Vibration Effect on Natural Convection Heat Transfer in an Inclosed Cubic Cavity

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Abstract. An experimental study was performed to clarify the effect of vertical mechanical vibration on natural convection at normal gravity in the air filled cubic enclosure (L=120mm) \((P_r = 0.71)\). In the enclosure, there were two vertical and opposing surfaces. The right wall was heated to a standardized heat flux surrounded by four other adiabatic surfaces, where the left wall cooled. Vibration stress was added to this heat transfer cell by vertically mounting it on the armature of electrodynamics vibratory (shaker). The experimental work was performed on a built rig which was mainly composed of a cubic enclosure cavity provided with a vibrator exciter as well as the necessary measurement instrumentation to fulfill the required investigations. At \(Ra = 7 \times 10^7\) the frequencies shedded to the enclosure (2,4&8) Hz and at \(Ra = 4 \times 10^8\) the frequencies shedded to the enclosure (3,6&9)Hz. Three type of tests for an experimental were carried out. The first one reached to steady state and then shedded the effect of vibration to the cubic enclosure (Interrupted Vibrations), while the second shedded the vibration (Continuous Vibrations) from the start at ascending frequencies and the third shedded the vibration (Continuous Vibrations) from the start at descending frequencies. From the results of the experimental investigations two main conclusions may be raised, in the case of Rayleigh number \((Ra = 4 \times 10^8)\), the gravitational thermal convection is dominant. And the motion of vibration does not enhance the transfer of heat exceptionally well. On the other side, in Rayleigh \((Ra = 7 \times 10^7)\), the thermal vibration convection is dominant, and the vibration greatly increased the rate of heat transfer. Also, the results show that an increase in the average Nusselt number with time as the vibrational Rayleigh number \(Ra_{vib}\) will increase as resulting of increasing the vibration frequencies. In addition, the higher the frequency of vibration reaches, the faster the steady states is achieved. And for that, two cases of Rayleigh number, the increasing frequencies are usually higher than those of downward frequencies. Finally, the results show reasonable agreement between the theoretical and the experimental results and the present results and available previous work.

Keywords. Cubic enclosure, Mechanical vibration, Vertical and horizontal directional vibration, Ascending and descending frequency, Interrupted and continuous vibrations.

1. Introduction
The study of natural convection in an enclosure has been studied for decades due to its comprehensive architecture. Tech applications, such as solar energy systems, electronics cooling devices, crystal growth processes and so on. However, most of the studies have concentrated on the static case, in which the
enclosure is placed. Fixed to an inertial frame and subjected to steady gravity that’s all [1]. The natural convection in the enclosure has produced a great deal. Dealing with publicity in the past, but studies of thermal convection in the enclosure caused at the same time by gravity and vibration, which is important in material processing or in heat transfer under a vibrational environment are very rare. [2]. To validate the mechanism of natural convection on the vibrating enclosure induced by the operation of the system becomes an urgent need for the design of precise and effective device. Vibrations are known to be among the most effective ways of affecting the behavior of fluid system, in the sense of increasing or reducing the convective heat transfer. In the past, a variety of similar studies have studied the effects of vibration of the natural convection. The effect of mechanical vibrations, as well as sound waves, on heat transfer, a number of research studies have researched the transition from body to an infinite atmosphere. Attention has been paid to the concept of using mechanical vibration as a way of improving heat transfer. Lord Rayleigh (1877) first studied streaming flow phenomena from the very starting, in conjunction with them of waves of the sound. In 1960’s, Russian scholars like Gershuni et al., (1963) Zen’kovskaya (1966), and Simonenko (1972) et al., in vibrational convection, is pioneered [3]. [4] thermal convection was in vertical rectangular enclosure packed with water. The vibration level was different and acceleration at heat transfer rate is to be understood. The heat transfer rate has been shown to increase by the vibration, particularly near the normal resonant natural frequency of the fluid column within the box. The overall increase in heat transfer was around50 percent achieved during the experiment. In [5], the vibration effect on the cooling process has been experimentally tested layer of liquid between the concentric cylinders. The finding revealed that the vibration frequency increasing lead to decrease the cooling time of the fluid. [1and 2] investigated the natural convection in an enclosure caused by variable acceleration and vibration, respectively. The Nusselt numbers varied with the relating parameters were obtained and succinct correlation equations were used to estimate corresponding parameters and the resonant frequency has been extracted. [6] completed the study of thermovibrational convection done [1 and 2] by providing vertical, cylindrical cavity thermo-vibration model and the heat transfer rate dependency through the device at Rayleigh numbers, Ra, of 0, 10^4, and 10^5 and vibrational Grashof numbers, Gr, of 10^5 and 10^6 are researched. Results revealed that a slightly enhanced heat transfer rates over the unmodulated case of vibrational convocations was (7-15%) at Gr = 10^5 and by (50-65%) at Gr = 10^6. The frequencies observed appeared very much in the line with the projections of the frequency equation of resonances.

\[ \omega_r = \sqrt{2GrPr} \]  

[1] estimated a valuable method to estimate (\(\omega_r\) the resonant frequency). Resonance frequency in low Rayleigh numbers, however, this equation was generalized to incorporate the influence of static gravitation field on the resonance frequency and the equation emerges.

\[ \omega_r = \sqrt{2GrPr^2 + 2GrPr^2 + 2Pr^2RaPr} \]  

(2)

It could be more suited for certain cases with a slightly different or equivalent Rayleigh number and vibrational Grashof number. [7] numerically studied the buoyant convection in a Boussinesq fluid side-heated square cavity was used as a benchmark [8]. The two vertical one sidewalls were held at various constant temperatures Th and Tc, respectively, and the horizontal walls were isolated. The computed results indicated that the resonance should occur when the oscillation frequency matches the fundamental mode of internal gravity oscillations. Nusselt was maximized at the resonance frequency and the convection activity in the interior was intensified. The vertical orientation oscillation induces quantitative shifts, but while the oscillation is in the horizontal direction it is more pronounced and qualitative. [9] experimentally studied the resonance of natural convection in a mechanically oscillating bottom wall with side-heated enclosure. Impact of forcing frequency (356 < \(\omega\) < 8556) and amplitude (0.03 < b < 0.06). The experimental findings show the amplitude, which is representative of the resonance of fluctuating air
temperature in the cluster top. The frequency improved with the rise in the Rayleigh number of the system and the rise in the force amplitude was minimally influenced.

[10] studied numerically the effects of vibrational heat surface on the natural convection in a vertical channel flow. The findings showed that, for the same Rayleigh number, the normal convection of a vibration-heated plate with a certain combination of frequency and amplitude was possibly smaller than that of a stationary state. Increased heat transmission of the heated layer, which occurs if the frequency for the same Rayleigh number, is more important than the critical frequency for heat transmission and vice versa. [11] proposed method of achieving forced oscillation by an arbitrary small-time force on steady thermostat field, and structure for deciding the intrinsic and universal modes of field resonance, regardless of the forced vibration methods, resonance markers and other. In the application of the resonant mode, the protocol also presented metrics to assess the effectiveness of a special forced vibration. [12] experimentally tested the impacts on Rayleigh exceptionally wide vibration spectrum or on thermo-diffusion in a microgravity setting that is subject to a constant temperature difference between two cubic cavity walls with different parameters with vibrational energy, such as frequency and amplitude. And they studied the effects on the change in forced vibration from low to high vibration Rayleigh number on the part differentiation in the blend. Digital optical digital interferometry has limited the experimental data from the ISS (The International Space Station) the world gravity. The test was conducted in a box containing a water solution with a temperature profile 10 K temperature gradient. A different combination of water and isopropanol with a negative Soret coefficient and a temperature differential similar to the test cases has been selected. The findings showed a linear relationship between the vibration Rayleigh number and the maximal separation was not found for the situation of minimum Rayleigh number. This is surprising because the growth in Rayleigh number is always connected to decrease in component separation. [13] provided the influence of heat transfer vibration surface on boiling nuclear pool of saturated water under atmospherical pressure. A mechanical vibrator at frequencies between (0 and 25 Hz) and amplitude (from 0 to 5mm) used to control and vibrate 19mm diameter circulatory copper test surface. In the field under analysis, a growth in heat transfer coefficient up to 123percent with the highest amplitude and frequency of vibration has been observed. Visualization of boiling phenomenon demonstrated significant improvements in the bubble frequency with decreased in the diameter of the starting bubble when the surface vibration was caused.

Consequently, the goal of this research is to study experimentally the comprehensive thermal convection heat transfer device, which is affected by gravity and vibration jointly in a cubic enclosed cavity at three cases and in vertical direction of vibration. Case I reaches a steady state and then begins the effect of vibration at each frequency. Case II and case III begin the effect of vibration from the transient at ascending and descending frequencies respectively caused by abrupt change of the wall temperature and vibration conditions. The phenomena for a Rayleigh number of $7\times10^7$ influenced by the vertical vibration are more remarkable than those for a Rayleigh number of $4\times10^6$; the case for Rayleigh numbers of $7\times10^7$ and $4\times10^6$ are then preliminarily studied and the vibrational Rayleigh number at each frequency and the dimensionless frequency of vibration at the table 1. In this case, the heat transfer process and the vibration perpendicular to the temperature gradient were the most significant factors influencing the situation. Particularly, the effects of the vertical vibration frequency on the heat transfer sequence of events are taken into consideration. This leads in case of disturbed vibration, to shift the vibration frequency from low to high frequency at case of interrupted vibration (reach to steady state then begin the effect of vibration), and at the case of continuous vibration (at ascending frequencies and descending frequencies from start of operating the system). The time history of the isothermal lines and Nusselt number are presented.
### Table 1. Characteristic dimensionless.

| Rayleigh Number | Frequency (fr) Hz | Dimensionless Frequency (ω) | Vibrational Rayleigh Number |
|-----------------|-------------------|-----------------------------|-----------------------------|
| 7×10⁷           | 2                 | 1.257                       | 2×10⁴                       |
| 7×10⁷           | 4                 | 2.513                       | 3×10⁴                       |
| 7×10⁷           | 8                 | 5.02655                     | 2×10⁶                       |
| 4×10⁸           | 3                 | 0.57                        | 7×10⁵                       |
| 4×10⁸           | 6                 | 1.13                        | 4×10⁶                       |
| 4×10⁸           | 9                 | 1.7                         | 1.6×10⁸                     |

2. **Data analysis (Analysis of heat transfer)**

The electrical power input to the hot wall was measured by using the current and the voltage across the heater. The heat is transferred in the following ways:

1- Heat loss by conduction to the surroundings.
2- Heat exchange by radiation in the enclosure.
3- Heat transfer by natural convection in the enclosure.

Total input power \( (I.P) = Q_{\text{conduction}} + Q_{\text{radiation}} + Q_{\text{convection}} \) \( (3) \)

To know the amount of thermal energy transmitted by natural convection, the following calculations have been performed:

2.1. **Calculation of thermal energy transferred by conduction**

Conduction heat loss is assumed to consist of the heat loss from the back side of the hot wall. The following equation was used to calculate this loss. From \[14\]

\[
Q_{\text{conduction}} = U.A_h(T_h - T_a)
\]

\[
R = \frac{1}{\frac{1}{h_o} + \frac{x_1}{K_1} + \frac{x_2}{K_2}}
\]

Where; \( h_o \) = Exterior convective heat transfer coefficient (ASHRAE 1981) =3.08 W/m².K, \( x_1 \) = the thickness of glass wool =50mm, \( K_1 \) = conductivity of glass wool = 0.038 W/m.K \( \) [14], \( X_2 \) = Aluminum thickness =10mm, and \( K_2 \) = Aluminum conductivity (6061T6) =167 W/m.K.

\[
Q_{\text{conduction}} = 0.0088(T_h - T_a) \text{ (Watt)}
\]

2.2. **Calculation of thermal energy exchanged by radiation**

The thermal energy transferred by radiation can be calculated from the following equation: from \[14\]

\[
Q_{\text{radiation}} = F_{12} \* \sigma \* (T_{hs}^4 - T_{cs}^4) \* A_c \text{ (Watt)}
\]

Where \( \sigma \) : Stephan-Boltzman constant = 5.669×10⁻⁸ (W/k⁴m²), From ref. [14], \( F \) = shape factor = \( \frac{2}{\varepsilon_s} - 1 \), \( \varepsilon_s \) = emissivity of heat exchange surface and its value for polished Aluminum is (0.04) [Holman, J.P 2008], \( T_h \) : Local hot surface temperature, \( T_c \) : Local cold surface temperature, and \( Ac \) : Cross sectional area (L*L).

The heat loss by conduction to the insulation and by radiation were calculated and estimated as (2% \( \leq \frac{q_{\text{conv}}}{q_t} \leq 5\%\))

2.3. **Calculation of the electrical power supplied to the heater**

The electrical power supplied to the heater was calculated from the following equation:

\[
I.P = I. V
\]

\( (8) \)
Where;  $I = \text{AC (current measured by clamp-meter)}$, and $V = \text{AC (voltage determined by digital voltmeter)}$.

2.4. Calculation of thermal energy transferred by convective from energy balance

The net heat transfer by natural convection between two walls was obtained from the equation: from [14]

$$Q_{\text{convection}} = I . V - (Q_{\text{conduction}} + Q_{\text{Radiation}}) \quad (9)$$

2.5. Calculation of heat transfer coefficient

The readings recorded by the thermocouple on the heat exchange surface were used to calculate the temperature gradient (heat flow) by natural convection due to a temperature differential between hot and cold walls forming a cubic cavity with air as a working fluid. The convective heat transfer coefficient was calculated as follows: from [Holman, J. P. 2008]

$$Q_{\text{convection}} = h . A_h (T_h - T_c) \quad (10)$$

$$h_{li} = \frac{q_{\text{convection}}}{A_h(T_{hi} - T_c)} \quad (i = 1 - 5) \quad (11)$$

$$h_{li} = \frac{q_{\text{convection}}}{(T_{hi} - T_c)} \quad (i = 1 - 5) \quad (12)$$

Where; $T_{hi}$ = local hot wall temperature, and $T_c$ = cold wall temperature.

2.6. Calculation of local nusselt number

The local Nusselt number was calculated from the following equation

$$N_u = \frac{h_{li}}{K_f} \quad (i = 1 - 5) \quad (13)$$

2.7. Calculation of average nusselt number

The average Nusselt number was calculated from the following equation

$$\overline{N_u} = \frac{1}{A_c} \int_0^c N_u(x) \, dz \quad (14)$$

$$\overline{N_u} = L \overline{h}/K_f \quad (15)$$

3. Application creative and evaluation process

Experiments have been conducted in an internal cubic enclosure of size 120mm high, 120mm wide and 120mm deep inner cubic enclosure. It consists of two aluminum and Plexiglass panels on the opposing vertical side-walls which form the cubic cavity’s four walls, aluminum plates and Plexiglass plates which form the four walls of a cubic cavity. Positioning of aluminum plates and suitable Plexiglass plates to obtain the desired transverse area and cavity aspect ratio. The Plexiglass plates reflect the isolated walls as (adiabatic surface), since the thermal conductivity (0.2) $W/m.K$ is measured for low thermal conductivity [14]. The right-hand side of the aluminum wall (270° reflecting the vertical enclosure, but heated from the right side [Shabana M. Diaa 1990], [15] was electrically heated by means of a Tungsten wire to provide a continuous heat flux the opposite side wall comprised of a water jacket made of copper pipe. This water jacket, which created a constant temperature of a cold. The vibration alternator (shaker) was situated in the center of the box where the unit comprises of the following parts as shown in figure (1):

1. Cubic enclosure (Test cell).
2. Closed circuit of water
3. Circuit of heater
4. Circuit of Thermocouple  
5. Device of Vibration  
6. Measurement framework  

Twenty K-type thermocouples with (15mm, 50mm and 90mm) in various sizes location within the box, as which can be seen in figure (2-a), (2-b) and (2-c).

The influence of vibration at normal gravity on heat transfer was tested by the analysis works by simulation local fluid temperature within the box at the plain observation points, as shown in figure (2-a). to pick a protocol to store data, descriptive research design was carried out. First, it was to determine which one to create a thermal layer and then place a vibration field on it, or to set these fields in the opposite direction. Having taken into consideration the planned potential uses for details to be generated, thermal field was first set up in a test device and then vibrational stresses were placed on the box. Two related heat flux values have been measured in three distinct situations. Case: Run on the experimental rig and wait until an equilibrium (steady state). The equilibrium temperature was established when the temperature difference of each temperature sensor did not alter ±0.1°C in excess of one thermocouple for a time of 60 to 90 minutes. The cube cavity was then activated at nominal amplitude at its primary responsibility. The frequency and amplification metrics measurements presented details for determination of vibration Rayleigh number. The pulse ended after two hours, the pulse stopped and the linear progression came back. Case II explores the influence of vibration in the cold and hot circuits for upward two set of frequencies from start of the beginning of operation. Case III explores the influence of vibration from the start of the operation of the cold and hot circuits for downward two range of frequencies.

Both air thermal fluid characteristics have been calculated at the average fluid temperature which called (film temperature $T_f$) as seen in the following equation:

$$ T_f = \frac{(Th + Tc)}{2} $$

(16)

Based on film temperature, dimensionless features are computed. The nominal heat flux with characteristic dimensionless quantity is shown in table (1).

**Figure 1.** Schematic diagram for the testing rig with the implemented vibration.
Figure (2-a). Schematic diagram of the testing cell

Figure (2-b). Temperature position of the planes into the cubic cavity

Figure (2-c). Orientation of temperature sensors on either plane of the cabin

**Figure 2.** Schematic diagram of the enclosure & orientation of temperature sensors
4. Discussion of results

4.1. Temperature fields

Experimental means temperature variation with time of air temperature activity between the hot and the cold plane in the cube enclosed for the case (voluntary, stable state reach, various frequencies implement and then reach to steady without vibration effect). The heat flux was set to two values \( q'' = 85 \) W/m\(^2\) for frequencies (2, 4 & 8) Hz and \( q'' = 946.017 \) W/m\(^2\) for frequencies (3, 6 & 9) Hz as shown in figure (3). The time when the box is under the influence of impulse vibration two (2 hours) at each frequency. Clear the redistribution of the temperatures was determined, (the cold wall on both planes was estimated from the cold wall \( Z = 0 \) mm to the hot wall \( Z = 120 \) mm) and the shaking of the cavity has a major effect. In addition, it should be noticed that the temperature differential distribution of the plains is closer to each other relative to the stable condition before shaking.

4.1.1. Case I. Figures (3-a) shows the time variation-temperature differences between the hot and cold walls at two values \( q'' = 85 \) W/m\(^2\) for frequencies (2, 4 & 8) Hz, and figure (3-b) reveals the time variation-temperature differences between the hot and cold walls at \( q'' = 946.017 \) W/m\(^2\) for frequencies (3, 6 & 9) Hz. It can be noted from this figure that the temperature difference decreases for the forced vibration increases. This is discussed later when the average Nusselt number results are presented.

4.1.2. Case II & III. Figures (4-a) and (4-b) illustrate the comparison of temperature differences \( \Delta T \) between the isothermal walls at constant heat flux \( q = 85 \) W/m\(^2\) and \( q = 946.017 \) W/m\(^2\), when comparing the temperature difference between the two cases (ascending and descending frequencies). It can be seen from these figures that at the beginning with a high forced frequency and gradually decreasing the frequency (descending case), the temperature difference at this case is higher than the opposite case, when starting at low frequency and gradually increasing (ascending case) and vice-versa. The flow intensity is also substantially increased at increased Rayleigh vibration number as a result of increasing vibration frequency at two amounts of Rayleigh number as a result of increasing vibration frequency, and the thermo-vibrational convection is dominant.

![Figure 3](image-url)  
**Figure 3.** Time-dependent behavior of temperature difference measured between the hot and cold walls with various frequencies (a) \( q'' = 85 \) W/m\(^2\) (b) \( q'' = 946.017 \) W/m\(^2\)
Figure 4. Time-dependent behavior of temperature difference measured between the hot and cold walls of the cavity with various frequencies ascending and descending (a) $q^*(\text{heat flux})=85\,\text{W/m}^2$ (b) $q^*(\text{heat flux})=946.017\,\text{W/m}^2$.

4.2. Heat transfer
A comparative analysis of heat transfer for the latest work and that of some previous non-vibration data researchers. The consensus is considered to be successful and is shown in Figure (5), [16-19].

Figure 5. Log-log ($\overline{N\nu}, \text{Ra}$). Similarities between the heat transfer data of other vibration free researchers.
4.2.1. (Interrupted vibrations)

- **When** \( q^* = 85W/m^2 \). Figure (6-a) shows the variation of mean value of the time-varying average Nusselt number \( \overline{Nu} \) for induced different frequencies (2,4&8)Hz and the comparison between them. It can be clearly seen from this figure that \( \overline{Nu} \) increases with the forcing frequency and amplitude of cubic cavity and then varies periodically. It is worth noting that there was a minor impact on heat transfer of the 2Hz vibration on the requirements of movement, in which the boundary layers were in the transition flow pattern prior to pulse. At the case for increasing the shaker table frequency to 4 Hz, a major development in the time-average Nusselt number \( \overline{Nu} \) was obvious. A noticeable increase of the time-average Nusselt numbers \( \overline{Nu} \) effect when vibration was implemented to the enclosure at 8 Hz. This impact increases with the duration of vibration shedding. The flow flocculation charged particles up-and-down. The local density isotherms adjacent to vertical walls are lower at \( fr=2Hz \) than at \( fr=8Hz \). This indicates that the thermal boundary layers are thicker and thus the thermal transmission rate on the vertical surface is weaker at \( fr=2Hz \) and \( Ra= 7 \times 10^7 \). Again, the flow flocculation is not noticeable as the vibration frequency rises to 4Hz. At \( Ra=4 \times 10^8 \) the thicker limit layer is increased at increased of vibration frequency and the heat transfer rate on the vertical walls is lower than at \( fr = 8Hz \).

- **When** \( q^* = 946.017W/m^2 \). Figure (6-b) depicts the time-variation of average Nusselt number for induced different frequencies (3,6&9) Hz and the comparison between them. It is clear that the time-average Nusselt number increases as the heat flux increases as well as for induced vibration. Since \( Q = \overline{T} A\Delta T \) coefficient, \( \overline{T} \), the corresponding decrease in \( \Delta T \), \( q^* = Q/A \) shall be kept constant for this work. As predicated, the heat transfer coefficient is higher for higher heat flux ,but the boundary thickness increases as the temperature differential increases, which causes (\( \overline{h} \)) to decrease downstream. The heat transfer rate is higher as the frequency increases steadily as the thickness of the boundary layer gradually decrease. Figure (7a&b) reveals the effect of increasing interrupted vibrational frequency on the thermovibrational on heat transfer rates \( \overline{Nu} \).

It can be seen from these figures \( \overline{Nu} \) increases with the increasing of the frequencies for two amounts of heat flux \( q^* = 85W/m^2 \) and \( q^* = 946.017W/m^2 \) respectively.

![Figure 6](image-url)

Figure 6. The variations of time- average Nusselt number for at Interrupted frequencies(a) (2,4&8) for \( q^* = 85W/m^2 \). (b) (3,6&9) for \( q^* = 946.017W/m^2 \).
4.2.2 Continuous vibrations. Figures (8-a) & (8-b) display the variance of the time-average Nusselt number at \( q^* = 85 \text{W/m}^2 \) and \( q^* = 946.017 \text{W/m}^2 \), respectively. When comparing the effect of seismic excitation on upward frequencies and downward, it can be shown that the mean Nusselt number is higher for ascending than for the descending frequencies. In this method, the vibration variance of all frequencies level is a significantly changeable system for the temperature profile enclosure field. The temperature profile is undergoing a surprising transition as the vibration frequency increases. In the case of a lower frequency, the flow in the enclosure is primarily caused by the buoyancy force. In the area adjacent to the hotter right wall, the fluid shifts upwards and that parallel to the cooler left wall, goes downwards. A circulation of the flow is then created in the enclosed cavity. The orientation of stream center is not stable, but it is time-based. The airflow in the structure is guided by the inertia influences of the of the vibration enclosure movement as the vibration frequency rises to be 8Hz or 9Hz of ascending. Figures (9-a) and (9-b) exhibit the three-dimensional experimental average Nusselt numbers of the induced vibration of the enclosure for various frequencies (2, 4& 8) Hz and (8,4&2)Hz ascending and descending, respectively with time (t) and Rayleigh vibration numbers \( Ra_{vib} \) for \( q^* = 85 \text{W/m}^2 \). Figures (10-a) and (10-b) show the three-dimensional experimental variations of the average Nusselt numbers of induced vibration of the enclosure for various frequencies (3,6&9)Hz and (9,6&3)Hz ascending and descending, respectively with time (t) and Rayleigh vibration numbers \( Ra_{vib} \) for \( q^* = 946.017 \text{W/m}^2 \). Figures (9a and b) and (10 a and b) reveal as a result of increasing the vibrational Rayleigh number \( Ra_{vib} \) the average Nusselt numbers increases and oscillating along the time of operating the vibration as a result of increasing the frequency.

![Figure 7](image1.png)

**Figure 7.** The influence of Interrupted frequencies of experimental-mean Nusselt number for (a) \( q^* = 85 \text{W/m}^2 \) (b) \( q^* = 946.017 \text{W/m}^2 \)

![Figure 8](image2.png)

**Figure 8.** The variations of time-average Nusselt number for Continuous Vibration (a) \( q^* = 85 \text{W/m}^2 \) (b) 946.017W/m².
**Figure 9.** Correlations of variations of time-average Nusselt Number with Vibration Rayleigh Number for $q = 85W/m^2$ with various frequencies (a) ascending frequencies (2,4,8)Hz, and (b) descending frequencies (8,4,2)Hz.

**Figure 10.** Correlations of variations of time-average Nusselt Number with Vibration Rayleigh Number for $q = 946.017W/m^2$ with various frequencies (a) ascending frequencies (3,6,9)Hz, and (b) descending frequencies (9,6,3)Hz.
4.3. Improvement parameter $E(\overline{Nu})$

To determine the improvement of the heat transfer by external oscillation, the enhancement factor is defined as follows, [9]:

$$E(\overline{Nu}) = \frac{\overline{Nu}_p}{\overline{Nu}_s}$$  \hspace{1cm} (17)

Improvement parameter (Enhancement factor) $(E)$ the ratio of the mean Nusselt number with vibration to the Nusselt number at the stable state. For each Rayleigh number, the percentage of increase in time-mean Nusselt numbers relative to frequency is proposed in tables (2). One more thing we will see is the enhancement element $(E)$ at all frequencies at different Rayleigh number, the upward vibration gradually rising and the descending vibration is reducing.

| Frequency (Hz) | $\overline{Nu}_s$ | $\overline{Nu}_p$ | Percentage gains of the vibrating frequency |
|---------------|------------------|------------------|---------------------------------------------|
| 2             | 8.125            | 9.5              | 14.5%                                       |
| 4             | 8.125            | 11.3             | 39%                                         |
| 8             | 8.125            | 13               | 60%                                         |
| 3             | 30.2             | 32.5             | 7.62%                                       |
| 6             | 30.2             | 36.48            | 20%                                         |
| 9             | 30.2             | 42.003           | 39%                                         |

5. Conclusion

In this study of natural convection, thermovibrational ‘vibrational force’ emerges in a vibrating cubic enclosure experimentally investigated. A horizontally oriented, differentially heated, air filled cubic enclosure of cross-sectional aspect ratio of one (square) was confined to vibrate about its longitudinal axis. Flow visualizations were used to qualitatively understand the interaction of gravitational and vibrational buoyant flows and their effect on heat transfer. The study revealed temperature patterns in the vibrational Rayleigh number range $(2 \times 10^4$ to $2 \times 10^6$) for a Rayleigh number approximately $7 \times 10^7$ and $(7 \times 10^6$ to $1.5 \times 10^8$) for a Rayleigh number approximately $4 \times 10^8$. The conclusions are as follows:

1. The convection process at first frequency of vibration rate $(2$ Hz at $7 \times 10^7$ and $3$ Hz at $4 \times 10^8$) with the highly fluctuating amplitude of the air temperature can lead to chaos through a sequence of period of applied vibration because of the interaction between the influential gravity field and the uniform less effective vibration force field. This chaotic fluctuation in the interval of applied low frequency was seen in the isotherm patterns of air temperatures at all around of enclosure, as well as in the symmetric profiles of the local Nusselt number at the isothermal walls. Low vibration frequency intensities were seen to have little effect to enhance the heat transfer by approximately 16.9% at 2 Hz and 7.62% at 3 Hz.

2. When vibration frequency increased from $(2$ to $4)$ Hz at $7 \times 10^7$ and from $(3$ to $6)$ Hz at $4 \times 10^8$, this second frequencies represent a transition from gravitationally dominated to vibrationally dominated flow fields. An increase in vibration intensities was noted an increasing effect to enhance the heat transfer 39% and 20% at $Ra=7 \times 10^7$ and $4 \times 10^8$ respectively. The flow remains periodic or quasi-periodic at this frequency; this was seen from isotherm patterns.

3. In the third value of higher vibration frequency at 8Hz for Ra=$7 \times 10^7$ the vibrational force became more dominant but at 9Hz for Ra $4 \times 10^8$ the gravitational thermal convection returns to dominates and the vibrational force begins to recede. In this region, the heat transfer increased steadily with the vibrational frequency.
4. The average Nusselt number at (8 and 9) Hz was correlated with the vibrational Rayleigh number which represents a measure of vibrational acceleration. The increasing in the heat transfer rate were 60% at Ra=7×10^7 and 39% at Ra=4×10^8, respectively which obtained for vibration frequency at (8&9) Hz. At this frequencies the vibration intensities has higher effect than (4&6) Hz to enhance the heat transfer rate.

5. For the two Rayleigh studies The amounts of the mean Nusselt number at upward frequencies are generally greater than those downward frequencies.

6. Increasing the vibration frequency, faster steady state is achieved.

### Nomenclature

| Symbol | Description | Unit |
|--------|-------------|------|
| \( \alpha \) | Thermal diffusivity | m^2/s |
| \( A \) | cross sectional area (L*L) | m^2 |
| \( \beta \) | Thermal expansion coefficient | 1/K |
| \( f_r \) | Frequency | Hz |
| \( T_a \) | Ambient (atmospheric) temperature | K |
| \( I.P \) | power supplied (I.P)= I.V =Q\text{conduction} + Q\text{radiation} + Q\text{convection} | W |
| \( I = AC \) | Current through heating strip | Ampere |
| \( V = AC \) | voltage measured | Volt |
| \( Q\text{conduction} \) | Conduction heat transfer rate=U.Ah(T_h - T_c) | W |
| \( R \) | Thermal resistance \( R = \frac{1}{U} = \frac{1}{h_o} + \frac{x_1}{K_1} + \frac{x_2}{K_2} \) | m^2.K/W |
| \( U \) | overall heat transfer coefficient | W/m^3.K |
| \( h_o \) | Exterior convective heat transfer coefficient (ASHRAE1981) | 3.08 W/m^2.K |
| \( X_1 \) | Glass wool thickness | 50mm |
| \( K_1 \) | Glass wool conductivity | 0.038 W/m.K |
| \( X_2 \) | Aluminum thickness | 10mm |
| \( K_2 \) | Aluminum conductivity (6061-T6) | 167 W/m.K |
| \( Q\text{radiation} \) | Radiation heat transfer rate from specimen | W |
| \( T_h \) | Temperature of hot surface | K |
| \( T_c \) | Temperature of cold surface | K |
| \( F \) | shape factor = \( \frac{2}{\pi} \) | |
| \( Q\text{convection} \) | Convection heat transfer rate calculated from energy balance method= \( h.A_h(T_h - T_c) \) | W |
| \( h_{lo,i} \) | \( \frac{q_i}{(T_h-T_c)} \) | W/m^2.K |
| \( \overline{\Delta T} \) | Average heat transfer coefficient | W/m2.K |
| \( \Delta T \) | Temperature difference | K |
| \( \text{Pr} \) | Prandtl Number | - |
| \( \text{Gr} \) | Grashof number | - |
| \( Ra \) | Rayleigh Number | - |
| \( Ra_{ vib} \) | vibrational Rayleigh number | - |
| \( b \) | Maximum amplitude of vibration | m |
| \( \omega = 2\pi f_r \) | Angular frequency of vibration | rad/sec |
| \( g \) | Gravitational acceleration,9.8 | m / s^2 |
| \( \nu \) | Kinematics viscosity | m^2/s |
| \( \rho \) | Density | kg / m^3 |
\[ \frac{N_{\text{local}}}{N_{\text{avg}}} = \frac{\frac{L_o}{\kappa_{l}}}{\frac{1}{\sum_{i=1}^{5} N_{\text{local}}}} \]

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