

Experimental Study Of Natural Convection Heat Transfer In An Enclosed Vibration Cavity

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ABSTRACT

Experimental study has been implemented to elucidate an affect of mechanical vibration at normal gravity on natural convection in cubic enclosure side (L=120mm) filled with air ($P_r = 0.71$) at two amount of heat flux (. The enclosure was comprised of two vertical and opposed surfaces. The right wall was heated at uniform heat flux where the left wall cooled was maintained atT_c, surrounded by four other adiabatic surfaces. Vibration stresses were applied to this heat transfer cell by mounting it vertically on the armature of electrodynamic vibratior. The experiments were carried out at for Rayleigh number range ($7 * 10^7 - 4 * 10^8$)and aspect ratio equal (1), frequencies from 2to 8 Hertz at q''=85W/m^2 and from 3 to 9 Hertz at 946.017W/m^2). In the high Rayleih number case (Ra=4*10^8), the gravitional thermal convection dominates, and the vibration motion does not enhances the heat transfer remarkably. In contrast, in low Rayleigh (Ra=7*10^7), the vibration thermal convection is dominant, and the vibration enhaces the heat transfer rate significantly. In addition, the higher the vibration frequency is, the quicker the steady stateis reached and for two cases of Rayeigh number at ascending frequencies is in general higher than that descending frequencies.

Keywords: Cubic Enclosure, Mechanical Vibration.

1. INTRODUCTION

The study of natural convection in an enclosure has been investigated for decades due to its extensive applications in engineering, like solar energy systems, electronics cooling equipment, crystal growth processes, etc. However, most of the studies have concentrated on the static case, in which the enclosure is fixed on an inertial frame and subjected to a constant gravity only. Natural convection in an enclosure has received a great deal of attention in the past, but studies on the thermal convection in an enclosure induced simultaneously by gravity and vibration[1993]. The effect of mechanical vibrations, as well as sound waves, on heat transfer from bodies in an infinite atmosphere has been studied by many investigators[1970]. The idea of using mechanical vibration as a mean for enhancing the heat transfer has received attention from the early beginning; Lord Rayleigh (1877)first analyzed the streaming flow phenomena, in connection with sound waves. In 1960's, Russian scientists including Gershuni at al., (1963) Zen'kovskaya (1966), and Simonenko (1972) et el., pioneered the study of vibrational convection, Yuan[2003]. Forbes et al [1970] studied experimentally the thermal convection in a vertical rectangular enclosure filled with water. They varied the vibration frequency and acceleration to find on the heat transfer rate. It was shown that the heat transfer rate was increased by the vibration, especially near the resonant natural frequency of the liquid column contained within the enclosure. Ivanova [1988] studied the vibration effect on the cooling process of the fluid layer between the concentric cylinders. When the wall temperature decreased abruptly, the results showed that increasing the vibration frequency decreased the cooling time of the fluid. Fu and Shieh[1992]studied numerically the effect of vibration frequency on the heat transfer mechanism in a square enclosure. The dimensionless frequency varied from 0 to 10^4 and considered three different values of the Rayleigh number of 0, $10^4$and 10^6. According to the results, the thermal convection can be divided into five characteristic flow regimes, in the high Rayleigh number (Ra=10^6) case, the gravitational thermal convection dominates, and the vibration motion does not enhance the heat transfer rate remarkably. In contrast, in the low Rayleigh number (=10^4) case, except in the quasi-static convection region, the vibration thermal convection is dominant, and the vibration enhances the heat transfer rate significantly. Fu and Shieh[1993]studied the effect of vibration frequency on the transient thermal convection, four vibration frequencies (100,900,1100,5000)with fixed Rayleigh number (Ra= 10^4) and vibrational Grashof number(G=10^6). The results showed that the transient process, from the stationary state to the steady flow state, was shortened by increasing the vibration frequency. Sung Ki Kim et al[2002] studied experimentally a resonance of natural convection in a side-heated enclosure with a mechanically oscillating bottom wall. Impact of forcing frequency ($356< \omega < 8556$) and amplitude ($0.03< b < 0.06$). The experimental results showed that the amplitude of fluctuating air temperature inside the enclosure peaks at a particular frequency of the bottom wall oscillation, which is indicative of resonance. The resonant frequency increased with the increase of the system Rayleigh number and it is little affected by the increased of forcing amplitude.

The main aim of the present experimentally investigation is to provide an additional description of the basic mechanism of the combined effect of gravity and vibration in fluid-filled and heat transfer in a simple differentially heated, air filled cubic enclosure that vibrates vertically, which is oriented in a direction parallel to the gravity vector. Thus, the excitation inside the enclosure is purely mechanical by the external oscillation (vibration exciter; shaker) of the enclosure. The frequency and amplitude measurements provided data for determining the vibration Rayleigh number.

2. Experimental Setup and Test Procedure

The experiments were conducted in a cubic enclosure of internal size 120mm in height, 120mm in width and120mm in depth. It consists of two opposite vertical side-walls aluminum plates and Plexiglass plates that form the four walls of a cubic cavity. The aluminum plates and appropriate Plexiglass plates have been positioned to achieve the desired cross-sectional area and aspect ratio of the cavity. The Plexiglass plates represent the insulated walls as (adiabatic surface) which have a low thermal conductivity which were measured (0.2)$\mathrm{W/m.K}$ $[Holman, J. P2008]$. The right side of aluminum wall (270 °which represents the vertical enclosure, but heated from the right side[Shabana M. Diaa 1990] was electrically heated by means of a Tungsten wire to deliver a constant heat flux the opposed side wall consisted of a water jacket made of copper channel. Thus water from a cold reservoir circulated this water jacket providing a constant cold temperature at the wall surface. Twenty K-type thermocouples with (15mm,50mm and 90mm) in different location inside enclosure as shown in figure (2-a) and (2-b). The vibration exciter (shaker) was fixed with the center of the enclosure the rig consists of the following parts as shown in figure (1):

1. Test cell(cubic enclosure), figure (3)
2. Closed water circuit
3. Heater circuit
4. Thermocouple circuit
5. Vibration system

6. Measuring system

Experimental procedure investigated the effect of vibration at normal gravity on heat transfer by measuring the local fluid temperature inside the enclosure at the points of measurement plains, as shown in figure (2-b). A qualitative analysis was performed in order to select a procedure for accumulating data. It was first necessary to decide whether to establish a thermal field and then impose a vibration field upon it or to establish these fields in the reverse order. In the light of the expected practical applications for the information to be produced, it was decided to first establish a thermal field in the test system and then to impose different vibrational stresses on the cell. Three separate cases were examined with two different values of heat flux.

Case I: Running on the experimental rig and waiting till reach steady state case. Steady state temperature was defined when the variation of the temperature of each thermocouple would not exceed $\pm 0.1°C$ over a period of 60 to 90 minutes. The cubic enclosure was then vibrated at its response frequency at incremental amplitude. The frequency and amplitude measurements provided data for determining the vibration Rayleigh number. After two hours stopped vibration then return to the steady state without vibration.

Case II examining the effect of vibration from start of operating the cold and hot circuits for ascending two set frequencies.

Case III examining the effect of vibration from start of operating the cold and hot circuits for two set descending frequencies.

All of the air fluid thermal properties have been evaluated at the mean fluid temperature, film temperature (Tf), which is called film temperature as given in the following equation:

$$T_f = \frac{(T_h + Tc)}{2} \quad \text{[Holman]} \tag{1}$$

Dimensionless parameters are calculated depending on film temperature. The nominal heat flux with the characteristic dimensionless quantities is given in table (1).

5. Discussion of Results

5.1 Temperature Fields

The experimental temperature field has been validated through comparisons with 3D (three dimensions) experimental results of thermovibrational convection in case at normal gravity and vibration in cubic cell for the cases of (2and 4) Hz with those of V. Shevtsova et al ,(2010) since these studies are carried out in small cubic test cell (L=3mm) using air as the working fluid and at the same range of frequencies are presented in table (2).The comparison shows good agreement between the two experimental works, the percentage deviation was found to be about (8.25% at 2Hz, and 7.6% at 4 Hz).

5.1.1 Case I:

Figures (4) show the experimental mean temperature of each plain variation with time. The air temperature behavior of each plane throughout the entire cubic enclosure for the case (transient, reach to steady state, applied the various frequencies and then reach to steady without the effect of vibration. The heat flux was fixed at two values ($q^{"}$=85W/m^2) for frequencies (2,4&8)Hz and ($q^{"}$=946.017W/m^2) for frequencies(3,6&9)Hz as shown in figure (5). The periodic when the cavity under the effects of vibration (two hours) at each frequency. It is clear that the temperatures distribution, (along the total of six plain locations were measured from the cold wall $(Z = 5mm)$ to the hot wall $(Z = 115mm)$) is significantly influenced by vibration of the cavity.

In addition, note that the temperature distribution of the plains are more closer to each other when compared with the steady state before vibration. At the forcing frequency increases the fluctuating amplitudes of temperatures are substantially augmented at around f=(2&4)Hz for $q''=85W/m^2$ and f=(3&6)Hz $q''=946.017W/m^2$ as portrayed in figure 4(a & b) and figure 5(a & b). It decreases with further increase of the forcing frequency as shown in figure 4(c) and 5(c).

5.1.2 Case II&III:

At count up to be start, there are marked fluctuation amplitudes for each mean temperature at all plains, especially at starting of vibration. As the frequency increases, the fluctuating amplitudes of temperature increases at around at all planes in the cubic enclosure for frequencies (2&4) Hz at$q''=85W/m^2$. It decreases with further increase of frequency 8Hz, the air temperatures fluctuate periodically by the sinusoidal oscillation of the cubic enclosure as portrayed in figure (6-a).

In the other case for countdown to be start, when the vibration frequency (8Hz) there is an increase in temperatures of air in cavity with fluctuating amplitudes and continuous this behavior at the frequencies (4&2) Hz as presented in the figure (6-b).

At $q''=946.017W/m^2$ the time-dependent behavior of temperatures fluctuation at each plains around the enclosure was the same at $q''=85\,W/m^2$ by increasing vibration frequency (3,6&9)Hz noted there is a large and sharp fluctuation in the behavior of temperatures of each plane in cubic enclosure at frequencies (3&6)Hz and decreasing the influence gradually by increasing the frequency at 9Hz to become the mean temperature behavior of each plane fluctuating sinusoidal considerably affected by the applied vibration as presented in figure (7-a) , the vice versa when starting vibration at high frequency, the decreasing in fluctuating amplitudes of temperatures with starting of high forcing frequency, when decreasing the frequency gradually that noticeable fluctuate and sharp as presented in figure (7-b).

5.2 Heat Transfer

The comparison of present work and that of several previous investigators at the no-vibration data. The agreement is seen to be good and shown in figure (8). Figures (9-a) & (9-b) show the variation of the time-average Nusselt number at $q''=85W/m^2$ and $q''=946.017W/m^2$ respectively. When comparison the effect of induced vibration for frequencies ascending and descending. It can be seen average Nusselt number more increasing for the ascending than for the descending frequencies.

5.4 Enhancement factor $E\ (\overline{Nu})$

In order to assess the heat transfer enhancement by an external oscillation, the enhancement factor is defined as follows:

$$E\ (\overline{Nu}) = \frac{(\overline{Nu})_p}{(\overline{Nu})_s} \qquad \text{(Sung Ki Kim al et 2002)} \qquad (2)$$

Enhancement factor (E) the ratio of the average Nusselt number with vibration to the Nusselt number at the steady state. Figures (10-a) and (10-b) portray comparison of heat transfer enhancement factor(E) with ascending and descending three different continuous frequencies (2,4&8) Hz at Raleigh number $7*10^7$and (3,6&9)Hz at Rayleigh number $4*10^8$ respectively.

One more things we can observe from these figures the enhancement factor (E) at all frequency at different Rayleigh number is increasing regularly for ascending vibration and is decreasing for descending vibration.

6 Conclusion

The conclusions of this investigation can be summarized as follows:

1. Natural convection heat transfer in an air-filled cubic enclosure has been shown to be increased with frequency increased. For each Raleigh number the percentage of increasing in the time-average Nusselt numbers with respect to frequency can be tabulated in table (3) and (4) as shown in figure(11-a) and (11-b).

2.In the high Rayleigh number case (Ra=4* 10^8), the gravitional thermal convection dominates, and the vibration does not enhance the heat transfer rate remarkably. In contrast, in the low Rayleigh number case (Ra=7*10^7), the vibration thermal convection is dominant, and the vibration enhaces the heat transfer rate significantly, as portrayed in in table (3) and (4).

3. The fluctuating amplitude of air temperature monotonically increased at low frequencies. However, it was little affected by the increase of oscillating amplitude of the enclosure at two amounts of heat flux.

4. It can seen from these figures the values of the average Nusselt number at ascending frequencies is in general higher than that descending frequencies for two cases of Rayleigh number.

5. the higher the vibration frequency is, the quicker the steady state is reached.

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Table (1) characteristic dimensionless

$q''_c(w/m^2)$	85	946
Pr	0.7282	0.7202
Gr	$8.24*10^7$	$5.55*10^8$
Ra	$7*10^7$	$4*10^8$

Table (2) the frequencies and amplitudes used in the experiment in normal gravity

	fr (Hz)	Amplitude(mm)	ΔT(℃)	Ra_{vib}
Present Work	2	30.65	21.8	26000
Shevtsova (2010)	2	60	20	31622
Present Work	4	60.8	18.5	300000
Shevtsova (2010)	4	40	20	56216

Table (3) $\overline{Nu}$ increasing at Ra=$7*10^7$ and Ra=$4*10^8$

Frequency (Hz)	$\overline{Nu}$	Percentage increasing according to vibrating frequency
2	9.5	14.5%
4	11.3	39%
8	14.2	60%
Ra=$4*10^8$		
3	33.5	9.5%
6	36.48	20%
9	45	37.6%

Symbol	Description	Units
α	Thermal diffusivity	m^2/s
A	cross sectional area (L*L)	m^2
β	Thermal expansion coefficient	1/K
f_r	Frequency	Hz
T_a	Ambient (atmospheric) temperature	K
I.P	power Supplied(I.P)= I.V $=Q_{conduction}+Q_{Radiation}+Q_{convection}$	*W*
I = AC	Current through heating strip	*Ampere*
V = AC	voltage measured	*Volt*
Qconduction	Conduction heat transfer rate$=U.A_h(T_h-T_a)=0.0088(T_h-T_a)$	W
R	Thermal resistance ,R $=\frac{1}{U}=\frac{1}{h_o}+\frac{X1}{K1}+\frac{X2}{K2}$	$m^2.K/W$
U	overall heat transfer coefficient	$W/m^2.K$
h_o	Exterior convective heat transfer coefficient (ASHRAE1981)	3.08 $W/m^2.K$
X1	Glass wool thickness	50mm
K1	Glass wool conductivity	0.038 $W/m.K$
X2	Aluminum thickness	10mm
K2	Aluminum conductivity (6061-T6)	167 $W/m.K$
Qradiation	Radiation heat transfer rate from specimen surface$=F_{12}*\sigma*(T_{hs}^4-T_{cs}^4)*Ac$	W
T_h	Temperature of hot surface	K
T_c	Temperature of cold surface	K
F	shape factor = $(\frac{2}{\varepsilon}-1)$	
$Q_{convection}$	Convection heat transfer rate calculated from energy balance method= $h.A_h(T_h-T_c)$	W
$h_{L_o i}$	$=\frac{q''}{(T_{hi}-T_c)}$	$W/m^2.K$
$\bar{h}$	Average heat transfer coefficient	*W/m2.K*
ΔT	Temperature difference	K
$Pr=\frac{\vartheta}{\alpha}$	Prandtl Number	-
$Gr=\frac{g\,\beta_r\,q''_c\,L_x{}^4}{k_r\,\vartheta_r{}^2}$	Grashof number	-
$Ra=\frac{g\,\beta_r\,q''_c\,L_x{}^4}{k_r\,\vartheta_r{}^2}$	Rayleigh Number	-
$Ra_{vib}\,\frac{1}{2}(\frac{b\omega\,\beta_f\,q''_c\,L_x{}^2}{K\upsilon})^2$	Pr 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

ν	Kinematics viscosity	m^2/s
ρ	Density	kg / m^3
$Nu = L_o \frac{h_{L_o}}{K_f}$	Local Nusselt number	-
$\overline{Nu} = \frac{L\overline{h}}{K_f}$	Average Nusselt number	-

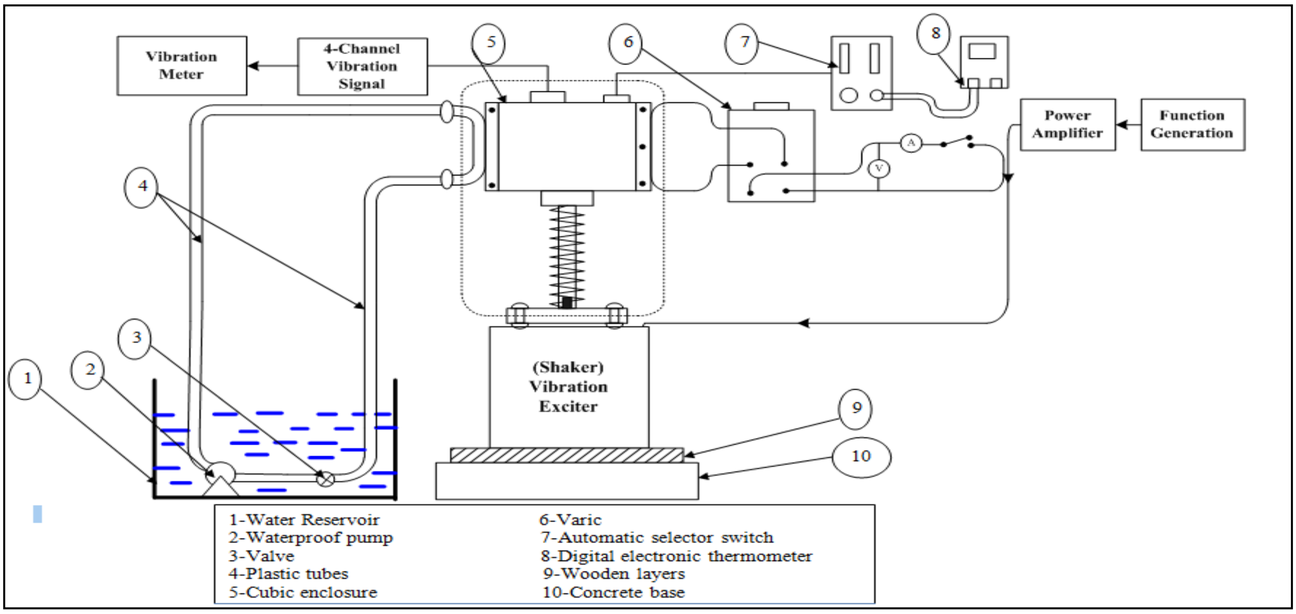

Figure 1. Schematic diagram for the testing rig with the implemented vibration

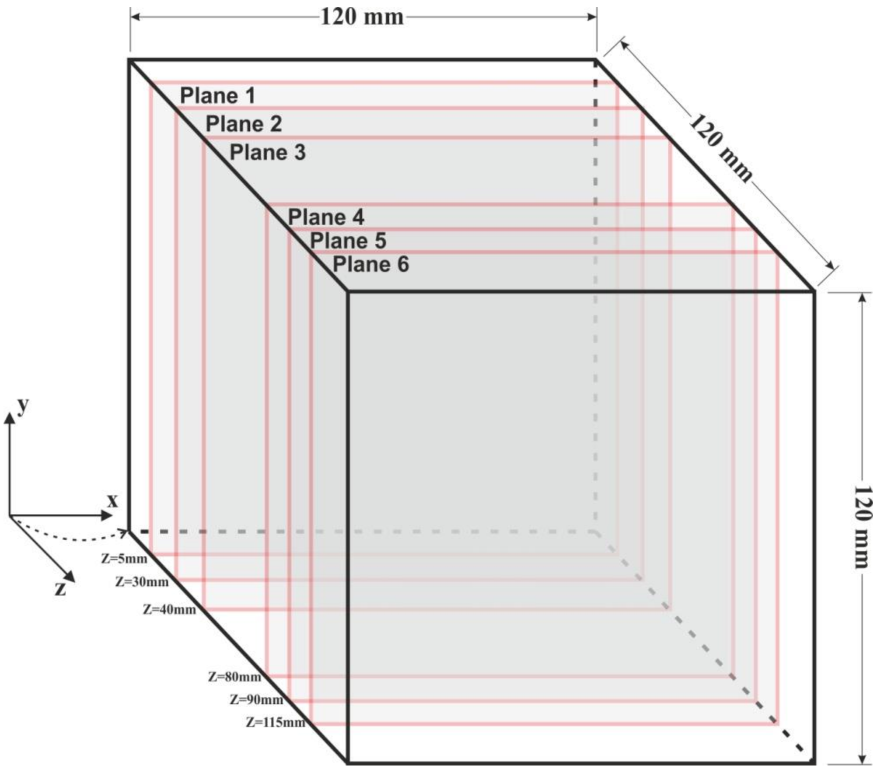

Figure 2a. Planes orientation for temperature measurement into space of the cubic cavity

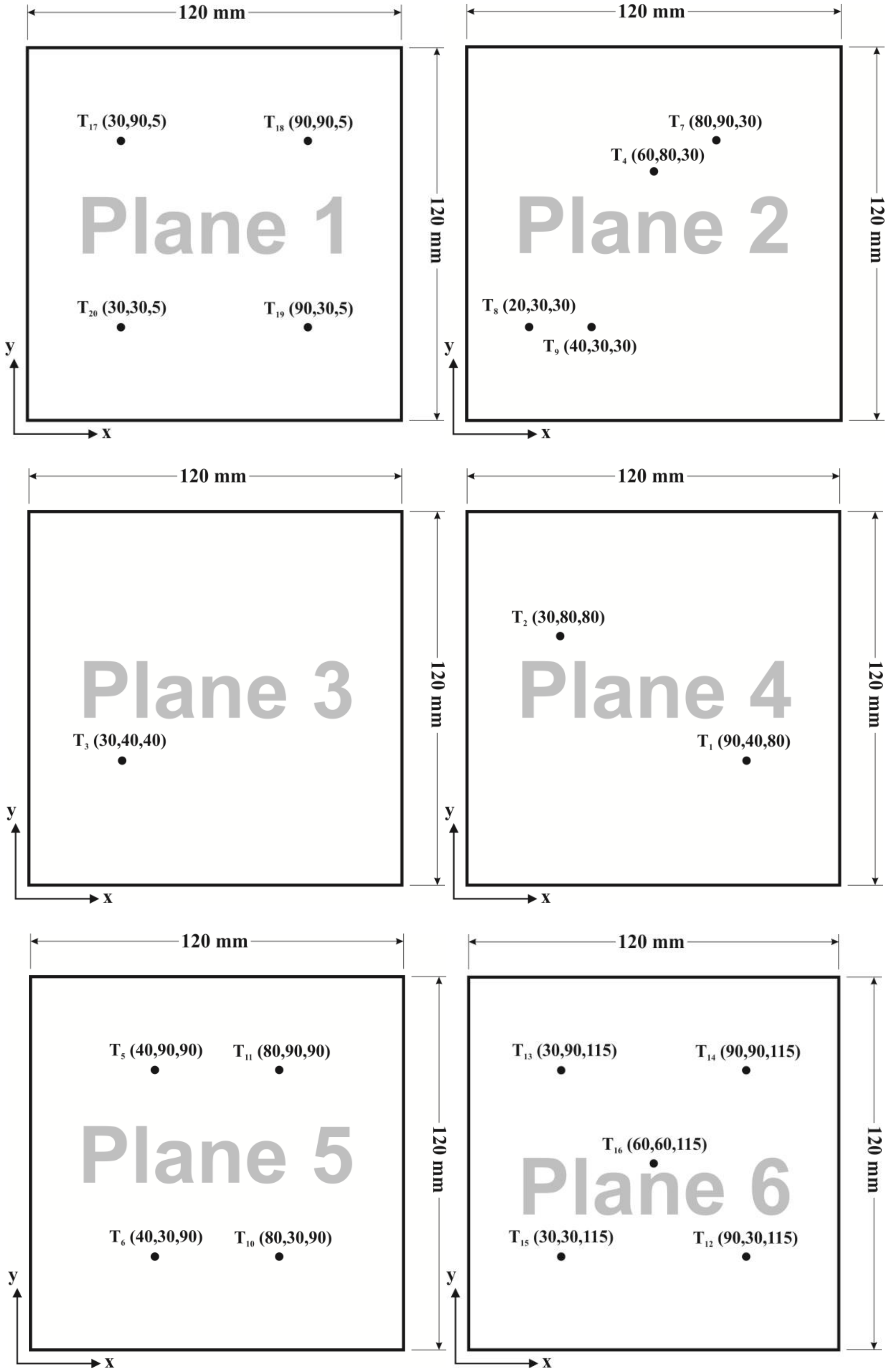

Figure 2b. Thermocouples orientation into each plane of the cubic cavity

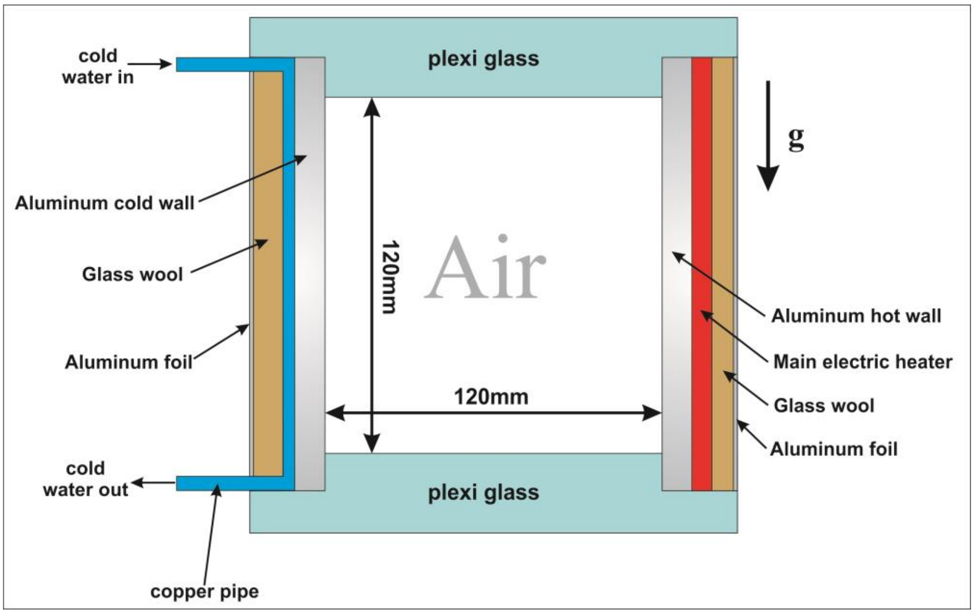

Figure3. Schematic diagram of the testing cell

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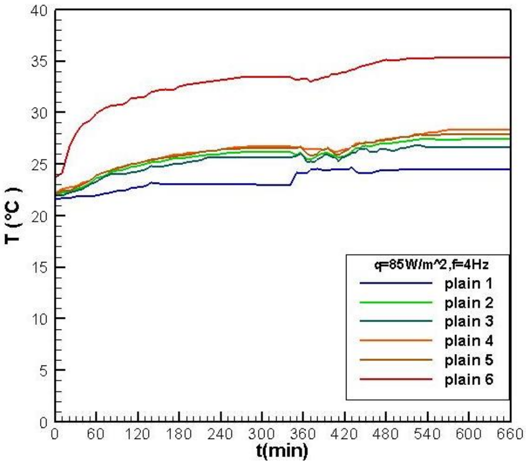
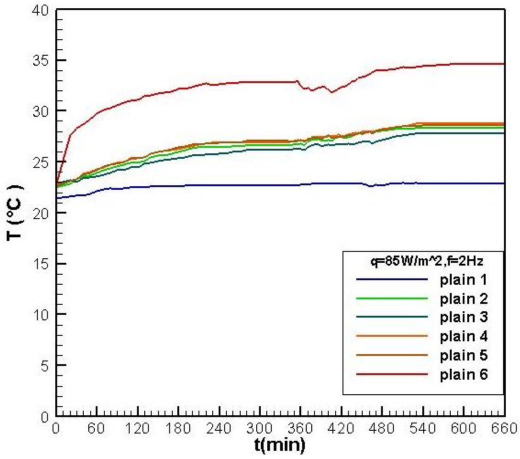
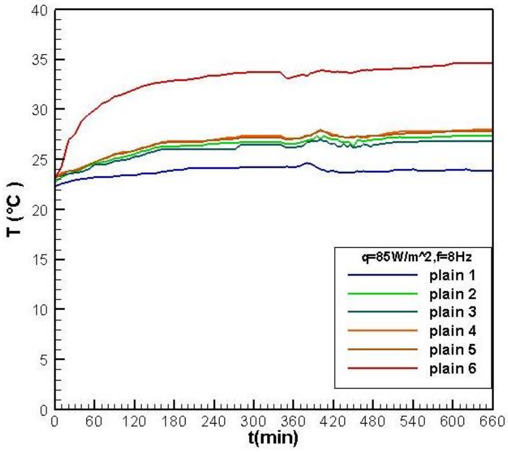


(a)

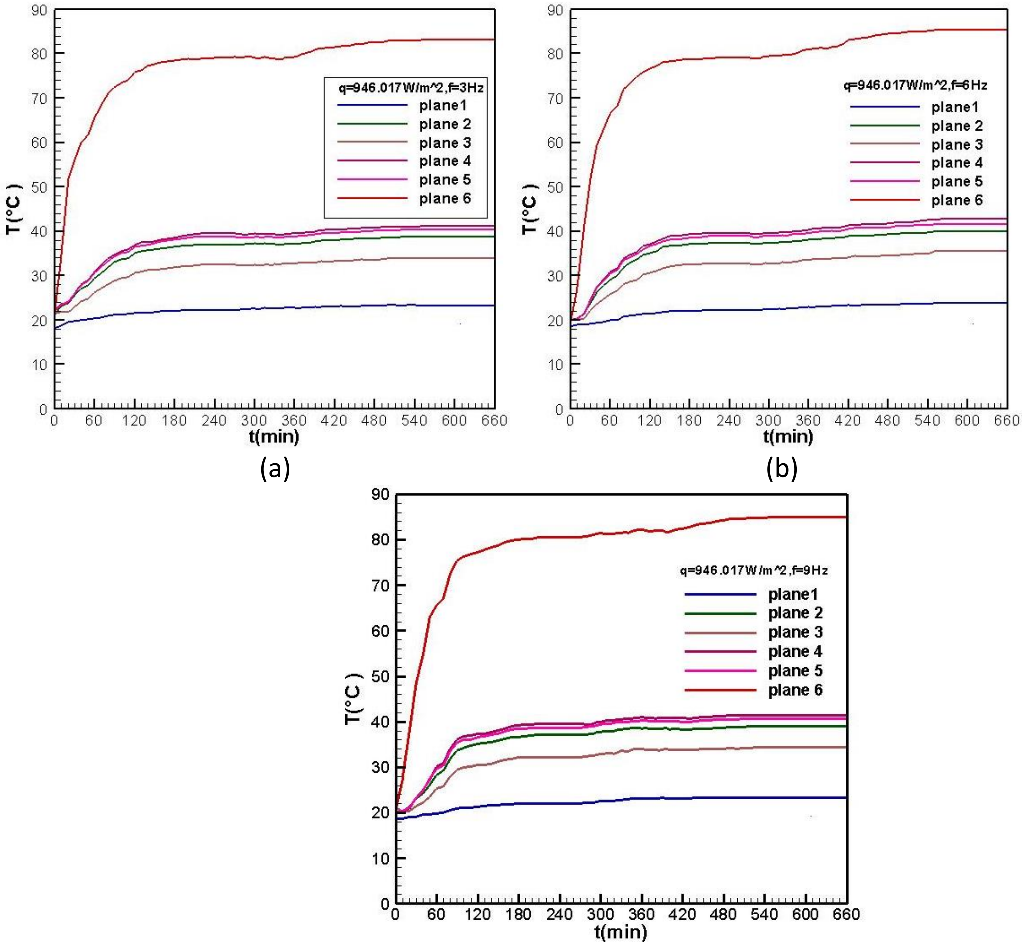


(b)



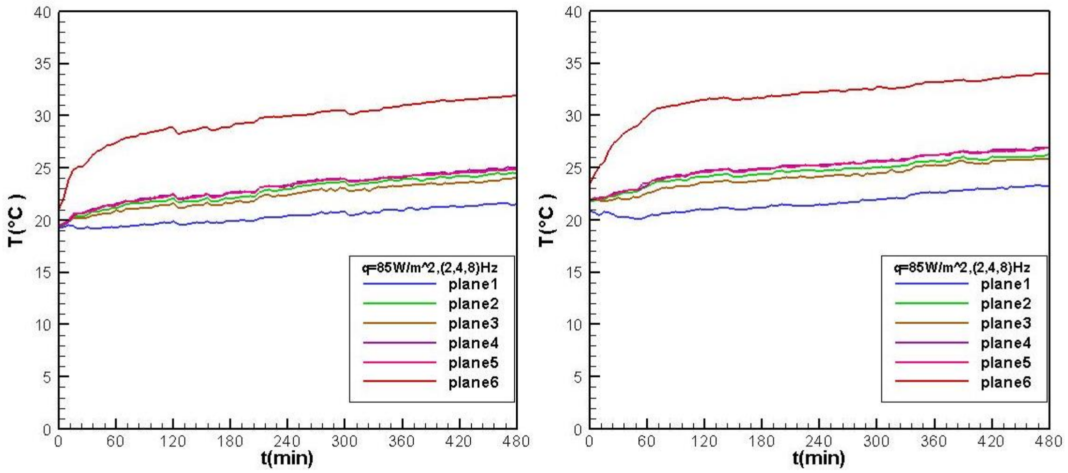
(c)

Figure 4. time-dependent behavior of temperature at $Ra = 7 * 10^7$ and q''=85W/m^2 time=(0-340)min without vibration, time=(340-480)min with vibration effect(a) $f = 2Hz$ (b) $f = 4Hz$ and (c)$f = 8Hz$, time=(480-660)without effect 0f vibration.



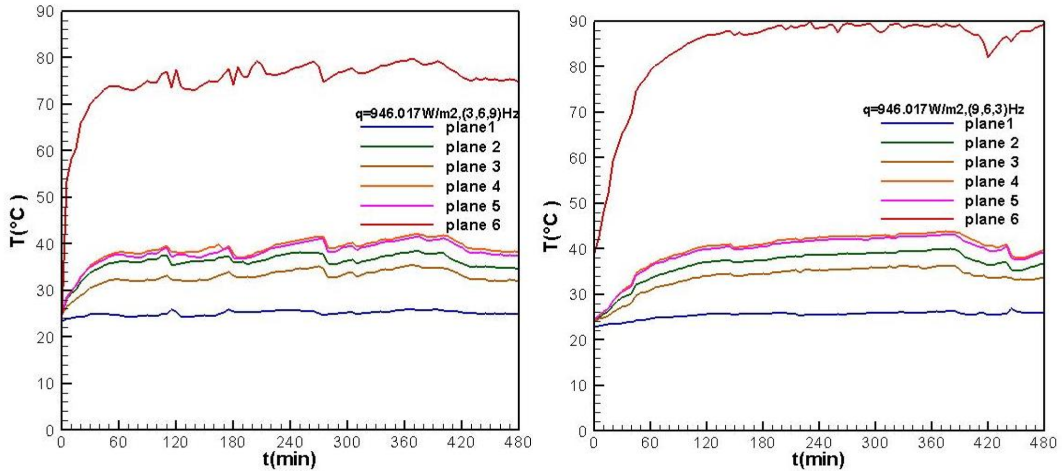
(c)

Figure.5 Time-dependent behavior of temperature at $Ra = 4 * 10^8$ and q''=946.017W/m^2 time=(0-270)min without vibration, time=(270-410)min with vibration effect (a) $f = 3Hz$ (b) $f = 6Hz$ and (c) $f = 9Hz$, time=(410-660)without effect 0f vibration .



(a) (b)

Figure.6 (a) Time-dependent behavior of temperature for ascending frequencies forcing vibration at time=(0-180)min, $f = 2Hz$, time=(160-320)min $f = 4Hz$ & $f = 8Hz$ at time=(320-480). (b) Time-dependent behavior of temperature for descending frequencies forcing vibration at time=(0-160)min, $f = 8Hz$, time=(160-320)min $f = 4Hz$ & $f = 2Hz$ at time=(320-480).



(a) (b)

Figure.7 (a) Time-dependent behavior of temperature for ascending frequencies forcing vibration at time=(0-180)min, $f = 3Hz$, time=(160-320)min $f = 6Hz$ & $f = 9Hz$ at time=(320-480). (b) Time-dependent behavior of temperature for descending frequencies forcing vibration at time=(0-160)min, $f = 9Hz$, time=(160-320)min $f = 6Hz$ & $f = 3Hz$ at time=(320-480).

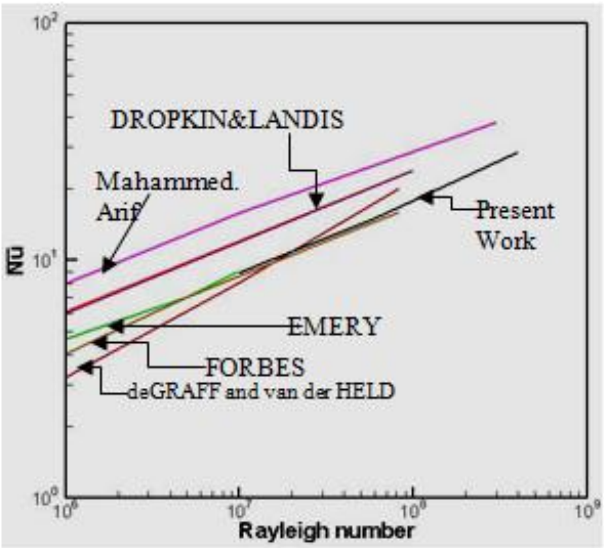


Figure.8 Log-log ($\overline{Nu}$ vs. Ra).The comparisons of heat transfer data with that of other investigators at no vibration.

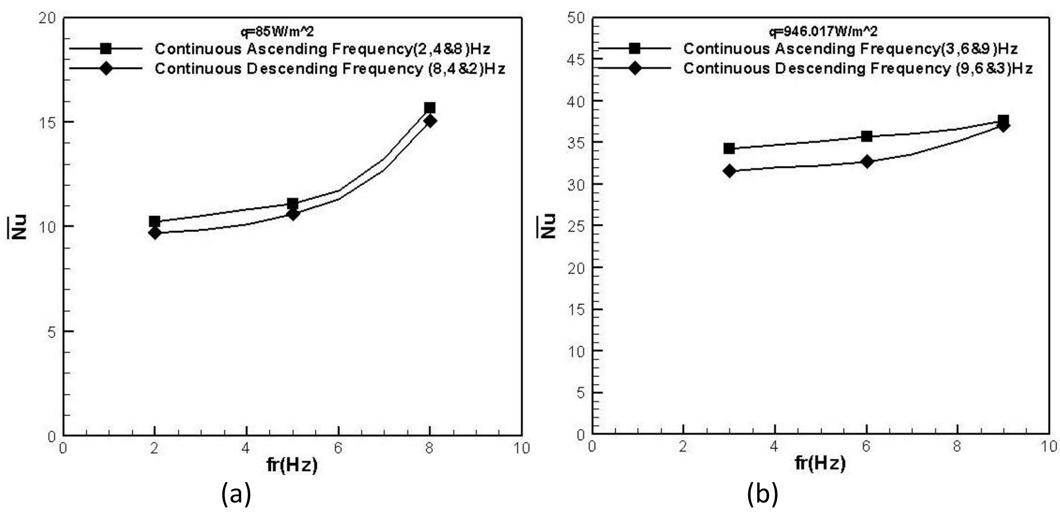


Figure (9) The influence of Continuous frequencies of experimental-average Nusselt number for (a) q''=85W/m^2 (b) q''=946.017W/m^2.

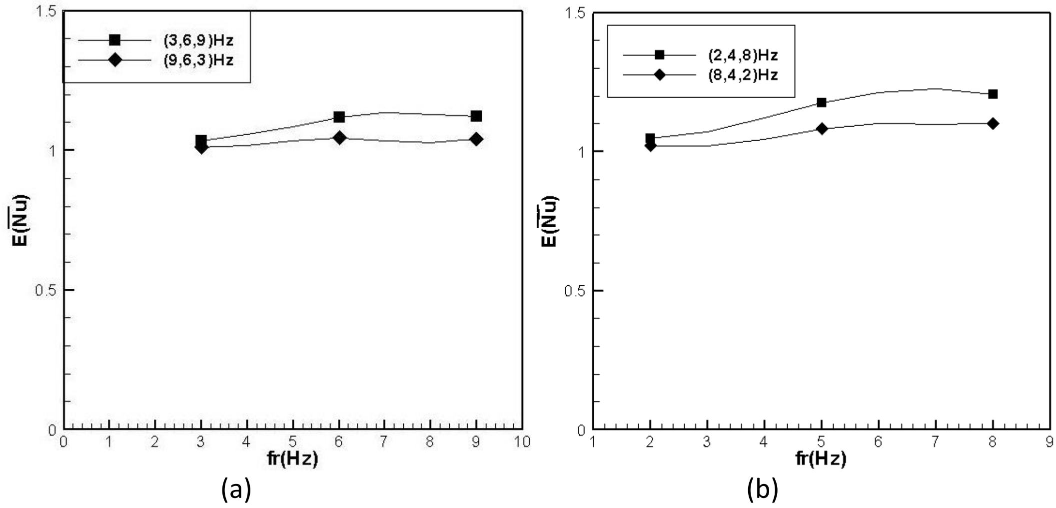


Figure.10 The variation of the heat transfer enhancement factor for various continuous frequencies (a) q''=85W/m^2 (b) q''=946.017W/m^2.

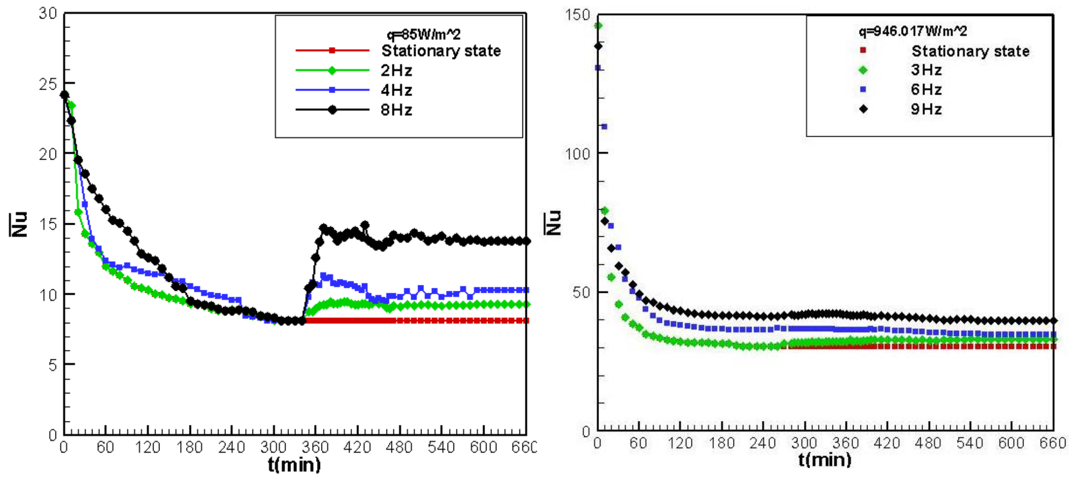


Figure. 11 The comparisons of variations the time- average Nusselt number for of Interrupted frequencies with the stationary state (a) q''=85W/m^2 (b) q''=946.017W/m^2.

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