a unit of (m/s). Figure 7 shows the effect of the vibrational intensity upon the average vibration Nusselt number for all levels that applied heat fluxes at five eccentric positions of the inner cylinder. The average vibration Nusselt number is increasing with the same trend of behavior, but its value affected by the frequency. As the vibrational frequency increased, the vibration's density (intensity) also increased. This resulted in a specified increase in the average vibrational Nusselt number, this is in agreement with [23]. As a result, the maximal amount for the aveage Nusselt number can be attained once the vibrational intensity be equal to (0.092) m/s at the vibration amplitude (4.6 mm peak to peak) at each and every applied frequency and for the five different positions of the inner cylinder.

In general, for all power levels in this experiment, there is a proportional increase in $\overline{Nuv}$ when the vibrational intensity increases, but the influence of vibration is limited at low power levels and increases at high power levels.

6.2. Effect of Inner Cylinder Position (Effect of Eccentricity) on $\overline{Nuv}$

Figure 8 illustrates the relationship between the vibrational average Nusselt number and the logarithmic vibrational Rayleigh number for various eccentric vibrational inner cylinders for Rayleigh number. One can observe that the vibration Nusselt number rises as the vibration Rayleigh number rises for each Rayleigh number, the vibration average Nusselt number rises as well (power levels increased). Also, it was found that the vibrational average Nuselt number is much higher when the vibrating heating inner cylinder is at vertical negative position. For each Rayleigh number employed in this work, its value is lower in the concentric location than in the negative vertical location, which is lower even than the positive vertical location of the inner cylinder. Regarding to the vertical eccentric cylinders, the upper portion of the heated internal cylinder as well as the upper portion of the cold external cylinder would be facing upwards and downwards, respectively. The lower portion of the heated internal wall as well as the lower portion of the cold external wall would be facing upwards and downwards respectively, in these cylinders. In this structure, more heat will be transferred from the top half of the internal wall than the bottom part. As a result, the efficiency of heat convection inside the upper space amid the upper half of the internal wall and the outside external wall will regulate the overall heat transferring from the internal wall. Regarding to negative eccentricity, the broad upper space aids in enhancing heat convection because of the vigorous movement for all convection currents. Under the identical conditions but with positive eccentricity, the minimal convection contribution is caused by the tight upper gap and insufficient temperature gradient underneath the internal wall.

6.3. Effect of vibrational Frequency on $\overline{\mathbf{Nu}}$v

Figure 9 demonstrates the vibrational average Nusselt number with the logarithmic Rayleigh number, for five different heated inner cylinder positions and for all the applied vibrational frequencies (0,2,5,10,15,and 20)Hz that were used in this experiment for both situations (i.e., when there is no vibration and when there is vibration). one could observe about the vibration average Nusselt number to be enhanced by mechanical vibration, and this enhancement is best seen in the negative direction, then in the centric position and less enhancement in the positive vertical positions. Also it can be seen that at the lower range of Rayleigh number, the values of the vibrational average Nusselt number are close to each other at the lower range of the vibrational frequency and begin to be far from each other at higher vibrational frequencies. For high ranges of Rayleigh number, the values of the vibrational average Nusselt number are far from each other and ascending along with ascending of the vibrational frequencies.

6.4. The gain in Nusselt number $\overline{\mathbf{Nu}}$v

The gain in Nusselt number due the mechanical vibration was estimated at the lower and higher value of the Rayleigh number $Ra = 5 \times 10^4, and\ 6.48 \times 10^6$, at all vibrational frequencies used in this work $= (0,2,5,10,15,20)Hz$, and for eccentric sites of the inner cylinder($\in= \frac{e}{L} = -0.667, 0\ , and\ +0.667$), the results are shown in Table 2, and Table 3.

$$Gain(\overline{\text{Nu}}) = \frac{(\overline{\text{Nu}})\text{v}-(\overline{\text{Nu}})}{(\overline{\text{Nu}})} \times 100\% \quad (15)$$

Finally, an attempt is made to correlate the results obtained from the present experiments. The current work heavily depend on dimensional numeric analyzing so as to establish equations that connect non - dimensional values, such as the average Nusselt number $\overline{\text{Nu}}$, Rayleigh number Ra, and the vibrational Rayleigh number Ra_v to each other. The empiric equations are divided into two sets, the first of which indicates the average Nusselt number in the absence of vibration. The general equation was obtained:

$$\overline{\mathbf{Nu}} = C(Ra(1-\in))^a \quad (16)$$

With a maximum error of $\pm 7.91\%$ at $\in= +0.667$, and at $\in= -0.667$, the maximum error was $\pm 7.0\%$

The second set of empiric equations compute the vibrating average Nusselt number $\overline{\boldsymbol{Nu}}v$ where there is no vibration, the general equation at each eccentric position is:

$$\overline{\boldsymbol{Nu}}v = C(Ra(1-\in))^a Rav^b \quad (17)$$

Where C, a, and b represent coefficients that their values could be achieved by utilizing analysis software IBM SPSS Statistics 13.0. With the maximum error was $\pm 7.83\%$ at $\in= -0.667$ at f=5 Hz.

7. CONCLUSION

In the current experiment, under the impact of vertical forced vibration of the heated inner cylinder, the natural convection developed in an annular channel amid two horizontal concentric and several vertical eccentric lengthy cylinders. The outside wall was cooled and kept at a steady temperature, whereas the internal wall of the annulus is heated with a roughly homogeneous heat flux. Along with the effect of Rayleigh number, and the effect of inner cylinder location (concentric and vertical eccentric), the natural heat convection process in the annulus depends and effects on the vertical mechanical vibration of the heated inner cylinder. The heated inner cylinder was subjected to various vibration frequencies at each eccentric location of the inner cylinder at a fixed aspect ratioRo⁄(Ri=2.6), and fixed Prandtle number Pr=0.703. The heat transfer rate for the natural convection between horizontal cylinders in a negative eccentric vertical configuration was found to be higher than that in the concentric case, which used the centric position as a reference for the enhancement result and is higher still than that in the positive eccentric vertical configuration when the internal cylinder is moved upward from a concentric position along a vertical line. Approximately, the increasing ratio in average Nusselt number relative to the centric position is 1.078674 and increases at ∈=-0.667, while it is 0.83417 and increases at ∈=+0.667 for Rayleigh number Ra= 〖5×10〗^4, and for Rayleigh number Ra= 〖6.48×10〗^6, respectively. The increasing ratio in average Nusselt number is 1.132971 at ∈=-0.667, while it is 0.934155 and increases at ∈=+0.667. It was found that the average Rayleigh number increased linearly with the Nusselt number for a concentric and all eccentric configurations in the presence and absence of vibration. Eventually, the findings show the increments in vibration frequencies lead to an increase in vibration average Nusselt number for all eccentric configurations and in varying proportions. The maximum augmentation was obtained in negative vertical eccentricity and at higher frequency in the scope of the current circumstances with the potential exception of the maximum eccentricity.

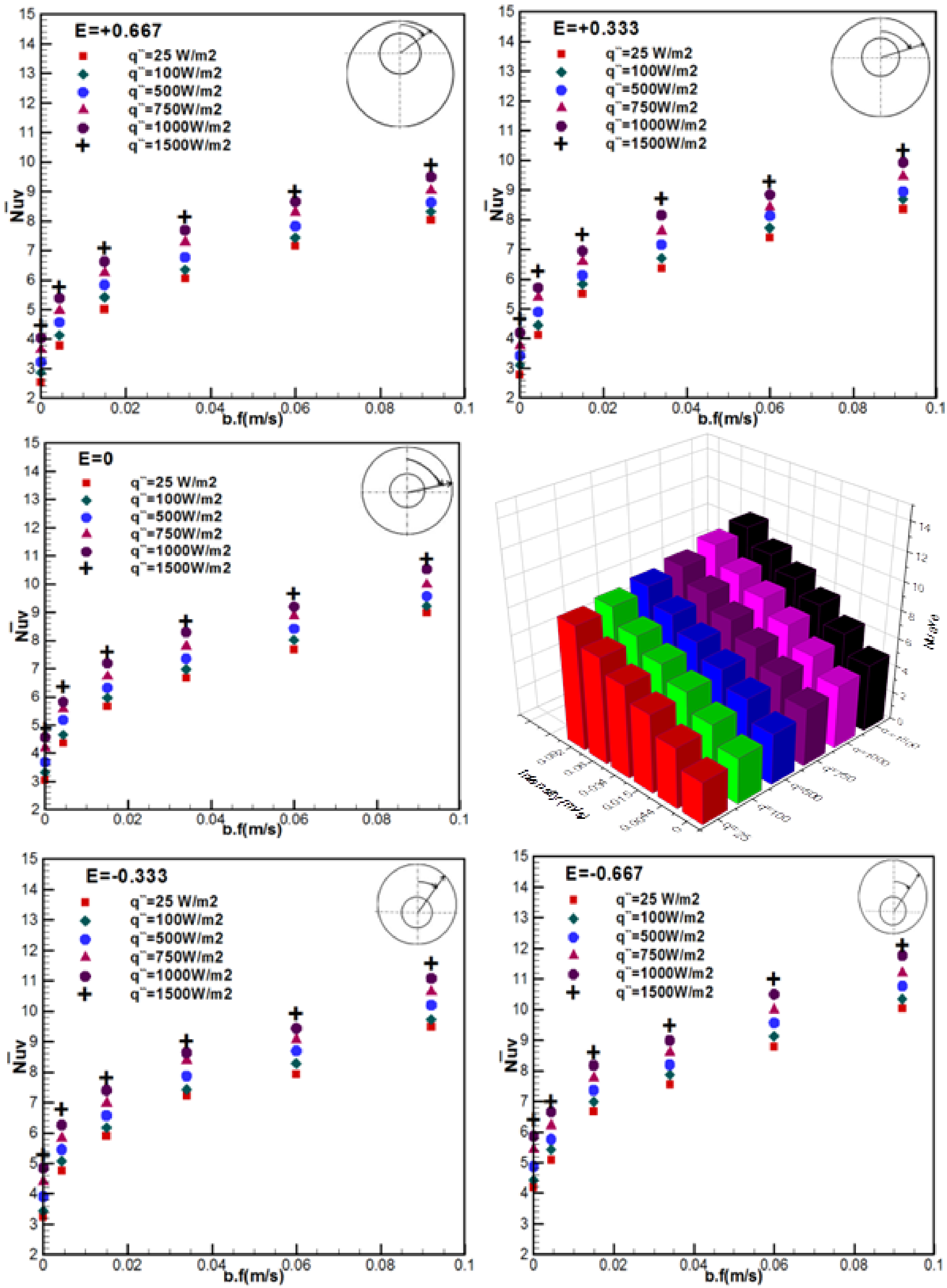

Fig. 7. The effect of vibration intensity on the average Nusselt number for all applied heat flux at each eccentricity.

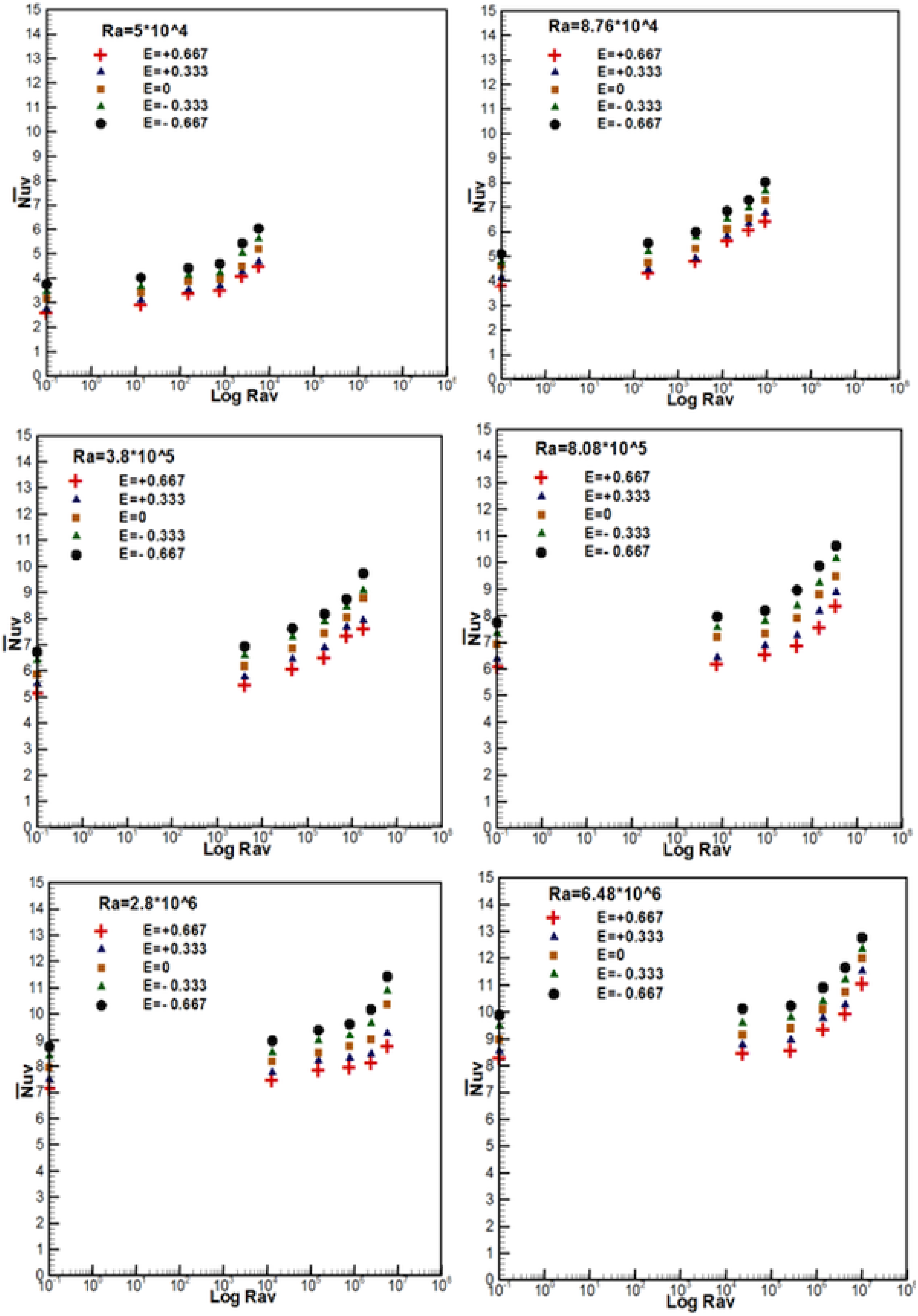

Fig.8. Vibrational average Nusselt varied with the logarithmic vibrational Rayleigh number at each Rayleigh number for different eccentricities

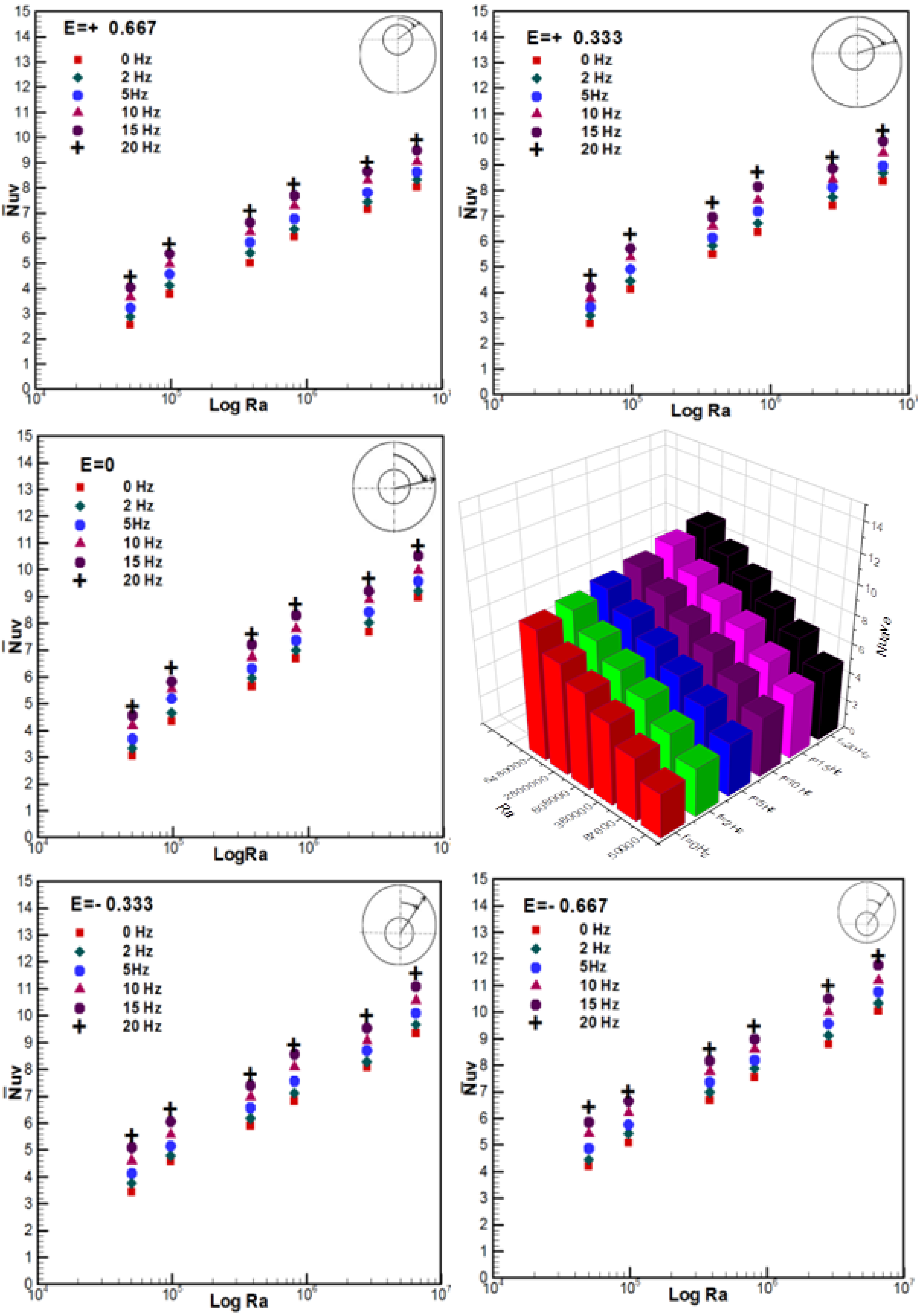

Fig.9. Average Nusselt versus with the logarithmic of Rayleigh number at all applied frequencies for each eccentric

Table 1. The ratio of increasing in average Nusselt number according to the eccentricities

Heat flux(W/m^2)	25	100	500	750	1000	1500
∈= +0.667	0.83417	0.840864	0.876719	0.908254	0.921459	0.934155
∈= +0.333	0.914396	0.915215	0.920518	0.95765	0.953407	0.966297
∈= 0.0	1	1	1	1	1	1
∈= -0.333	1.035575	1.04649	1.047433	1.056691	1.057619	1.064121
∈= -0.667	1.078674	1.102337	1.114052	1.121537	1.130909	1.132971

Table 2.: $\boldsymbol{Gain}(\overline{\mathbf{Nu}}) = \frac{(\overline{\mathbf{Nu}})\mathbf{v}-(\overline{\mathbf{Nu}})}{(\overline{\mathbf{Nu}})} \times \mathbf{100\%}$ at $\boldsymbol{Ra} = \mathbf{5 \times 10^4}$

f(Hz)	∈=-0.667	∈=0	∈=+0.667
0	0	0	0
2	3.789970928	1.275566794	0.066113316
5	4.409297636	2.708536405	0.634825513
10	12.94426972	11.08624164	7.940882556
15	20.69568236	16.19494147	13.80764335
20	34.29165989	29.96649678	22.50601475

Table 3. $\boldsymbol{Gain}(\overline{\mathbf{Nu}}) = \frac{(\overline{\mathbf{Nu}})\mathbf{v}-(\overline{\mathbf{Nu}})}{(\overline{\mathbf{Nu}})} \times \mathbf{100\%}$ at $\mathbf{Ra} = \mathbf{6.48 \times 10^6}$

f(Hz)	∈=-0.667	∈=0	∈=+0.667
0	0	0	0
2	13.21686988	9.740053321	7.624531889
5	30.95760255	23.34798395	19.36311496
10	36.67899771	25.86179462	24.29291765
15	59.103177	48.99750849	48.43345106
20	86.66933125	76.65264216	62.40585348

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